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# Agricultural Tractor Powertrains: Fundamental Characteristics and Opportunities for Intelligent Control

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# Abstract

The use of microprocessor-based control systems on agricultural tractors has eased operator burden by allowing changes to tractor and implement settings to be made with little physical effort. However, maintaining the optimum tractor-implement settings whilst encountering the variable nature of agricultural conditions still requires a high level of operator skill, partly due to the need to adjust individual sub-system controllers. CAN-bus communication between electronically controlled vehicle sub-systems provided a new opportunity to enhance vehicle powertrain operation, by intelligently integrating control of the sub-systems. The aim of the project was to develop ways to improve the operational characteristics of a tractor powertrain, by investigating system behaviour, and identifying opportunities for intelligent control.

Market research was undertaken which highlighted power-split continuously variable transmissions as a credible alternative to powershift-type transmissions in specific specialist applications where the additional purchase price could be justified. However, there is little scientific evidence to suggest that there are significant improvements in overall vehicle performance to be gained through the use of a CVT tractor compared to a well operated powershift-type transmission. Improvements to gearshift quality and more intelligent use of the powertrain control features could ensure powershift-type transmissions remain competitive for the foreseeable future.

A dynamic mathematical powertrain model was developed for a 100kW, 16 speed semi-powershift transmission, four wheel drive tractor based on fundamental Newtonian principles. With the addition of implement models, this allowed accurate representation of the tractor-implement system and provided a platform to develop improved vehicle control strategies. Validation of the model with experimental data showed it was an accurate representation of the real system.

The steady state and transient field performance of the tractor operating with a mouldboard plough, a power harrow and a laden trailer was determined for a number of tractor-implement configurations across a range of conditions. This provided a

large dataset for this vehicle for use in this, and other investigations. The level of powertrain loading for field experiments was found to be influenced by soil type, implement working width and depth as well as forward speed and engine speed. For the road investigation, the surface quality and terrain were major influencing factors on performance. It was found there was considerable variation in tractor response to the different gearshift types experienced in the semi-powershift transmission: the non-powershift changes being severe, particularly during downshifts; double-swap powershifts were markedly more severe than single-swap shifts.

A unique investigation of the tractor driveline torque loss characteristics across the full operating spectrum using the axle dynamometer identified that the torque losses for this transmission are predominantly speed, rather than torque related. A mathematical model was developed to predict driveline torque losses from transmission output speed, flywheel torque and the number of power-transmitting gears in mesh. The axle dynamometer was also used to successfully replicate field loading patterns in real time.

Throughout this investigation a number of undesirable powertrain characteristics were identified. Potential improvements to vehicle performance through the development of solutions to these characteristics have been made either through analysis of field data, experiments with the axle dynamometer, or using the developed mathematical model.

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"Do not go where the path may lead,
go instead where there is no path and leave a trail"

**Ralph Waldo Emerson (1803-1882)**

# Table of Contents

# List of Figures

# List of Tables

# Nomenclature

(unless otherwise stated in the text)

| | | |
|---|---|---|
| $e(t)$ | Error signal from the feedback loop | Nm |
| $g$ | Acceleration due to gravity (9.81) | $m/s^2$ |
| $k_1$ , $k_c$ | PID proportional coefficient | |
| $k_2$ | Integral coefficient | 1/second |
| $k_3$ | Derivative coefficient | second |
| $k_{crit}$ | Limiting value of gain for stability | |
| $m(t)$ | Controller output | Nm |
| $r$ | Rear wheel loaded radius | mm |
| $r_a$ | Rear axle ratio | |
| $r_d$ | Driveline gear ratio | |
| $r_o$ | Distance from axis of pendulum to wheel centre of gravity | m |
| $r_{pe}$ | ratio of engine speed to P.T.O. speed | |
| $r_t$ | Transmission gear ratio | |
| $r_{te}$ | Ratio of transmission output to engine speed | |
| $s$ | Expression in the Laplace domain | 1/second |
| $t$ | Expression in the time domain | second |
| $t_{CALC}$ | Calculated t-test statistic | |
| $v_a$ | True forward speed | km/h |
| $v_t$ | Theoretical forward speed | km/h |
| | | |
| $A_{HT}$ | Theoretical power harrow workrate | ha/h |
| $A_{PT}$ | Theoretical plough workrate | ha/h |
| $B_S$ | Boost status | |
| $B_{\%}$ | Boost percent torque | % |
| $C_{RR}$ | Coefficient of rolling resistance | |
| $D_{HA}$ | Average harrow working depth | mm |
| $D_{HL}$ | Harrow left side working depth | mm |

| | | |
|---|---|---|
| $D_{HR}$ | Harrow right side working depth | mm |
| $D_P$ | Plough working depth | mm |
| FBF | Feedback factor | rpm/mg/stroke |
| G | Gear number | |
| $G_c(s)$ | Transfer function of a standard PID controller | |
| $G_m$ | Number of torque transferring meshed gear pairs | |
| $H_A$ | Total horizontal (draught) force | kN |
| $H_L$ | Left link horizontal (draught) force | kN |
| $H_R$ | Right link horizontal (draught) force | kN |
| $H_{RR}$ | Rolling resistance force | kN |
| $H_T$ | Top link horizontal (draught) force | kN |
| $I_D$ | The lumped inertia of the differential components | $kg.m^2$ |
| $I_E$ | Engine (including flywheel) inertia | $kg.m^2$ |
| $I_{DF}$ | Total driveline inertia referenced to the flywheel | $kg.m^2$ |
| $I_{TA}$ | The lumped inertia of the front transmission section | $kg.m^2$ |
| $I_{TB}$ | The lumped inertia of the middle transmission section | $kg.m^2$ |
| $I_{TC}$ | The lumped inertia of the rear transmission section | $kg.m^2$ |
| $I_W$ | Wheel inertia about its centre of rotation | $kg.m^2$ |
| $I_{WF}$ | Wheel inertia referenced to the flywheel | $kg.m^2$ |
| $I_{WO}$ | Wheel inertia about the point 'o' | $kg.m^2$ |
| $M_I$ | Implement mass | kg |
| $M_T$ | Tractor mass | kg |
| $M_W$ | Rear wheel & tyre mass | kg |
| $P_{crit}$ | Time period of controller output oscillation | second |
| $P_D$ | Drawbar power | kW |
| $P_F$ | Flywheel power | kW |
| $P_L$ | Lost power (fieldwork) | kW |
| $P_{LOSS}$ | Driveline power loss | kW |
| $P_P$ | P.T.O. power | kW |
| Q | Fuel quantity injected | mg/stroke |
| R.M.S. | Root mean square | |
| S | Wheelslip | % |

| | | |
|---|---|---|
| S.E.M. | Standard error of the mean | |
| $T_A$ | Total axle torque | kNm |
| $T_{ACC}$ | Acceleration torque | Nm |
| $T_{BH}$ | P.T.O. torque (external sensor, ref at P.T.O.) | Nm |
| $T_D$ | Flywheel draught torque | Nm |
| $T_d$ | Term used in the calculation of $k_3$ | second |
| $T_{DV}$ | Vehicle torque demand | % |
| $T_E$ | Engine torque | Nm |
| $T_{EA}$ | Equivalent axle torque | kNm |
| $T_{EF}$ | Equivalent flywheel torque | Nm |
| $T_{E\%}$ | Engine torque | % |
| $T_F$ | Flywheel torque | Nm |
| $T_{FA}$ | Flywheel acceleration torque | Nm |
| $T_{F\%}$ | Flywheel torque | % |
| $T_i$ | Term used in calculation of $k_2$ | second |
| $T_L$ | Torque at the left rear axle | kNm |
| $T_{LOSS}$ | Torque loss through the driveline | Nm |
| $T_{PTO}$ | P.T.O. torque from tractor sensor (ref at flywheel) | Nm |
| $T_R$ | Torque at the right rear axle | kNm |
| $V_A$ | Total vertical force | kN |
| $V_L$ | Left link vertical force | kN |
| $V_R$ | Right link vertical force | kN |
| $W_H$ | Power harrow working width | m |
| $W_P$ | Plough overall working width | mm |
| | | |
| $\delta$ | Engine speed droop | % |
| $\delta_I$ | Engine speed droop mode | |
| $\varepsilon_r$ | Rear hitch rockshaft position | % |
| $\varepsilon_t$ | Foot throttle position | % |
| $\eta$ | Driveline efficiency | % |
| $\eta_T$ | Tractive efficiency | % |
| $\theta$ | surface slope | radians |

| | | |
|---|---|---|
| $\tau$ | Period of oscillation | seconds |
| $\omega_E$ | Engine output / transmission input speed[1] | rpm |
| $\dot{\omega}_E$ | Engine acceleration | rad/s$^2$ |
| $\omega_F$ | Fan speed | rpm |
| $\omega_{FL}$ | Full-load engine speed | rpm |
| $\omega_{NL}$ | No load engine speed | rpm |
| $\omega_P$ | P.T.O. speed | rpm |
| $\omega_T$ | Transmission output speed | rpm |
| $\dot{\omega}_T$ | Transmission output acceleration | rad/s$^2$ |

[1] Only whilst a gear is engaged does this also give transmission input speed, measurement is made at the engine crankshaft.

# 1 Introduction

## *1.1 Background*

Humans as power units are limited to less than 0.1kW continuous output and therefore are worth almost nothing as a primary source of power (Liljedal *et al*, 1989). In order to receive adequate return for their labour, agricultural workers must control power rather than being a power source. Agricultural tractors have been a key element in the mechanisation of farm work, helping to achieve the three primary objectives of farm mechanisation as defined by Goering and Hansen (2004):

1. to reduce the drudgery of farm work;
2. to increase the productivity of farm workers;
3. to increase the timeliness and quality of farm work.

Agricultural tractors came into existence at the start of the 20$^{th}$ Century and by 1920 were beginning to replace the horse as the main power source on farms. Whilst some development occurred during the inter-war years, it was the Second World War and the period immediately afterwards which saw a massive increase in the number of tractors as attempts were made to feed the population with a reduced workforce available. During this time features synonymous with the modern tractor began to emerge, such as diesel engines, pneumatic tyres, three-point (3pt.) hitch, power take-off (P.T.O.) and implement draught control. With this basic form established, further tractor developments have taken the form of increasing average engine power (Agricultural Engineers Association, 2005) and an increase in the sophistication of the tractor features, aiming to improve manpower productivity in the face of reduced agricultural profitability and a shrinking labour force.

Britton (1989) reported that the number of people engaged in agriculture in the UK had fallen from 2 million in 1851 to 1.1 million in 1951, by 1986 this figure had fallen to 0.6 million people. Fortescue (2005) reported that, as of December 2004, there were only around 100,000 full-time agricultural employees in the UK and a further

120,000 part-time. This massive reduction in labour occurred at the same time as a huge increase in agricultural production: this has been as a result of many technological developments, not least agricultural mechanisation. The United States National Academy of Engineering (2000) created a shortlist of the greatest engineering achievements of the 20$^{th}$ Century; agricultural mechanisation featured at number seven, with much of the praise being directed at the development of the tractor.

During the evolution of the tractor, continual effort has been made to improve tractor safety, comfort and ease of use for the operator. This took a step-change during the early 1980s with the introduction of electronics to tractors, initially through the introduction of electro-hydraulic implement control in large tractors (Cox, 1988), avoiding the need to bring hydraulic hoses and control valves into the tractor cab. Around the same time basic electronic sensing and cab display systems were introduced, for example, to accurately display forward speeds to the operator, the driving force behind these changes being the need to improve accuracy of spraying and spreading fertiliser. A short time later electronics began to be used for control purposes on major elements of the tractor, for instance in the transmission to make gearshifts, as described by Cox (1997). In this case the operator would initiate the shift, after which a microprocessor-based controller would supervise the operation of the clutches to change gear, whilst preventing shifts which could be potentially damaging to the transmission.

A major milestone was the introduction of the Massey Ferguson 3000 series tractors in 1986. This range of tractors featured a level of electronics never previously known in mass market tractors. In addition to controlling the powershift gears in the semi-powershift transmission, functions such as the differential lock and four wheel drive were automatically controlled to prevent driveline damage, for example disengagement at high forward speeds. The tractor featured electronic implement draught control (EDC) and slip control, by monitoring true forward speed and wheel speed. The cab boasted a control and display panel to allow the operator to monitor parameters, such as engine speed and set limits, for example maximum wheelslip.

Since the introduction of the 3000 series, the complexity of agricultural tractors has continued to increase, both in terms of mechanical vehicle sub-systems and their control. The now-widespread use of microprocessor-based electronic controllers allows far greater flexibility than previous methods, although the tendency remains for designers to treat each vehicle sub-system (such as the engine or transmission) as an individual module. The introduction of Controller Area Networks (CAN) (see Section 3.1) has helped progress towards the integration of vehicle sub-systems and has reduced wiring complexity. However CAN-bus still tends to be limited to displaying information to the operator and performing safety based functions, rather than integrating the various tractor-implement systems as proposed by Scarlett (1993).

Microprocessor-based controllers have allowed skilled operators to configure tractor sub-systems and implements easily, according to the required task and given conditions. However, the variable nature of the agricultural operating environment means that conditions rarely remain at the levels for which the implement was originally configured. The most common causes of this phenomenon, known as dynamic load variation, could include changes in soil strength; changing the implement draught force and changes in topography. If dynamic load variation is severe enough operator intervention may be required to maintain progress. This intervention can itself result in a sudden change in loading - a load transient. Examples of load transients include a change in gear or a change in throttle position thereby changing engine speed.

If the effects of these dynamic load variations and load transients could be minimised, through intelligent control of the tractor powertrain, the driveability and comfort of the tractor would be improved, helping to maintain progress or optimise fuel consumption, depending on the desired strategy, whilst making the operator's task easier or allowing him to concentrate on other aspects of the operation (Scarlett, 2001). The consequential effect of optimising the control of the powertrain would be the enhancement of component life due to uniform loading and a reduction of peak loads.

## 1.2 Definitions

There are numerous conflicting terms describing the various physical components used to generate and transmit power to the ground. In order to avoid confusion a number of terms are defined and used throughout this investigation, drawing from the terminology used by Liljedal *et al* (1989) and Gillespie (1992), shown in Figure 1-1.

1. Powertrain

The powertrain is all the elements of a vehicle responsible for the generation and transmission of power to the wheels or the P.T.O. shaft. The primary elements of a powertrain include the engine, clutch, transmission, driveshaft, differential and axles.

2. Driveline

The driveline is very similar to the powertrain, apart from the exclusion of the engine. 'Drivetrain' also describes the same components.

3. Transmission

The transmission is the part of the driveline responsible for making changes to the rotational speed and, inversely, the torque transmitted.

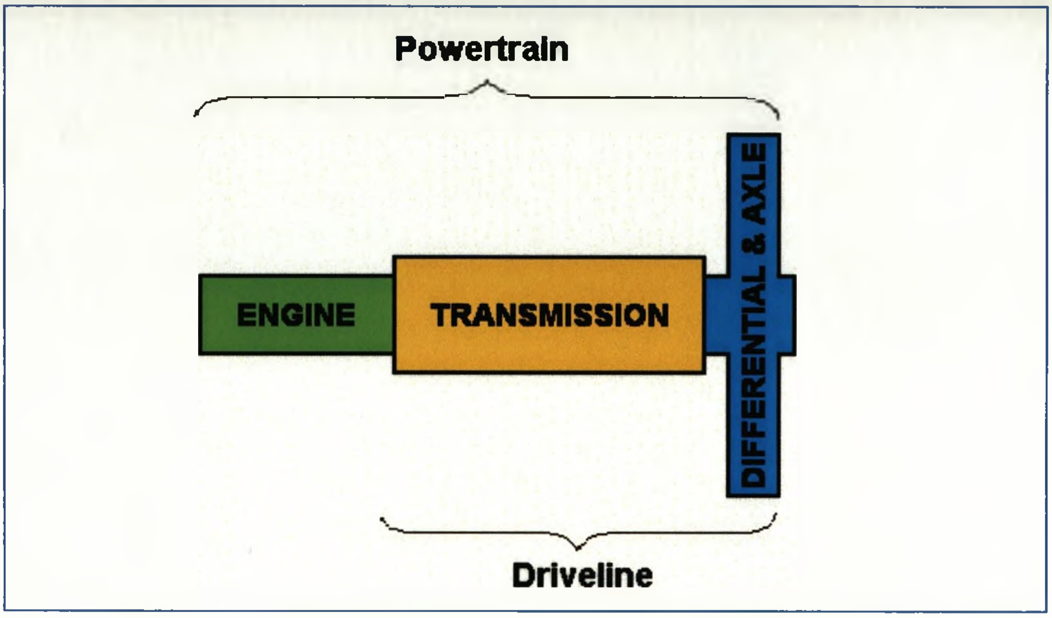



*Figure 1-1 – Tractor powertrain components*

## 1.3 Aim

The overall aim of this investigation was to develop methodologies to improve operational characteristics of agricultural tractor powertrains by investigating system behaviour and identifying opportunities for intelligent control.

## 1.4 Objectives

1. To develop methodologies to quantify and replicate engine and transmission loadings experienced during typical vehicle operation.
2. To develop mathematical representations (models) of tractor engine, transmission and driveline systems.
3. To utilise derived vehicle system models and refined loading histories to:
   - investigate vehicle response to both dynamic loading and load transients; and
   - develop alternative engine and transmission control strategies, to improve vehicle operating characteristics in specific environments.
4. To develop techniques to acquire information regarding vehicle local operating environment to permit adaptation of desired vehicle system characteristics.
5. To investigate the European market for agricultural tractors and to consider the technical performance of the transmission fitted to the test tractor against other transmission systems available.

## 1.5 Outline Methodology

This investigation can be divided into five major elements, which when brought together present a tractor powertrain model and sufficient data to allow the development of improved control strategies. These elements are:

1. To undertake a review of agricultural tractor driveline hardware and related control strategies, to investigate likely future transmission developments and to identify the key control requirements for an intelligent control system.
2. To conduct experiments in order to quantify the types, magnitudes and duration of loading experienced by the tractor powertrain whilst undertaking representative tasks for the size of vehicle. Two significant types of loading must be considered: dynamic load variation and load transients likely to be experienced whilst undertaking a given task, with a predefined implement configuration.
3. To develop a mathematical model to investigate the theoretical behaviour of the tractor powertrain. This model will be developed from first principles with the use of test and manufacturers' data, as appropriate, and validated with field data.
4. To review the performance and impact of the Continuously Variable Transmissions (CVT) in agricultural tractors, both from the user and manufacturer perspectives. To consider the cost-benefit relationship between this transmission design and the more traditional semi and full powershift transmissions, and to consider the likely future direction of the European market.
5. To identify powertrain control issues and to suggest modifications in order to improve the overall behaviour and performance of the tractor.

## 1.6 Project Structure

A structure of how the elements of the investigation link together is presented in Figure 1-2. This diagram also indicates the chapter location of each part of work.

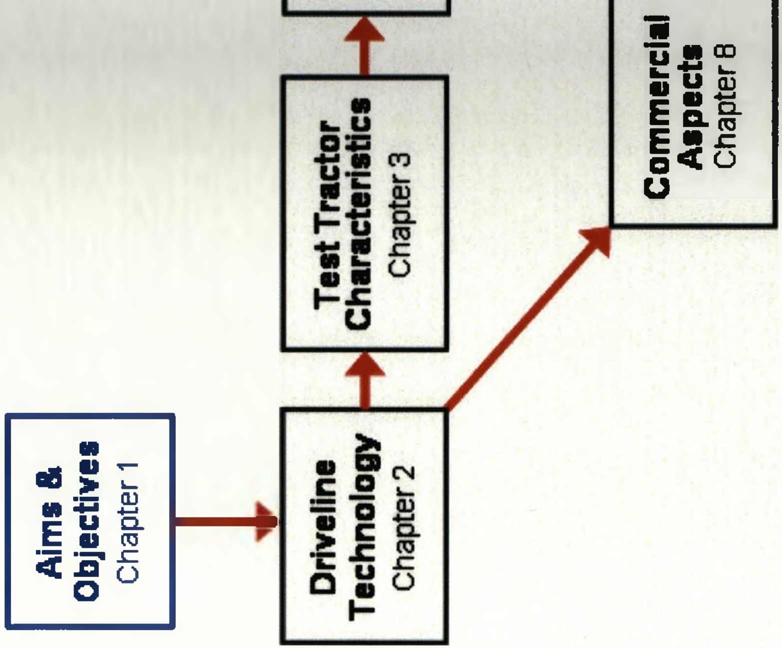
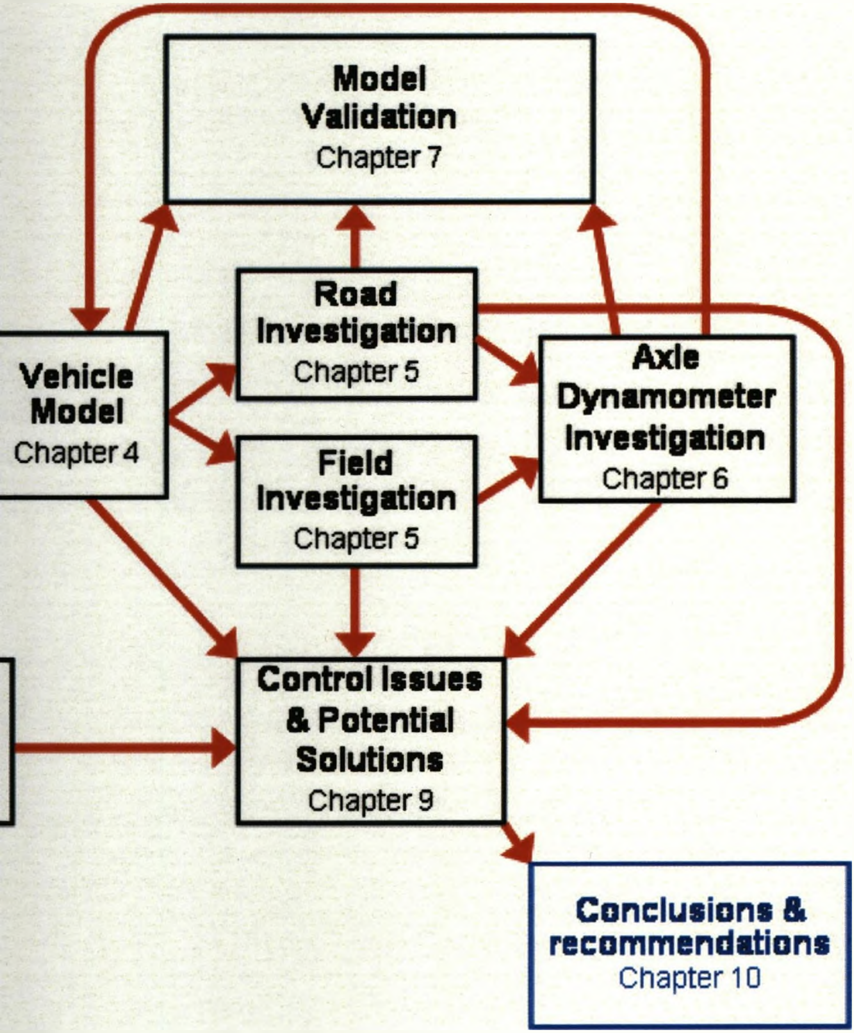



*Figure 1-2 – Project Structure*


Model
Validation
Chapter 7
Road
Investigation
Chapter 5
Vehicle
Model
Chapter 4
Axle
Dynamometer
Investigation
Chapter 6
Field
Investigation
Chapter 5
Control Issues
& Potential
Solutions
Chapter 9
Conclusions &
recommendations
Chapter 10

# 2 Review of Driveline Technology

A review of driveline technology has been undertaken, to put the test vehicle into context with respect to other tractors, and to provide a basis for the identification of powertrain system characteristics and development of intelligent control strategies. The major differences between tractor drivelines concern the transmission design and control system (i.e. the hardware and the control software). This section reviews the different transmission types, their origins and their control features, currently available on UK-specification tractors. Technical comparisons between different transmission types are made and consideration is given to potential future tractor transmission designs.

The tractor driveline, following Liljedal *et al* (1989), can be defined as having three main functions:

1. to transmit power from the engine to the wheels, P.T.O., hydraulic pump and other auxiliary devices;
2. to convert the engine torque and speed into the torque and speed required by the particular drive;
3. to provide means for operator control, through clutches and speed ratio selection for the traction and P.T.O. drivelines.

The majority (90%+) of tractor transmissions are of the stepped-ratio type, the main variations being in the number of forward speeds available and the method of gear selection. The last ten years have seen a rise in the popularity of power-split CVTs for tractors. This allows the link between engine speed and forward speed, fundamentally inherent with stepped-ratio transmissions, to be broken.

## 2.1 Mechanical Transmissions

### 2.1.1 Sliding Mesh

The principle of sliding mesh gear changes was first used in a motor vehicle gearbox in 1895 (Nunney, 1998). Gear engagement is achieved by sliding a spur gear along a splined shaft until it meshes with its opposite gear. Control is entirely mechanical, using gear levers. A high level of operator skill is required to effect a gear change whilst the vehicle is moving, in order to synchronise the two shaft speeds allowing the change to be made without grinding or 'crashing' the gear teeth together. Alternatively, the tractor must be stationary when the change is made. This transmission is very efficient, but as it is not easy to use it only tends to feature on lower-cost utility tractors. It is still produced by Massey Ferguson in Turkey for use in low-specification tractors for sale worldwide and Mahindra presently produce tractors with sliding mesh transmissions for sale in the USA (Mahindra, 2005).

### 2.1.2 Constant Mesh

In constant mesh transmissions gears are maintained permanently in mesh, but at least one of the pair is free running on its shaft. To engage drive the gear is locked to the shaft, usually through a toothed collar (dog clutch). As gears are constantly meshed, helical rather than spur gears can be used. This both reduces the running noise and increases the potential load capacity of the transmission. Operation of the shift collar is still done mechanically, therefore the same shift problems occur as with sliding mesh transmissions. As with sliding mesh transmissions, this design also tends to be widely used for simple, lower-cost and specification products of many global tractor manufacturers.

### 2.1.3 Synchromesh

The issue of shifting constant mesh gears whilst the vehicle is moving was solved by the introduction of male and female cones, one on the gear and one on the shaft. As the ratio change is made (mechanically) the cones meet prior to the collar and therefore synchronise the speeds of the two elements, overcoming the inertia of the downstream driveline components. Synchronising the shift elements allows changes to be easily made whilst the vehicle is moving, but at an increased cost and reduced efficiency. This increased cost and complexity, up until relatively recently, resulted in many tractors not featuring complete synchromesh transmissions. Often the range changes or the lowest gears were non-synchromesh. This practice is not common today due to improvements in synchroniser design and cost reduction measures.

Synchromesh transmissions form part of many modern tractor transmissions, although the gear selection method has been modernised substantially. Pure mechanically-actuated synchromesh transmissions tend to feature upon smaller, low specification agricultural tractors (up to 70kW), often as a transmission option aimed at livestock farmers. A typical example is the Synchro Command™ 12F/12R transmission for the New Holland TS-A tractors (New Holland UK Ltd, 2004). Two gear levers are used: one to select one of four gears, the other to select one of three ranges.

Even for livestock farms the popularity of mechanical synchromesh transmissions has diminished substantially in recent years. This is due to the increased reliability and functionality available with the more sophisticated transmissions and the increased use of hydraulics and electronics to improve transmission control.

## 2.2 Semi-Powershift Transmission (Single Ratio Change)

The need for tractors to perform a variety of tasks led to an increase in the number of transmission gear ratios available, in order to maximise operational efficiency (Tinker, 1993). The variable load conditions encountered in agriculture necessitate frequent ratio changing if high operational efficiency is to be maintained (Jordan *et al*, 1989). Mechanically-actuated transmissions require the clutch pedal to be depressed, thereby interrupting power transfer to the wheels during a gearshift. This reduces tractor performance and often results in less shifts being made, thereby allowing the tractor to operate below its optimum configuration. The natural progression for transmissions was the provision of a means for the operator to change speeds whilst retaining power transfer and momentum in each of the transmission gears. This allowed the operator to reduce the tractor forward speed when required, for example during changes in soil resistance or crop volume, without loosing momentum as would occur with a conventional gearshift.

The first of the two-speed semi-powershift transmissions was the 'Torque Amplifier' launched by International Harvester in 1954 (Renius, 1992). Other large manufacturers soon followed suit with Massey Ferguson launching the 'Multi-Power' transmission in 1962 (Elfes, 1961), John Deere the 'Hi-Lo' transmission in 1967 (Roberts, 2002b) and Ford the 'Dual Power' system in 1973 (Roberts, 2002a). All these were add-on elements to an existing transmission and, with the exception of 'Multi-Power', they operated on the same principle, namely a planetary gearset where two elements are locked to provide a direct drive (1:1) or one element is locked to provide a reduction in forward speed. A multi-plate clutch is used to select the direct drive ratio, with the under drive ratio being actuated by disengaging the clutch and engaging a second clutch or an overrunning clutch or a brake band. Culpin (1976) discusses the operation of this arrangement in more detail.

The 'Multi-Power' transmission was a countershaft constant mesh gear design with a pair of gears for high and another for low ratio. In low ratio the driven gear is connected to the layshaft by a spring loaded overrunning clutch. In high ratio the driving gear is connected to the input shaft with a multi-plate clutch. During operation in low range the clutch is disconnected but all gears rotate as a result of their constant mesh. As the clutch is engaged for high ratio, the transmission speed is increased causing the low ratio overrunning clutch to run free and power to be transferred by the high ratio path only. This design provided a number of positive as well as negative aspects (Farnworth, 2005). There is no engine braking in low ratio as there is no permanent mechanical connection between the engine and wheels due to the overrunning clutch. This same design does mean that when proceeding uphill in high ratio, when the clutch is depressed, the tractor will not roll backwards as the transmission locks up due to the conflicting transmission paths as the wheels attempt to rotate.

When these transmissions were introduced all were hydraulically operated via a lever operated spool valve. Clutch operation was either on or off, with no modulation or feathering. During fieldwork wheelslip damped the shift, but a downshift on the road could be particularly abrupt. The gear selection method has since been updated, for example through the introduction of a push-button solenoid-operated clutch on the Ford 'Dual Power' transmission (Emmadi and Tanzer, 1981). More recent transmissions of this type, such as those launched by Steyr and Fiat (1987 and 1988), have moved away from epicyclic design, instead featuring a layshaft with a pair of multi-plate clutches to select either the direct drive or the ratio reduction (Reiter and Renius, 1988).

Although the popularity of these transmissions peaked in the late 1980s following the introduction of semi-powershift transmissions, they are still available today albeit as an option for small and medium sized tractors (up to approximately 110kW). The New Holland TS-A tractor range is available with the addition of a button operated electro-hydraulic gear change in addition to the remaining mechanical synchromesh transmission. Some manufacturers have retained this transmission as the only one available for smaller tractor ranges, the Massey Ferguson 5400 series being a relatively new example (Neunaber and Wilmer, 2005a) using an electro-hydraulically selected pair of gear ratios in conjunction with four lever-selected gears. An increased use of electronics, including the ability to undertake non-powershift range changes via an additional button on side of the gear lever, make this transmission more user-friendly and a lower-cost alternative to the more complex transmissions available.

## 2.3 Semi-Powershift Transmissions (Multiple Ratio Change)

Once the operator had the ability to make a powershift change, the next development was to increase the number of powershift steps for each gear, leading to what is commonly known as a semi-powershift transmission. The most common current development is a transmission offering four powershift steps in conjunction with a number of manually selected gears.

David Brown was the first manufacturer to offer such a system, with the introduction of the Hydra-Shift transmission in 1971 (Bailey, 2002). The four powershift steps are achieved with two planetary gearsets each working on the principles previously described. The four speeds are obtained by directly engaging both, none or either one of the direct drive clutches. Actuation of the unit is via a hand operated lever operating a hydraulic control valve. The unit does have some semi-automatic functions in that when the tractor master clutch is depressed the unit resets to the first powershift gear without moving the lever. Then, on moving off, it very quickly shifts up again as the system hydraulic pressure is increased. For lightly loaded transport tasks this function is ideal, however the quick upshift under heavy load can result in the engine stalling.

Most semi-powershift transmission concepts have moved away from being based around a number of planetary elements, instead incorporating four clutches and four layshaft gears with the actuation of two of the clutches giving the four powershift steps. The advent of electronics in tractor control systems made this arrangement possible within the boundaries of comfort and performance expected by the operator.

One of the first transmissions of this type was the 16F/16R transmission launched on the Case Maxxum tractor range in 1989 (Renius, 1994). A lever is used to select one of four powershift gears, Hall-effect sensors detect the lever position and a microprocessor then energises one of four solenoid valves to actuate the appropriate clutch packs (Ross and Panoushek, 1990). The use of microprocessor control allows clutch fill rates to be adjusted for each gear, as well as permitting incorporation of a basic hydraulic fluid temperature compensation system. The four mechanical synchromesh ranges are operated with a second lever, causing problems particularly when changing into the top range of gears on the road (at 13-16km/h) when both levers need to be moved.

As discussed by Hall (1992), a potentially better solution was developed by Ford New Holland in 1991 for the 'Series 40' range of tractors. The main gear lever has two positions, as does an additional range lever, giving a total of four synchromesh ranges. The powershift gears are selected with push buttons on the side of the main gear lever and actuated via a microprocessor controller. This actuates two of the four powershift clutches to select one of the four powershift ratios available. When the clutch is depressed and the main gear lever moved forward, the transmission control software selects the correct combination of clutches for the next gear in sequence - particularly important for road work. Clutch fill times are minimised to reduce the possibility of torque interruption, but the microprocessor is programmed with individual shift profiles to optimise each 'up' and 'down' powershift. This transmission is still used on the current New Holland TS-A series tractors of up to 100kW rated power (Pearce, 2004a), although only one gear lever is now used. The mid synchromesh change is now hydraulically actuated at the request of the operator.

In addition to the four powershift step transmissions, other variants have been developed by manufacturers. Valtra 'T-series' (to 128kW rated engine power) tractors feature three powershift steps in addition to 12 mechanically-selected gear ratios (Wilmer, 2004). New Holland and Case feature six powershift steps with three electro-hydraulically operated ranges on the TM/MXM tractor ranges (Neunaber and Wilmer, 2004). Recently, some manufacturers have introduced semi-powershift transmissions with six or eight powershift gear ratios in conjunction with electro-hydraulically actuated synchromesh gears (Neunaber, 2005, McCarron, 2005).

Semi-powershift transmissions are very popular and these latest launches signal the intent of manufacturers to continue to promote them for larger tractors. The reason for their popularity is the perceived efficiency improvements over full powershift transmissions, as well as a lower purchase price. In terms of operational performance, the provision to make range changes electro-hydraulically is often sufficient in most situations. Microprocessor control has also improved the automated features available with these transmissions, even for non-powershift gear changes. Features currently available include automatic shifting at pre-determined engine speeds and some transmission controllers, such as the Renault 'Quadractiv', even account for the foot throttle position, in addition to engine speed, to adjust the pre-set upshift engine speed (Williams, 2002). With further refinement in control systems, including more integration between the engine, transmission and other vehicle control systems, the relatively high efficiency of these transmissions could see them continue to remain the most popular transmission option for UK and European agriculture for the next five (perhaps ten) years.

## 2.4 Full Powershift Transmissions

The first full powershift transmission was the Ford 'Select-o-Speed' transmission on the 6X series tractors, launched in the USA in 1959 and Europe in 1964. The ten forward and two reverse gears were achieved from a series of epicyclic gearsets, with gears selected by a single lever. The transmission suffered many mechanical problems and its lack of refinement during gear shifts resulted in a limited popularity. Nonetheless, other major manufacturers pursued the development of their own full powershift transmissions, such as the eight speed full powershift transmission fitted to the John Deere 4020 tractor from 1964 (Day, 2002). Full powershift transmissions tended to be reserved for high horsepower tractors, particularly for North America but they became more popular in Europe during the 1980s (Renius, 1994).

The Case 'Magnum' tractor range, launched in 1988 and still available today, is a good example of a full powershift transmission. As described by Eike and Stoever (1999), the transmission comprises of a three-speed gear unit, a three-speed range unit and a two-speed hi-lo unit, giving 18 forward and 4 reverse speeds. Drive is made by engaging one clutch from each section, meaning there are five disengaged clutches, leading to potentially higher losses than with similar semi-powershift transmissions. The control of this transmission, originally mechanical, is now undertaken electronically. The complexity of the shifts also varies with each gear because either one, two or three pairs of clutches need to be engaged or disengaged at once.

An alternative variant providing the same forward speeds is the Funk powershift transmission, historically used on tractors such as the New Holland '70 series' transmission (Holtmann, 1998), and more recently on articulated John Deere tractors (following the purchase of Funk by John Deere). The transmission can provide up to 18 forward and 9 reverse gears through a gear-shifting unit and a range-shifting unit. The gear-shifting unit has four shafts with 12 gears, six of which are splined to the shaft. Two of the remaining six gears are engaged at any one time by multiplate clutches to provide nine speeds. The range-shifting unit has five shafts with 10 gears, seven of which are splined to their shafts. One of the remaining three gears is engaged by a multiplate clutch to provide a high, low or reverse range. This means there are a total of nine multiplate clutches, three of which are engaged at any time, this being one more than the Magnum system. The electronic control of the unit is simple with pulse-width modulated solenoid valves for each clutch being controlled by a microprocessor-based controller. The only information used to make the gear shift is the speeds of the transmission input and output shafts. This relatively simple system still allows provision for automatic shifts from 10$^{th}$ gear onwards and user-programmable gear matching between forward and reverse.

In addition to the potentially higher losses from a high number of disengaged clutches, the pumping elements required to maintain sufficient pressure to actuate the clutches also leads to increased power losses. The increased number of meshed gears, especially in the four/five shaft Funk transmission, can also increase frictional losses. Typically, the efficiency of a full range powershift transmission can be below 85% (Goering and Hansen, 2004). The complex design and number of components also means this transmission type can be expensive to manufacture. It therefore tends to be reserved for high powered tractors, typically greater than 150kW, although it can be an option on lower powered tractors.

## 2.5 Continuously Variable Transmissions

### 2.5.1 Introduction

Whilst increasing the number of gear ratios in a transmission assists in the task of maximising operational efficiency, the development of CVTs allows this concept to be fully exploited. CVTs allow the forward speed to be matched to the task to be performed and the conditions, independently of engine speed. The forward speed can also be changed without interrupting the power flow. These allow the operator to maximise operational efficiency, but the main drawback of this transmission system is the poorer mechanical efficiency.

### 2.5.2 Hydrostatic Transmissions

The first CVT was developed at the National Institute of Agricultural Engineering in 1954 (Hamblin, 1956). The transmission consisted of a variable displacement engine-mounted pump and fixed displacement wheel motors. The basic design was developed in conjunction with Lucas Industrial Equipment during the late 1950s and later a large number of prototypes were tested in agricultural and industrial applications (Eyles and Edghill, 1970). The first commercial hydrostatic tractor was built by Eicher in 1965, but high prices, efficiency problems and a maximum speed of 25km/h resulted in only 15 units being sold up to 1972 when it was withdrawn (Beunk, 2002c).

International Harvester produced probably the most successful hydrostatic tractor range, known as the 'Hydro' and launched in 1967. Models ranged from 45kW to 84kW. As discussed by Beunk (2002), they were popular for applications where draught performance was not of primary importance, such as P.T.O. operations. Compared to the Eicher models, the 'Hydro' was commercially quite successful. Renius (2005) estimates that around 10,000 of these tractors were sold before production ceased in 1985. The 'Hydro' suffered from efficiency problems with the hydrostatic transmission. These were improved through the use of two mechanically selected gear ranges, however the transmission was still considered to be less efficient than stepped transmissions of the era. Pure hydrostatic transmissions have tended not to feature for general agricultural tractors since 1985, mainly as a result of their poor efficiency and higher cost. For very small tractors they have gained popularity, as for telescopic handlers, as a viable alternative to torque converter (hydrokinetic) transmission.

### 2.5.3 Power Split CVTs

The poor efficiency of hydrostatic transmissions has led to the development of the power split CVT over the last 15 years. The basic concept (outlined in Figure 2-1) is that engine power is divided between two branches, one mechanical and the other hydrostatic. In the mechanical branch a stepped-ratio is applied and in the hydrostatic branch a variable ratio is applied. The two branches are then collected together again, often through the use of an epicyclic gearset.

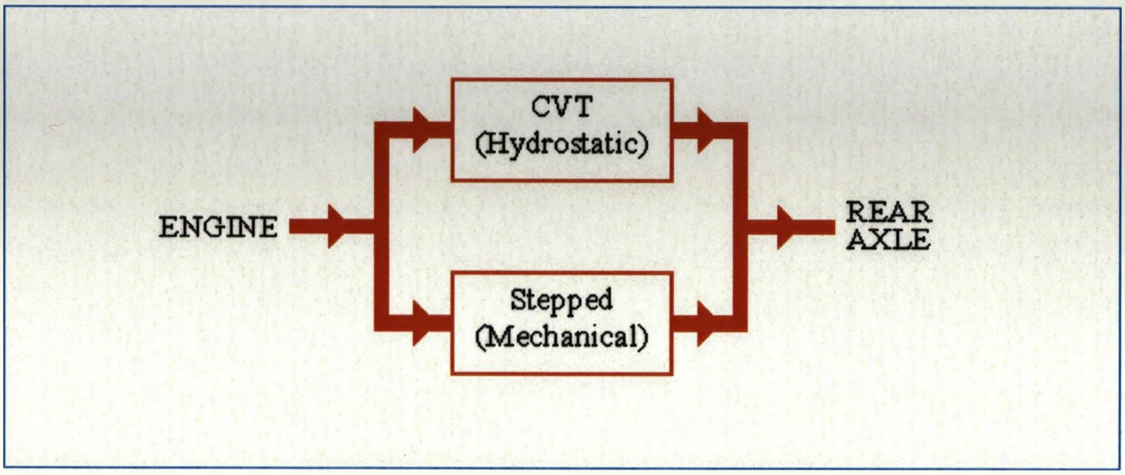



*Figure 2-1 – CVT power split concept*

This type of transmission was first presented by Fendt at the Agritechnica show in 1995 and launched as the 'Vario' transmission system in 1996 (Dziuba and Honzek, 1997). CVTs from Claas, Steyr, ZF and John Deere were all launched shortly afterwards.

There are two basic directions concerning the development of power split CVT transmissions:

1. small power capacity and limited ratio hydrostatic units with between four and eight mechanical gear ratios automatically selected; and
2. larger power capacity and wider ratio hydrostatic units with only two mechanical gear ratios, usually operator-selected.

Claas, Steyr, ZF and John Deere all pursued the first type, whereas Fendt have developed the second type. This part of the investigation discusses only the technical specification and performance of these transmissions. Section 8 gives consideration as to the market demand for these transmission systems.

#### *2.5.3.1 Fendt Vario*

Dziuba (1997) states the primary reason for pursuing such design, based primarily on hydrostatic units, was to avoid problems of additional complexity and electronic shift system when incorporating gears. This design required the development of specialist hydrostatic units with an axial variable displacement piston pump capable of swash plate angles from -30° to +45° and two axial piston motors capable of angles of 0° to +45°. These wide angles allow for a greater change in displacement and therefore ratio. The hydrostatic element of the transmission is complemented by two conventional user-selected ranges giving speeds up to 32km/h and 50km/h respectively. For light duties it is sufficient to use the high range only. To aid heavy transport an automatic shift (non-powershift) was introduced in 2002 (Renius, 2005). When moving off from rest, power transfer through the transmission is purely by hydrostatic means. To increase forward speed the pump and motor displacements are both increased, thereby increasing the speed ratio. Once the pump has reached maximum displacement the motors move back towards zero displacement again. The tractor speed and the proportion of power transmitted mechanically therefore continue to increase. At maximum forward speed the mechanical branch is transmitting 100% of the power.

Originally launched for large tractors (191kW) the 'Vario' system was quickly developed for four tractor ranges from 63kW to 221kW. Since 2004 this transmission design has also been used for the Massey Ferguson 'Dyna-VT' tractor models (Neunaber and Wilmer, 2005b). Fendt is a unique tractor manufacturer in that no alternative mechanical transmissions are offered. This has resulted in approximately 40,000 'Vario' units being produced by the end of 2004 (Renius, 2005) despite numerous criticisms about the complicated nature of the electronic transmission control system and the sheer number of buttons and switches in the cab (Neunaber, 1998, Pearce, 2001, Pearce, 2004b). Basic forward speed is controlled by either the joystick or travel pedal. For experienced operators the Fendt Tractor Management System (TMS) allows integration of engine and transmission control, providing the operator can determine how to operate the complex system. The system features the ability to adjust the relationship between engine and transmission speed changes, as well as cruise control and programmable reverse speed reduction from forward speed setpoints.

#### *2.5.3.2 Claas HM-8 and HM-II*

Both Claas powersplit CVT transmission concepts have been developed for the 'Xerion' systems tractor, although neither have yet made it into volume production. The current 'Xerion' tractor now uses a ZF CVT transmission. The HM-8 (1996) has one direct gear and seven power split ranges, keeping the proportion of power transmitted hydrostatically low and overall efficiency high. Peak efficiencies of 92% have been measured upon the HM-8 (Fredriksen, 1994). The HM-II (1999) uses five power split ranges and one direct gear. Simple dog clutches are used to change between the mechanical ranges, as shifts occur at synchronous speeds, but the system therefore requires a complex electronic control system to initiate the ratio shifts.

#### *2.5.3.3 Steyr S-Matic*

The Steyr CVT was launched in 2000, shortly before the company was purchased by ZF. The transmission was reviewed in detail by Aitzetmuller (2000). Its design consists of four mechanical gear ranges achieved via two epicyclic units, with engagement selected by dog-clutches at synchronous speeds (similar to the Claas method). This ensures that proportion of hydrostatic power transfer is no more than 50%, this being achieved by a variable swashplate pump and motor. An epicyclic unit is used to couple the mechanical and hydrostatic branches of the transmission. The transmission is used by Steyr, Case and New Holland in their CVT tractors and McCormick have recently announced their intention to use the S-Matic in the future (Renius, 2005). The simplicity of transmission operation, especially when compared to the Fendt transmission, has been generally well received (Neunaber, 2000a, Pearce, 2001). Forward speed control is achieved by a travel pedal (in place of the foot throttle). Three cruise control memories allow different working speeds to be easily set. When one of these is actuated the pedal travel is scaled to achieve the preset forward speed at maximum depression. 'Eco' mode provides an automatic means of controlling both transmission ratio and engine speed, given a specified forward speed demand, allowing the engine speed to be restricted and transmission ratio maximised.

#### *2.5.3.4 ZF Eccom*

A similar design to Steyr is used whereby four mechanical gear ranges help to keep the proportion of power transfer through the hydrostatic branch below 40%, utilising a variable swashplate pump and motor for the hydrostatic element. Four ranges are achieved through epicyclic gearsets, engaged using friction clutches. Presented in 1997, this transmission design was first used by Deutz-Fahr in its 'Agrotron TTV' tractors from 2000 (Neunaber and Wilmer, 2003), with John Deere using it since 2001 (Neunaber, 2000b) and Claas proposing to use it for their new 'Xerion' tractor.

The Deutz-Fahr system features three operating modes: automatic, P.T.O. and manual. In automatic mode vehicle speed is controlled on a joystick, pushing forward to increase speed and back to decrease speed. A separate dial controls acceleration rate and a further dial adjusts the acceptable engine speed range. P.T.O. mode is activated as soon as the P.T.O is engaged and this allows the operator to set the maximum engine speed reduction below a target speed. In the manual mode, the auto functions are not used and the foot pedal reverts to a conventional throttle only, the joystick then sets the travel speed.

#### 2.5.3.5 *John Deere AutoPowr*

The John Deere AutoPowr transmission utilises a variable wide-angle (45°) axial-piston pump and a fixed displacement motor. The mechanical transmission branch is via a planetary gearset, whereby different elements are locked by multiplate clutches to give two ratios. An additional planetary unit is used to obtain reverse drive. Launched in 2001 upon the John Deere 7010 series tractors (Beunk and Wilmer, 2003), this system is similar to the Fendt 'Vario', so similar efficiencies can be expected.

Driving control for this transmission is undertaken by a manual lever only. The lever position selects one of the two ranges and a thumbwheel on the lever is used to set target speed. The system is very intuitive and easy to use and the lever allows two cruise speeds to be set.

## 2.6 Potential Future Tractor Transmissions

### 2.6.1 Belt and Chain Drives

The use of a belt or chain, to transmit power from one variable diameter pulley to another, is the major variable transmission type presently used by most automobile manufacturers. Modern belts and chains are made of steel and work on a push principle, whereby the force is transmitted on the compression side of the belt. As torque is transmitted by friction there is a need for clamping forces between the pulley and the belt at high torque levels, thus reducing the efficiency. The 60kW Munich research tractor developed in 1988 (Renius, 1992) was fitted with a push chain drive in conjunction with a stepped mechanical range unit providing two forward and one reverse gears. The variator unit had an efficiency of 90%, but never reached production, partly due to the downfall of the commercial partner. Whilst these transmissions are likely to remain important for small cars, their limited torque capacity and their inability to obtain a sufficient range of gear ratios means they are unlikely to be a serious contender for future tractor transmissions.

### 2.6.2 Toroidal Traction Drives

Toroidal traction drives are currently being developed for automotive applications, although the concept has been in existence for many years but has never been converted into a commercial success. The Torotrak transmission (Field and Burke, 2005) is based around an epicyclic gear train to split the power into two branches, one a mechanical direct branch and the other including a variator. The variator is a friction drive unit, inside which are two pairs of disks which are internally hollowed into a doughnut shape (toroid). Hard rolling bodies transmit power through an elasto-hydrodynamic fluid film (Tinker, 1993). Three of these rollers are located between each of the two pairs of disks. The rollers transmit drive from the outer engine drive discs to the central output disks via special traction fluid. Each roller is attached to a hydraulic piston which controls the angle of the roller to the toroid and therefore the effective ratio, although the system actually controls the torque transmitted and therefore the ratio (Torotrak, 2005a). The level of torque the unit is capable of transmitting is proportional to the roller diameter, and currently units are being developed for agricultural tractors.

A number of potential advantages to agriculture cited (Torotrak, 2005b) are:

- cheaper and more efficient than a hydrostatic CVT;
- geared neutral allowing hill-hold features;
- clutchless direction change;
- improved durability of other components;
- ability to maintain constant torque, e.g.. maximum power;
- high overdrive for transport reduces engine speed required;
- improved fuel consumption.

Clearly this transmission has a potential to be developed for agricultural use in the future, most likely as a competitor to hydrostatic power-split CVT.

### 2.6.3 Electric Drives

Diesel-electric tractors have been considered as an alternative transmission design. Beunk (1999) discusses the concept, which consists of an engine driven generator which in turn powers an electric motor. Although at present this design has not progressed past the working concept stage, the infinitely variable forward speeds possible mean this transmission is clearly a potential development in the future, providing that a number of efficiency and safety issues can be resolved.

## 2.7 Comparison of Powershift-type & power split CVT Transmission Performance

The major technical advantage a power split CVT has over stepped-ratio transmissions is the ability to provide any desired vehicle forward speed independently of engine speed. Vehicle forward speed can be perfectly matched to the conditions being experienced or the forward speed can be maximised (rather than selecting the closest ratio). During high power requirements, the engine can be set to provide the maximum power output at all times, with the transmission adjusting forward speed as required. Comparisons between a Fendt Vario 926 and a similarly sized John Deere powershift tractor undertaking ploughing duties showed that the CVT average speed was a minimum of 10% higher than the powershift tractor (Isensee *et al*, 2001).

On the basis of the poor efficiency of the early hydrostatic transmissions, most workers have considered that CVT efficiency will be worse than a powershift-type transmission. Comparative overall data for different transmission types are rare, and possibly of limited value given the large variation in efficiency between different gears, speeds and torques.

*Table 2.1 – Comparisons of Transmission Efficiency (Okamoto et al, 1988)*

| Transmission Design | Efficiency |
|---|---|
| Constant Mesh | 80% |
| Power Synchroshift | 79% |
| Hydraulic Direction Shuttle | 77% |
| Semi Powershift | 74% |
| Full Powershift | 72% |
| Hydrostatic Transmission | 62% |

Data from Okamoto *et al* (1988), gives an overall comparison of different transmission types, the simplest constant mesh transmission providing the highest overall efficiency (see Table 2.1). Care should be taken as this data is 17 years old and design improvements could have improved these figures. No indication is given as to whether these figures are an average across the full operating ratio range. Its age also means no power split CVTs are included.

Tinker (1993) presented a comparison between efficiency and ease of use for different transmission types, which has been simplified to only include the main transmission types discussed in this Section (see Figure 2-2).

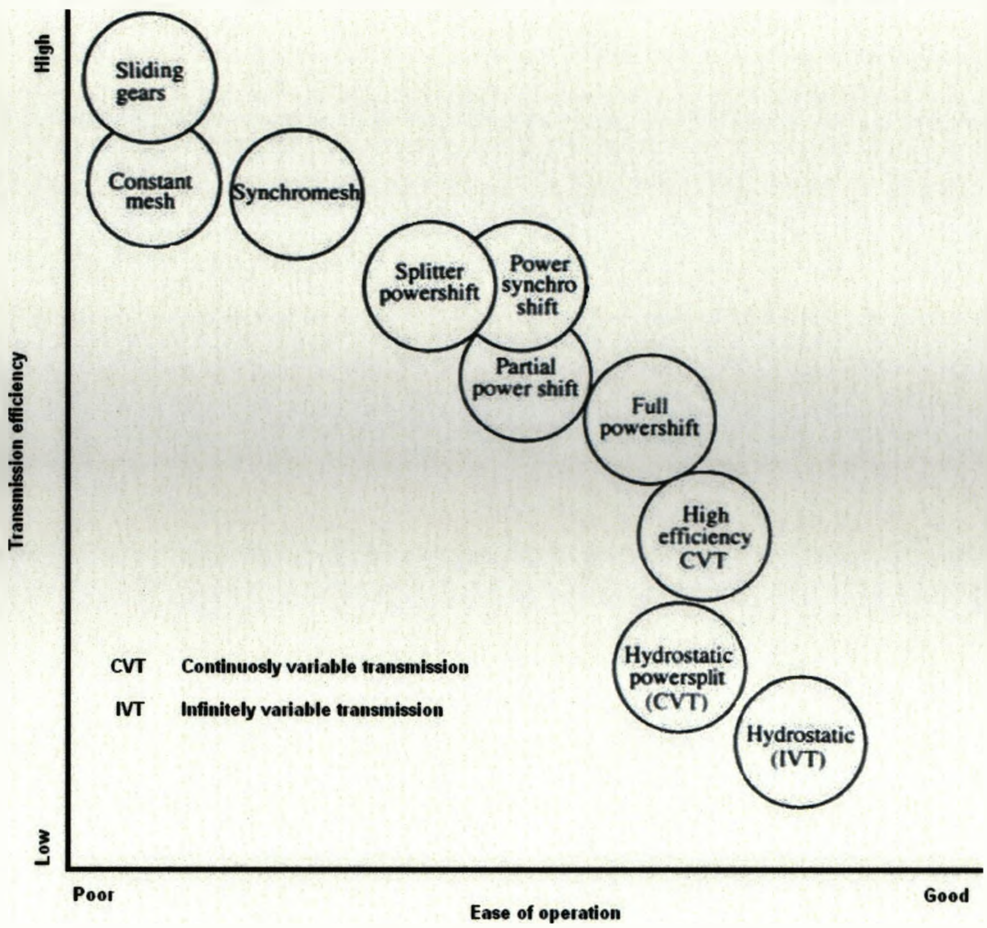



*Figure 2-2 - Schematic comparison of efficiency and ease of use for various transmission types (after Tinker, 1993)*

Pearce (2001) presented the first real comparative analysis between the different CVT transmissions available from Fendt, Steyr (fitted to a Case tractor) and John Deere, against a semi-powershift Steyr 9145 tractor. Tests were undertaken to determine the drawbar efficiency as a percentage of the measured engine power on a concrete test track, with efficiencies measured at engine speeds relating to maximum power for a range of forward speeds. The results of the tests are presented in Figure 2-3. It can be seen that none of the CVT transmissions reached the levels of efficiency achieved by the semi-powershift transmission, although at certain operating points there was little difference. However, typical forward speeds, as stated by Witney (1988), for mouldboard ploughing are between 5km/h and 10km/h and between 3km/h and 4km/h for power harrowing. At these speeds it can be seen the CVTs are considerably less efficient that the semi-powershift, especially the Case and John Deere tractors.

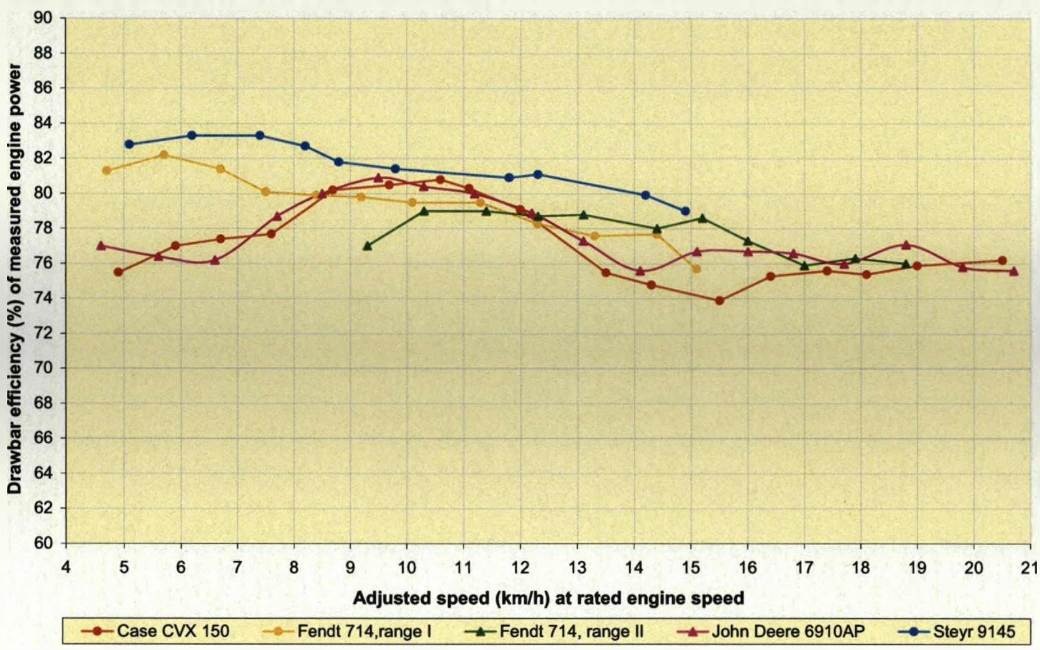



*Figure 2-3 - How transmission efficiency of CVTs compare to a semi-powershift transmission (Steyr 9145) across a range of forward speeds (courtesy: Farmers Weekly)*

The efficiency chart also highlights the effects of varying the proportion of power transfer through the mechanical and hydrostatic branches in the Case and John Deere transmissions. It is possible to identify the peak transmission efficiencies at each point where power transfer through the mechanical branch is maximised. The Fendt design, utilising an extremely efficient hydrostatic transmission branch, produces two relatively flat curves across the speed range. The two curves reflect the user-selectable ranges on this tractor.

During the trials the tractors were also driven with an 18 tonne payload trailer on a 42km road route. Fuel consumption was improved by a minimum of 10% from the Steyr semi-powershift tractor figures by the CVT tractors, the John Deere and Fendt bettering the figure by 20% albeit with considerably better fuel efficient engines than the Steyr tractor (Fendt 246g/kW.h, Deere 258g/kW.h, Case 283g/kW.h, Steyr 293g/kW.h). Whilst some of these improved fuel efficiency figures originate from the engine, the ability to maintain maximum forward speed at lower engine speed helped the CVT equipped vehicles. However, driving style of the tractors, particularly the semi-powershift, could have influenced these results.

Acceleration performance between a power split CVT and a full powershift tractor was investigated by the University of Kiel (Germany) and reported by Beunk (1998). The CVT tractor had faster acceleration than the powershift tractor, mainly because no time was lost during gearshifts but also because the transmission allowed the engine to always be operating at the point of maximum power output. The 28 tonne CVT tractor-trailer combination reached 40km/h in 30 seconds, compared to 56 seconds for the full powershift equipped vehicle. The main point of contention with this work was that the powershift transmission was 22kW less powerful than the 191kW Fendt CVT, although the improvement gained in acceleration outweighed the additional 10% engine power.

There is still a lack of truly comparative data which shows the superior performance of power split CVT-equipped tractors. The majority of improvements presented are as a result of the improved interaction between the engine and the transmission, highlighting the general benefits for any powertrain system of combined engine and transmission control. Indeed, where a direct comparison between transmissions, excluding engine influences is made, the semi-powershift transmission proved superior, especially at typical fieldwork speeds.

Ultimately the final purchasing decision is not made solely on performance but also the additional cost of the tractor, potentially higher maintenance costs and the additional perceived complexity of operation. These all potentially count against the CVT transmission. These philosophies are investigated further in Section 8.

## 2.8 Summary

Since the first powershift gear change element in 1954, the complexity of the tractor driveline has steadily increased. Full and semi-powershift transmissions have been developed with an increasing number of gear ratios, providing a wide range of forward speeds for different agricultural applications. The advent of electronics has enhanced the functionality of transmissions which incorporate powershift or electro-hydraulic synchromesh elements, as well as providing some feedback to the controller on operating conditions such as oil temperature or engine speed, thereby allowing the gearshift process to adapt accordingly. The widespread use of electronics offers potential for future improvements in transmission control.

Power split CVT transmissions are now available from a number of manufacturers, the major benefit being the ability to set vehicle forward speed independently of engine speed. The data presented suggests improvements in workrates and reductions in fuel consumption could be possible, but due to the interaction of other factors the picture is not clear. Indeed, when transmissions are compared directly the power split concept still suffers from poorer efficiency. A truly direct comparison of two otherwise identical tractors is required, and with a choice of transmissions, including a power split CVT, now available from manufacturers such as John Deere or Massey Ferguson this should now be possible. Of the future tractor transmissions discussed, the toroidal traction drive is the most likely to find commercial applications in agriculture.

This review highlights the need for combined engine and transmission control to improve overall vehicle performance. If improvements to control systems, including some interaction between those of the engine and transmission, were implemented on tractors equipped with semi-powershift transmissions their performance could be improved, leading to their continued dominance of the European market.

# 3 Test Tractor Characteristics

## 3.1 General Overview

The study utilised a New Holland TSA135 prototype tractor, built in Basildon by CNH. As the largest model in the TSA range, the tractor makes widespread use of electronic control and monitoring systems (see Figure 3-1) particularly in relation to engine, transmission and 3pt. hitch systems, making it ideal for this investigation. In addition, the TSA range of vehicles encompasses the current average UK tractor size.

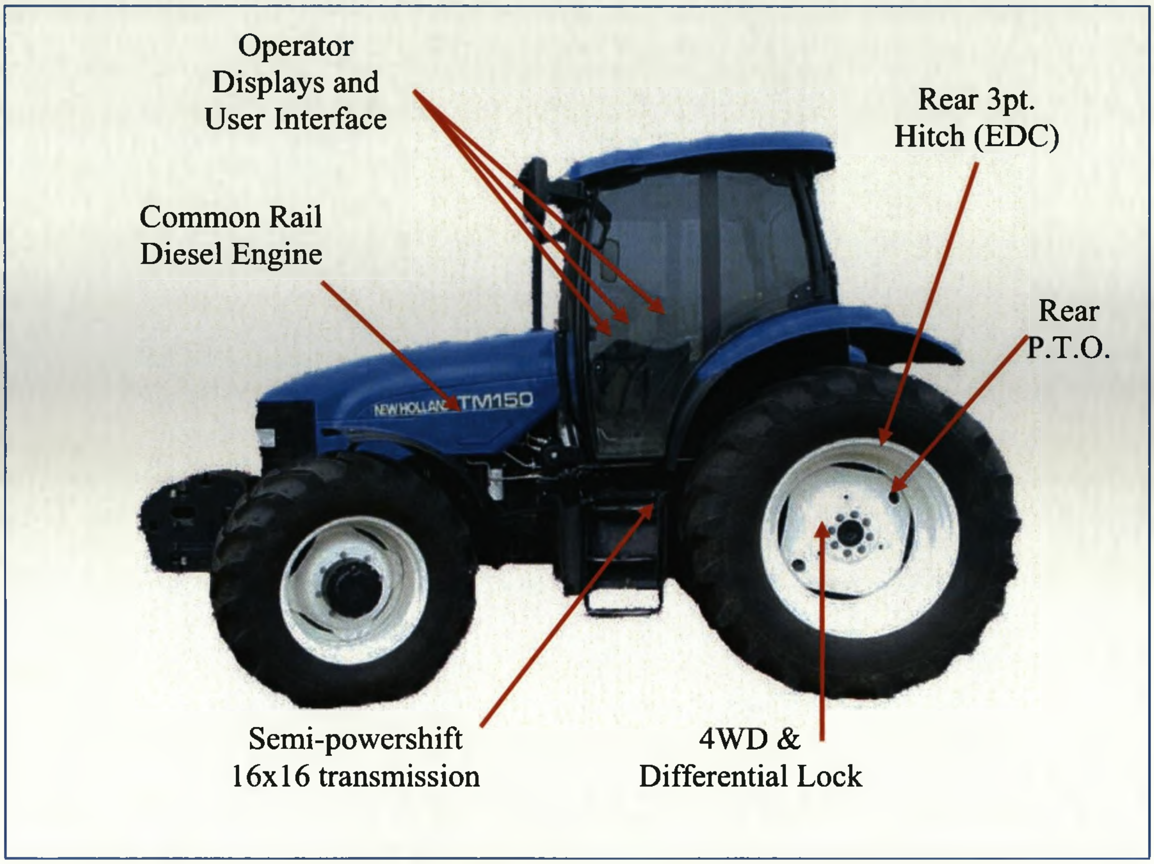



*Figure 3-1 – Test tractor with key electronically-controlled sub-systems*

The test tractor was fitted with a six cylinder (four valves per cylinder), turbocharged, diesel engine utilising an electronically-controlled (Bosch) high-pressure common rail injection system capable of generating 100kW (134hp) @ 2200rpm at the flywheel (to *ISO TR14396* - International Standards Organisation, 1996). Provision of electronic fuel injection control allowed engine power output to be increased ('boosted') to 119kW (160hp) under high levels of P.T.O. load (see Section 3.2).

The vehicle's 16x16 semi-powershift transmission comprised of four ranges of four powershift gear ratios, giving the ability to shift between ratios in a given range under load without an interruption of power. A flywheel-mounted damper assembly was located between the engine and transmission, there being no master clutch. Vehicle motion was halted by disengaging the hydraulic clutch packs used for engaging the powershift gear ratios. Vehicle direction was selected using a column mounted shuttle lever. The transmission output was connected to a conventional hydro-mechanically-lockable differential and a rear axle incorporating epicyclic reduction units.

The test tractor had five electronic control units (ECUs) for the engine, vehicle, instrument cluster, enhanced keypad and gear display, which were interconnected via a Controller Area Network (CAN-bus). The CAN-bus is a serial communication system used to transmit digital data across a twisted pair of wires from one ECU to another. This allows sensor information to be used by any ECU without additional wiring. The system allows wiring complexity to be reduced dramatically, more information to be made available to the operator, easy error detection and improved noise robustness as well as improved control integration between different ECUs.

The test tractor CAN-bus communication was based on the SAE J1939 standard. Each message was made up of 29 bits, including message source and destination bits, data bits and error bits. The 8 bit data field of each message allowed a number of parameters to be transmitted at once. Formats for many standard parameters (e.g. engine speed) are specified in the vehicle application layer part of the standard, J1939-71 (Society of Automotive Engineers, 2002).

During the investigation, the CAN-bus permitted the acquisition of information from tractor-based sensors by means of an existing diagnostic plug (see Section 5.2.1). Further test tractor specification details are provided in Appendix A1.1.

## *3.2 Engine Control Features*

The test tractor engine could generate more torque than either the traction driveline (transmission, differential, rear axle) or P.T.O. driveline could transmit individually, without potential long-term damage. As fuel injection was electronically-controlled, engine torque output was restricted to acceptable ('unboosted') levels in all operational conditions except during non-static P.T.O. tasks. In this situation, provided there was sufficient torque division between the traction and P.T.O. drivelines, engine power was 'boosted' allowing up to the maximum engine torque output to be generated. The degree of boost ($B_{\%}$) was dependent on the torque division between the two drivelines. The rate of change between 'boost' levels was restricted to prevent excessive torque-speed fluctuations. The test tractor engine output characteristics in 'unboosted' and 'boosted' operating modes, as measured at the P.T.O. shaft by means of an eddy current dynamometer, are depicted in Figures 3-2 and 3-3.

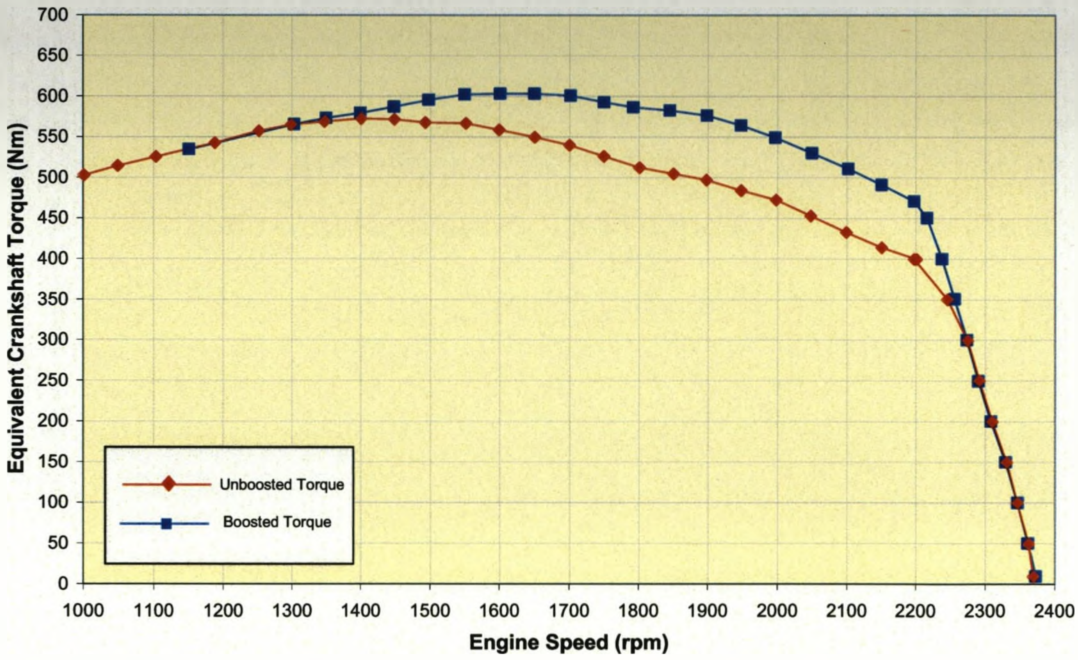



*Figure 3-2 – Engine torque-speed characteristics, as measured at the P.T.O.*

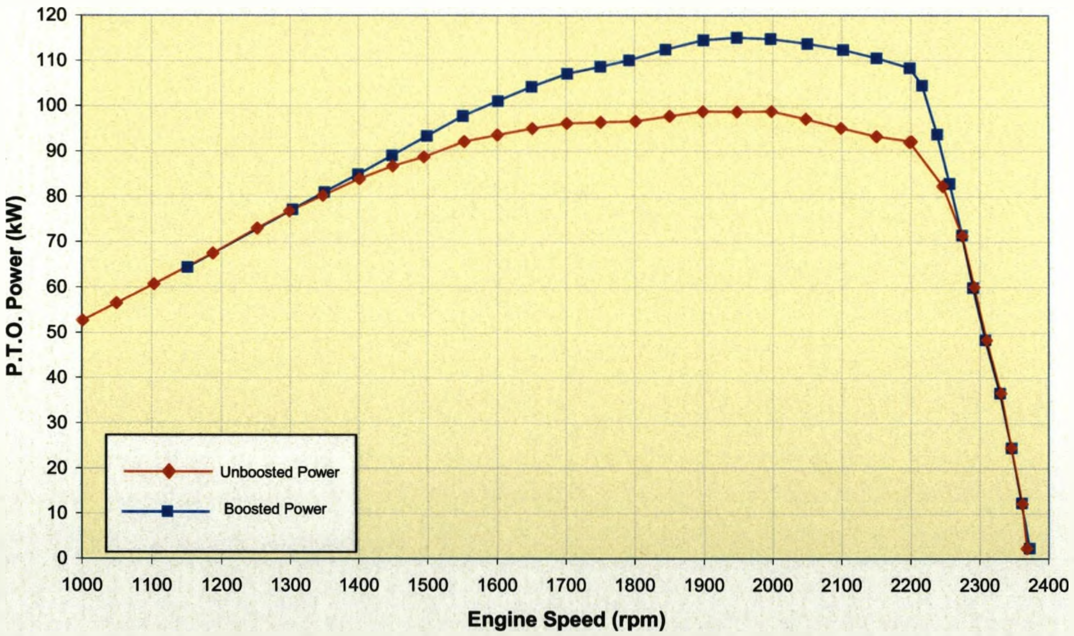



*Figure 3-3 – Engine power characteristics, in 'Boosted' and 'Unboosted' operating modes*

The power boost feature increased the power levels available in the working range of the engine (see Figure 3-3) by altering the torque-speed characteristics. In 'unboosted' mode, peak torque occured at 1400rpm, whereas in 'boosted' mode it was found at 1600rpm (see Figure 3-2).

A diesel engine is an inherently unstable system, because inlet airflow is unrestricted and fuel delivery is a function of engine speed. Without some form of closed loop engine speed control (through fuel delivery adjustments) there would be nothing to prevent engine speed from increasing to destruction or reducing until the engine stops. Speed 'governing' can be achieved through mechanical, hydraulic, pneumatic or electronic means. On-road vehicle diesel engines usually feature minimum-maximum governing, whereas agricultural diesel engines typically use 'all-speed' governors to control engine speed throughout the working range. An intentional (and often inherent) characteristic of a governor is a proportional response to a change in engine load, known as 'droop' ($\delta$), where engine speed reduces in response to increasing load. This is usually expressed as a percentage change from full-load engine speed ($\omega_{FL}$) to no-load engine speed ($\omega_{NL}$), as defined in Equation 3-1.

$$\delta = \frac{\omega_{NL} - \omega_{FL}}{\omega_{FL}} \times 100 \qquad \textit{Equation 3-1}$$

Whilst greater proportional response (droop) improves the vehicle driveability (Kimberley, 2004), in agricultural vehicle applications it is often desirable to have a smaller droop to reduce engine speed variation due to potentially high changes in applied load. Whilst mechanical governors frequently deliver a prescribed droop ($\approx$ 8%), electronic governing allows the implementation of different governor droops to suit the immediate application. The test vehicle featured three droop settings ($\delta_I$):

| | | | |
|---|---|---|---|
| 0 | = | 0% | constant engine speed mode (driver selects mode and set point) |
| 1 | = | 5% | general field operations ($\leq$ 12km/h) |
| 2 | = | 12% | road operations (>12km/h) |

These three alternative droop settings are shown diagrammatically in Figure 3-4:

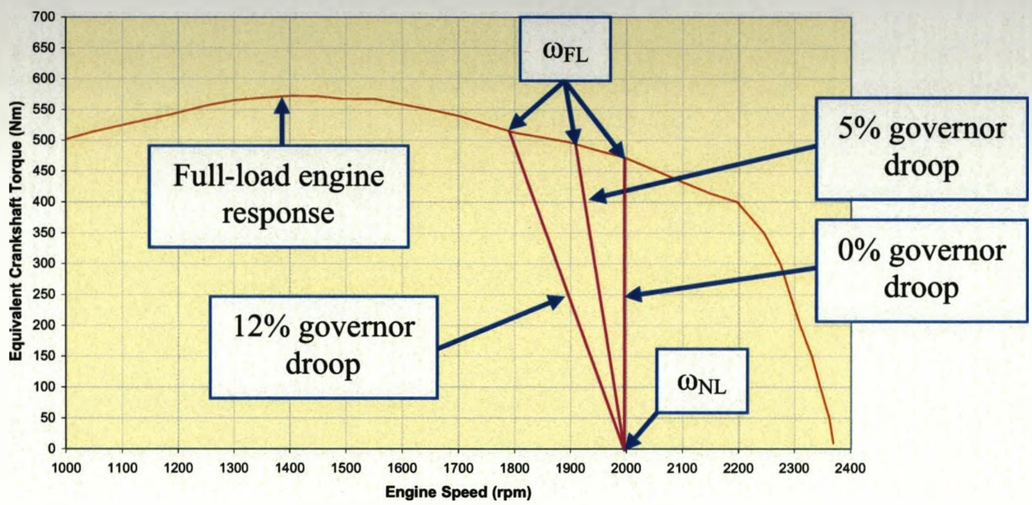



*Figure 3-4 – Engine speed characteristics with three alternative 'droop' settings*

## 3.3 Tractor Torque Measurement & Sensor Calibration

### 3.3.1 Flywheel Torque Sensor – Design

The test tractor was fitted with a flywheel torque transducer, a development of a standard flywheel damper assembly. Dampers are fitted to diesel engines to reduce the effect of periodic engine torque fluctuations, and thereby reduce the likelihood of driveline damage and gear noise (Nunney, 1998). The damper consists of a circular outer casing, bolted to the flywheel assembly, and an inner disc, fixed to the transmission input shaft. The disc is supported from the casing by circumferentially located compression springs which dampen the oscillatory movement between the two parts. The degree of spring compression is attributable directly to the torque produced by the engine. The New Holland design (Sedoni *et al*, 1996) uses this relationship as a means of flywheel torque measurement. Four equidistantly spaced metal tongues, attached to the inner disc, protrude through the centre (in a no-load condition) of slots cut into the periphery of the flywheel assembly (see Figure 3-5). When subjected to a torque through the damper unit to the flywheel assembly, the springs compress and therefore the position of the tongues changes in their slots. The springs in the damper disk assembly are a combination of pairs and individual springs of different stiffness and lengths resulting in a variation in the degree of compression and hence non-linear deflection of the tongues relative to the torque input.

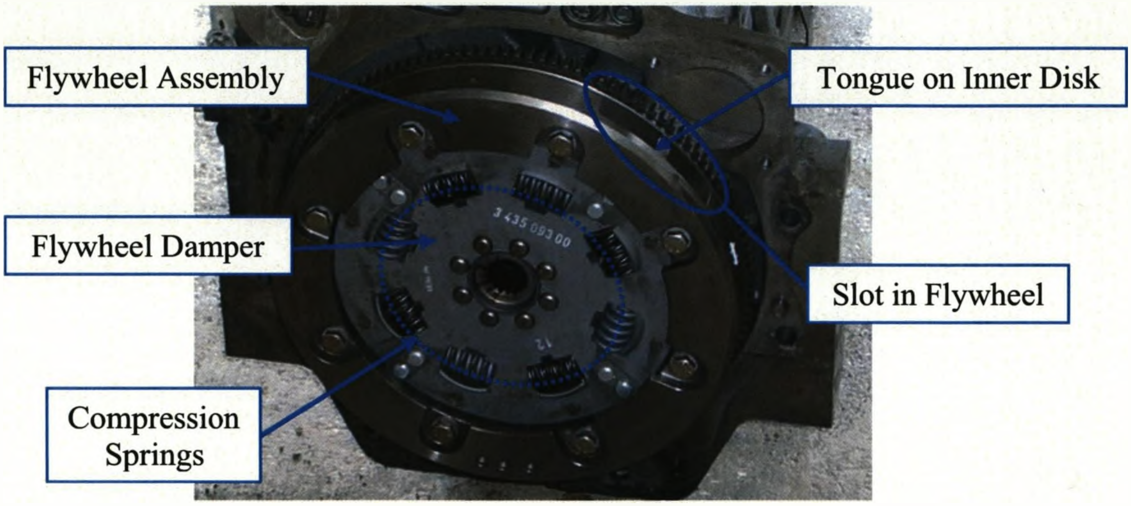



*Figure 3-5 – Flywheel torque sensor*

A metal-proximity sensor (Hall-effect) is used to sense the metallic components and the assembly face. By calculating the ratio of the metallic elements relative to the air gap, the degree of compression can be determined. Deviation from a zero-torque ratio would indicate either a positive torque (increase in ratio) or a negative torque (reduction in ratio). An internal calibration is then used to produce the output value transmitted within a message upon the CAN-bus, from where it was acquired and converted from % torque to Newton-metres (see Section 3.3.2).

### 3.3.2 Flywheel Torque Sensor – Initial Calibration

During engine torque-speed performance determination, the tractor-based CAN-bus flywheel torque signal was recorded at each point on the two full-load torque curves, together with P.T.O. torque from the eddy-current dynamometer. The two values were also recorded at a number of partial governor loads to determine sensor response across the engine torque-speed range (see Figure 3-6).

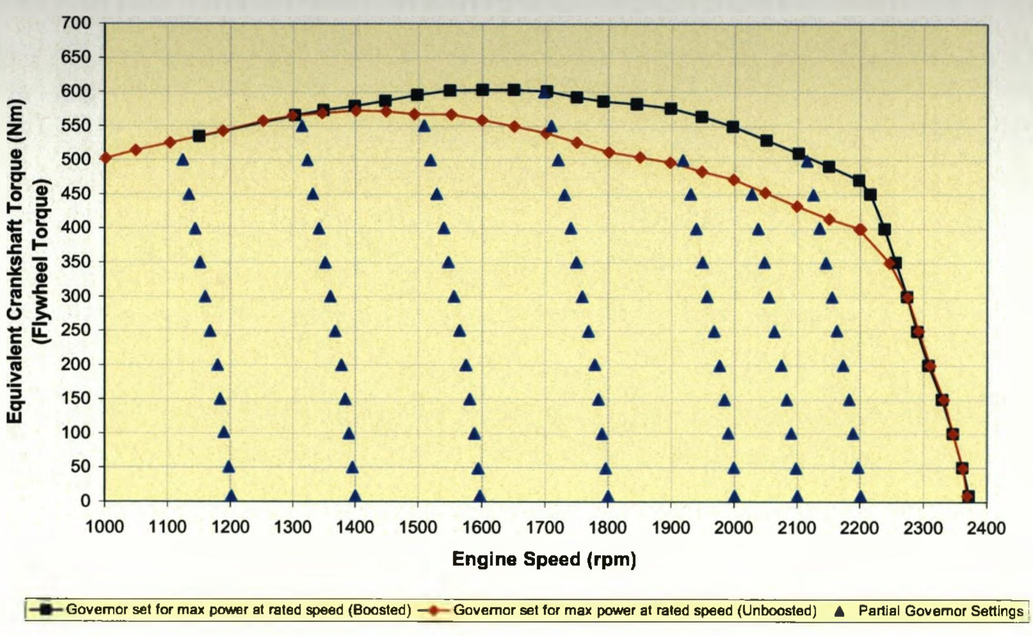



*Figure 3-6 – Range of values used for torque calibration*

Using a P.T.O. dynamometer allowed the sensor to be calibrated in-situ, but introduced a degree of uncertainty as a result of frictional and viscous losses along the P.T.O. driveline. The flywheel torque sensor operating principles and the Hall-effect transducer location would not allow calibration with the engine removed from the tractor. In addition the P.T.O. related losses were considered to be small in this driveline configuration. The flywheel torque sensor output ($T_{F\%}$) was plotted against the equivalent crankshaft torque derived for the P.T.O. measurements ($T_F$) and linear regression was used to determine the relationship (Equation 3-2), shown in Figure 3-7.

$$T_F = \frac{T_{F\%} - 28.47}{0.123} \qquad \textit{Equation 3-2}$$

As is standard practice, the known variable ($T_F$) from the dynamometer was used to predict the unknown variable ($T_{F\%}$); the equation was then rearranged to give the prediction equation. This allows the parameter likely to have the least amount of variability to be used as the basis for the model.

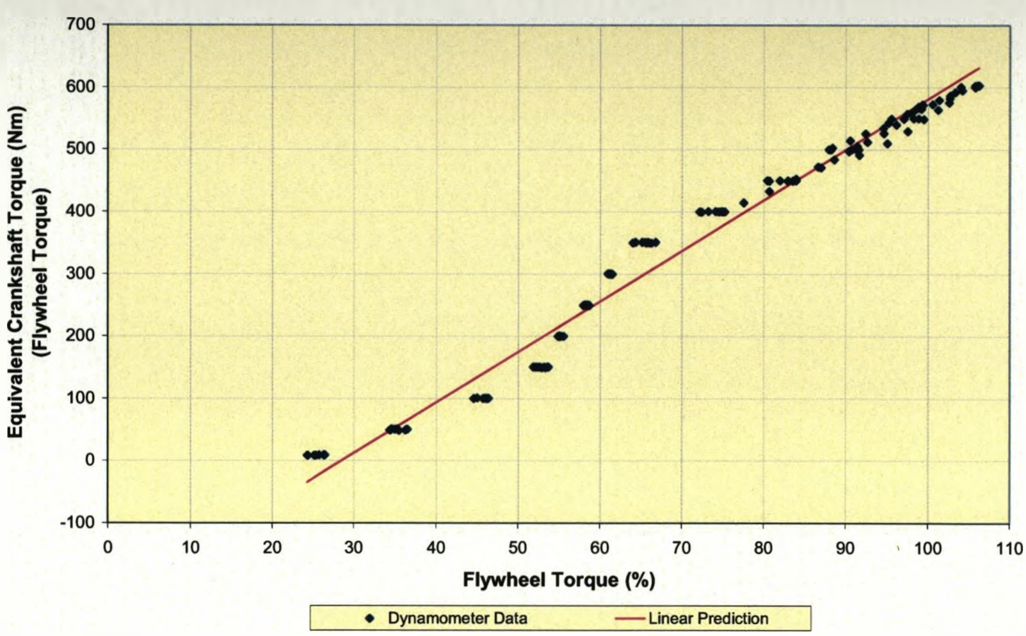



*Figure 3-7 – Initial linear flywheel torque calibration*

Whilst the regression equation established a significant relationship between the two variables, the root mean square (R.M.S.) error of prediction of equivalent crankshaft torque (flywheel torque) was 27.1Nm (see Appendix A3.2 for further regression statistics). It was also clear that the relationship had a non-linear element in the lower operating range of the sensor (see Figure 3-7) which required further investigation to improve the model. As discussed in Section 3.3.1, this non-linear characteristic was expected. The relationship seen is also influenced by the vehicle controller software which included an algorithm which attempted to correct for the non-linearity.

Figure 3-7 also shows small changes in $T_{F\%}$ for the same dynamometer load. This was considered to be potentially related to the different partial governor curves used. A regression model, including engine speed as well as flywheel torque, showed that speed was not statistically significant at the 95% level in influencing $T_{F\%}$ (see Appendix A3.2 for regression statistics), so was not included in any further analysis.

### 3.3.3 Flywheel Torque Sensor – Two Stage Calibration

Multiple regression techniques were used to determine an improved fit to the data, using the standard method of minimising R.M.S. error (Wallach and Goffinet, 1997). Two relationships were developed: an exponential relationship (Equation 3-3) for the lower torque values and a linear relationship (Equation 3-4) for the upper torque values (see Figure 3-8).

$$T_F = \frac{LN\left(\dfrac{64.947 - T_{F\%}}{42.568}\right)}{LN(0.992264)}$$ *Equation 3-3*

$$T_F = \frac{T_{F\%} - 11.921}{0.15572}$$ *Equation 3-4*

The two equations were solved simultaneously for $T_{F\%}$, the crossing point between them being 61.3%. R.M.S. error was reduced from 27.1Nm for the single model to 9.3Nm for the combined equations (see Appendix A3.2 for further regression statistics).

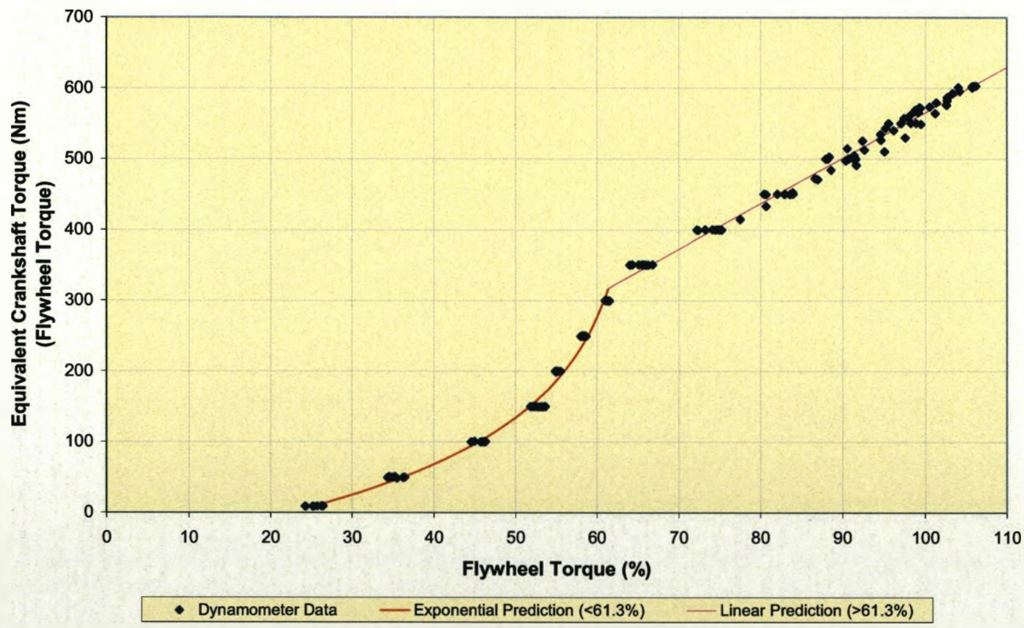



*Figure 3-8 – Two stage flywheel torque calibration*

### 3.3.4 Negative Flywheel Torque Measurement

In most situations the engine applies a positive torque to the vehicle driveline but there are situations when a net negative torque is applied to the flywheel, i.e. during "engine braking" when the vehicle is descending a hill with a laden trailer. It was assumed the flywheel damper's negative torque characteristics would be a two-dimensional mirror of the positive torque relationship (i.e. a 180° rotation around the origin point) due to the symmetrical properties of the damper either side of its zero torque position, but this required validation. With no means of applying negative torque through the tractor driveline in a controlled manner, a novel method was devised to determine the relationship.

Motoring data, that is when a test engine is driven by an external source, was provided by the engine manufacturer. The manufacturer was unable to supply flywheel torque data relating to the motoring data, as although the flywheel is attached to the engine crankshaft, the Hall-effect sensors are located in the transmission casing and were therefore not present when the data in question was acquired. The data consisted of the torque required to drive a hot engine at a range of speeds once fuel supply is stopped. Figure 3-9 shows the motoring data, from which a quadratic relationship between driven engine speed and motoring torque was obtained.

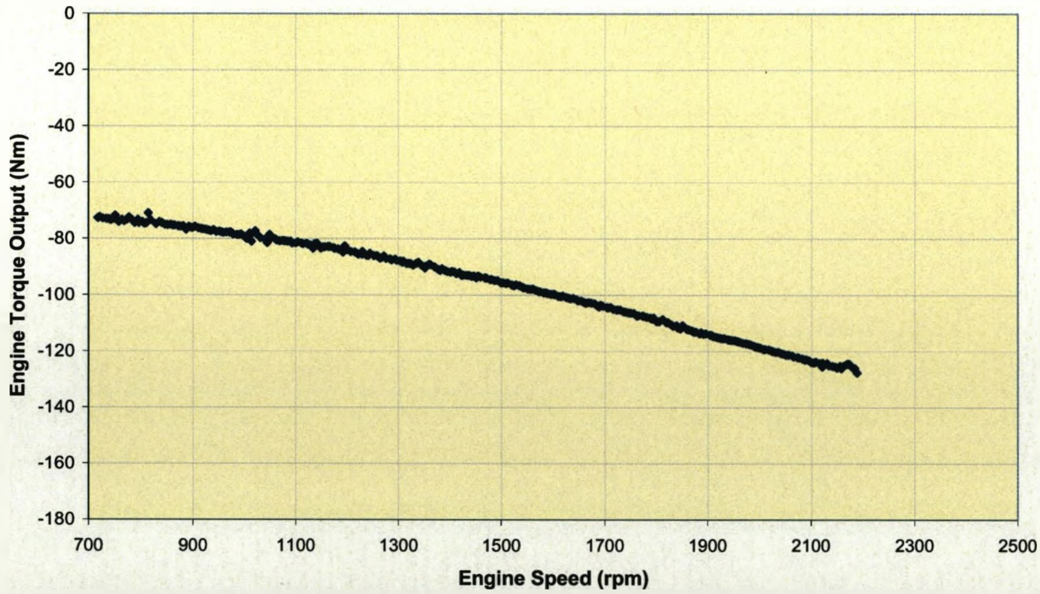



*Figure 3-9 – Engine motoring data*

A series of tests were performed, whereby the test tractor and a laden trailer were operated down an incline in each of the upper transmission gear ratios (number 13 - 16), allowing the tractor engine to be 'driven' by the combined mass of the tractor and trailer. Flywheel torque percent ($T_{F\%}$) together with the engine speed were recorded and a relationship derived.

These two equations were solved for engine speed, allowing a prediction to be made for negative flywheel torque (in Newton-metres) given a known $T_{F\%}$. This was not actually used for the calibration but merely to validate (or discount) the mirror relationship previously suggested. The vehicle motoring prediction is shown in Figure 3-10 along with a 180° rotation of the lower positive torque calibration curve. The point of rotation ($T_{F\%} = 22$) was determined by resolving Equation 3-3 to zero.

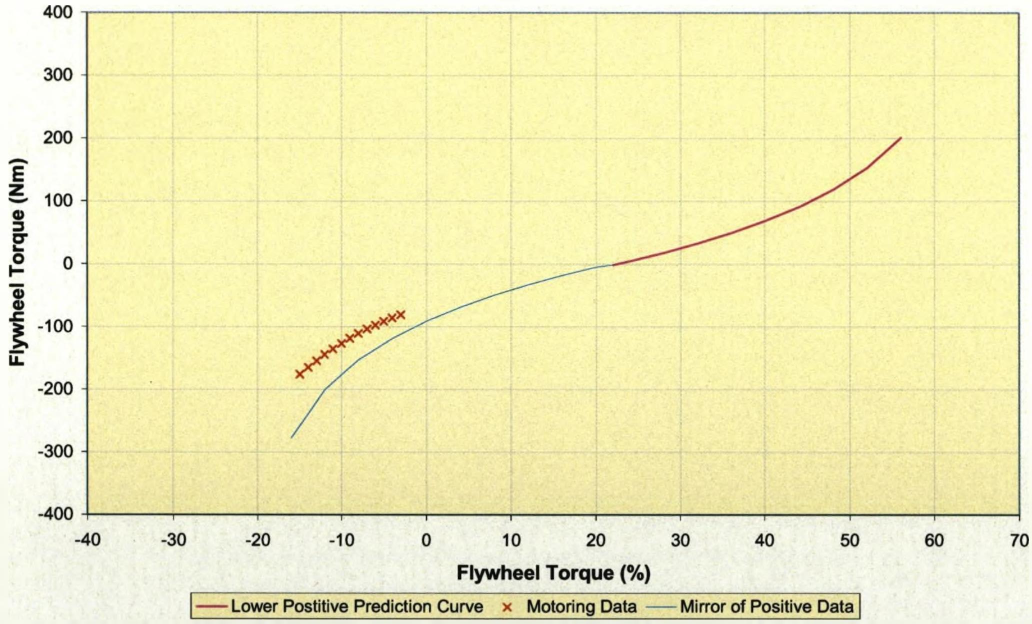



*Figure 3-10 – Motoring data with the proposed negative calibration curve*

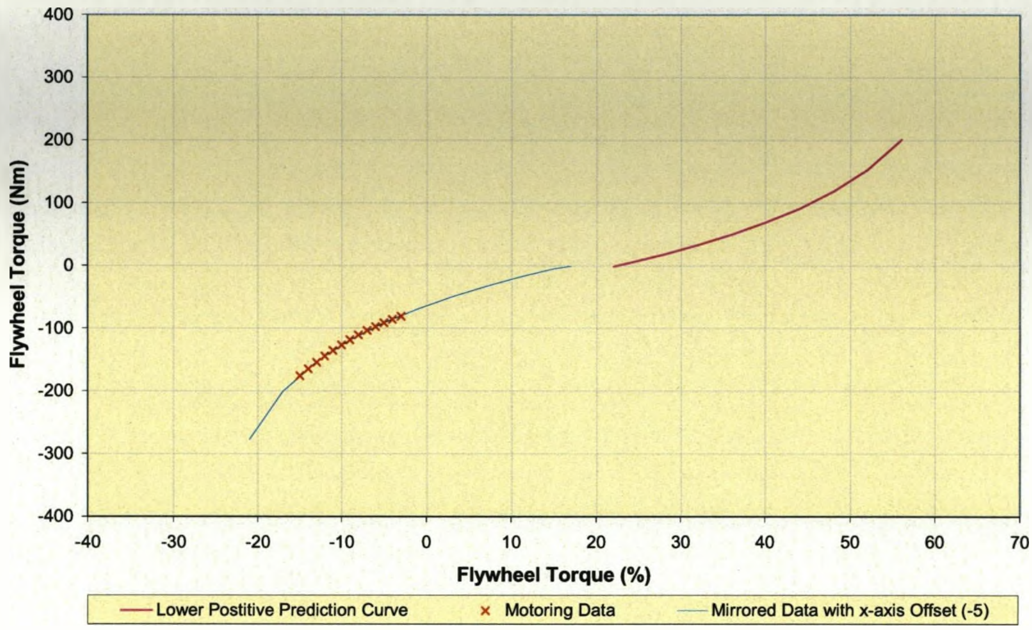



*Figure 3-11 – An offset (-5 in x-axis) negative torque curve*

Whilst the motoring data supported the proposed relationship, an offset was present: this offset must be horizontal rather than vertical to avoid a discontinuity around the zero torque position. A horizontal offset resulted in an extended zero torque position around the relaxed flywheel spring positions, allowing a small flywheel deflection for no change in torque. To allow for this, the negative curve was offset by -5% along the x-axis, equivalent to 2.5% sensor output either side of a zero torque position. This new curve passes through the line of the motoring data see (see Figure 3-11).

Multiple regression was used to determine an equation for negative torque, based on the offset curve for flywheel torques below 22%. A quadratic equation provided the closest response (Equation 3-5). The final calibration curve, based on this relationship, is shown in Figure 3-12 alongside the motoring data.

$$T_F = (5.6168 \times T_{F\%}) - (0.1277 \times {T_{F\%}}^2) - 62.543 \qquad \textit{Equation 3-5}$$

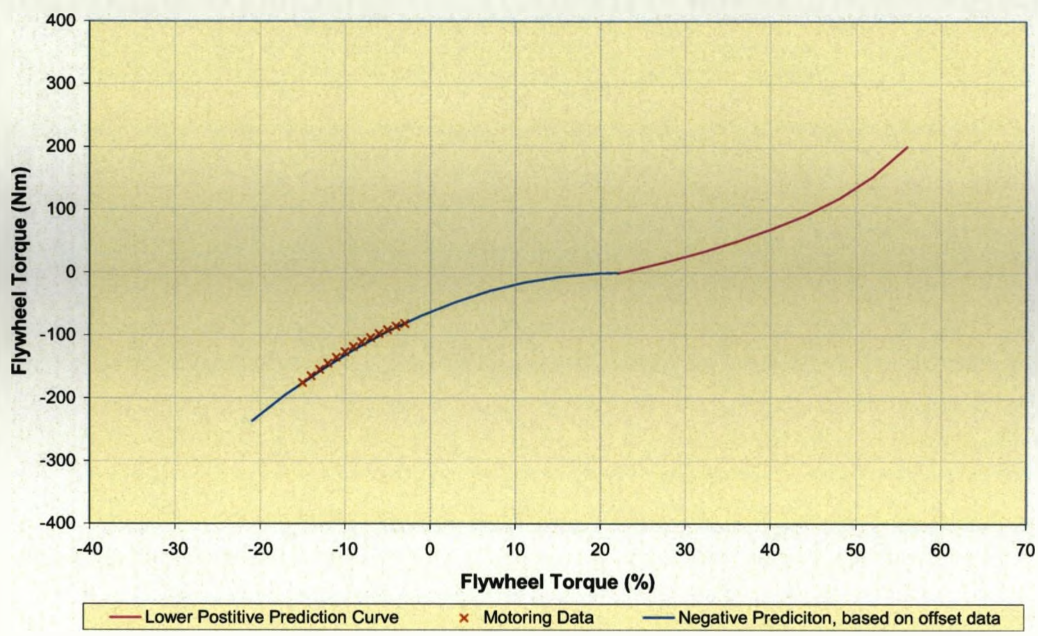



*Figure 3-12 – Final negative torque prediction equation and curve*

### 3.3.5 Flywheel Torque Sensor Calibration Summary

The three proposed relationships to give a flywheel torque (Nm) output from the acquired CAN-bus flywheel torque percent values were obtained, and then used for all subsequent parts of the investigation (see Figure 3-13).

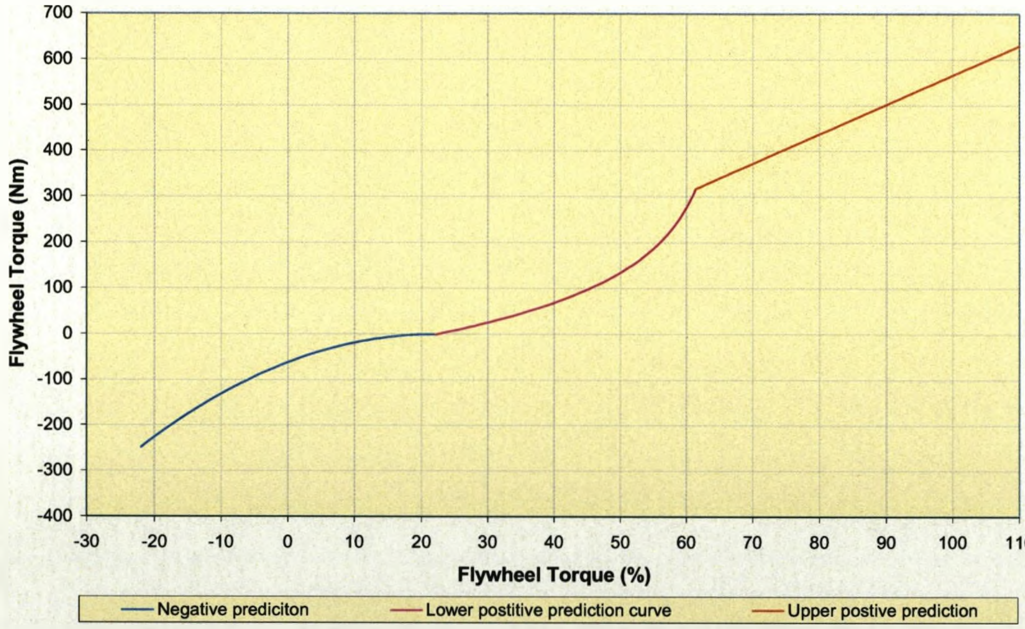



*Figure 3-13 – Three flywheel torque calibration models*

The final relationships and their conditions of use are:

1. $T_F = \dfrac{LN\left(\dfrac{64.947 - T_{F\%}}{42.568}\right)}{LN(0.992264)}$ **(for 22 < $T_{F\%}$ < 61.3);**

2. $T_F = \dfrac{T_{F\%} - 11.921}{0.15572}$ **($T_{F\%} \geq 61.3$);**

3. $T_F = (5.6168 \times T_{F\%}) - (0.1277 \times T_{F\%}^{\;2}) - 62.543$ **($T_{F\%} \leq 22$).**

### 3.3.6 P.T.O. Torque (Vehicle Sensor)

As the P.T.O. driveline is a single shaft from the flywheel to a speed reduction unit at the rear of the transmission, the degree of shaft twist (angular elastic deflection) is proportional to the transmitted P.T.O. torque. The position of the P.T.O. shaft at the flywheel is known from the Hall-effect sensor and associated software identifying the damper disc tongues. A second Hall-effect sensor determines the phase-lag of a tone wheel relative to the flywheel tongues (see Figure 3-14). The tone wheel is splined to the P.T.O. driveshaft prior to the reduction unit; therefore phase-lag between the two indicates shaft twist. The vehicle software determines P.T.O. torque from this information and outputs the resultant torque (in Newton-metres) relative to the flywheel as part of a CAN-bus message.

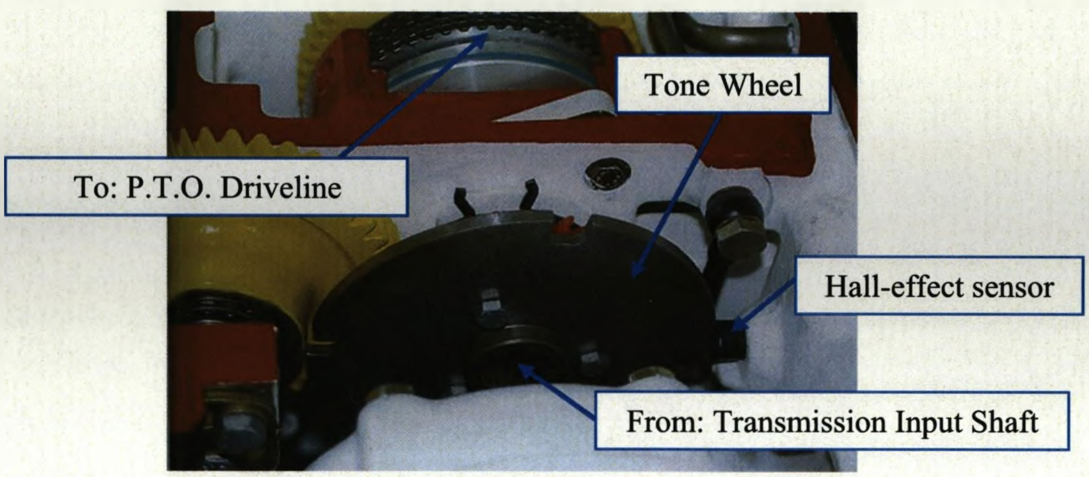



*Figure 3-14 – Tone wheel and Hall-effect sensor for determining P.T.O. Torque*

## 3.4 Transmission Design & Operation

### 3.4.1 Transmission Design

The test tractor transmission may be considered as three individual sections, as shown in the transmission cutaway model (see Figure 3-15).

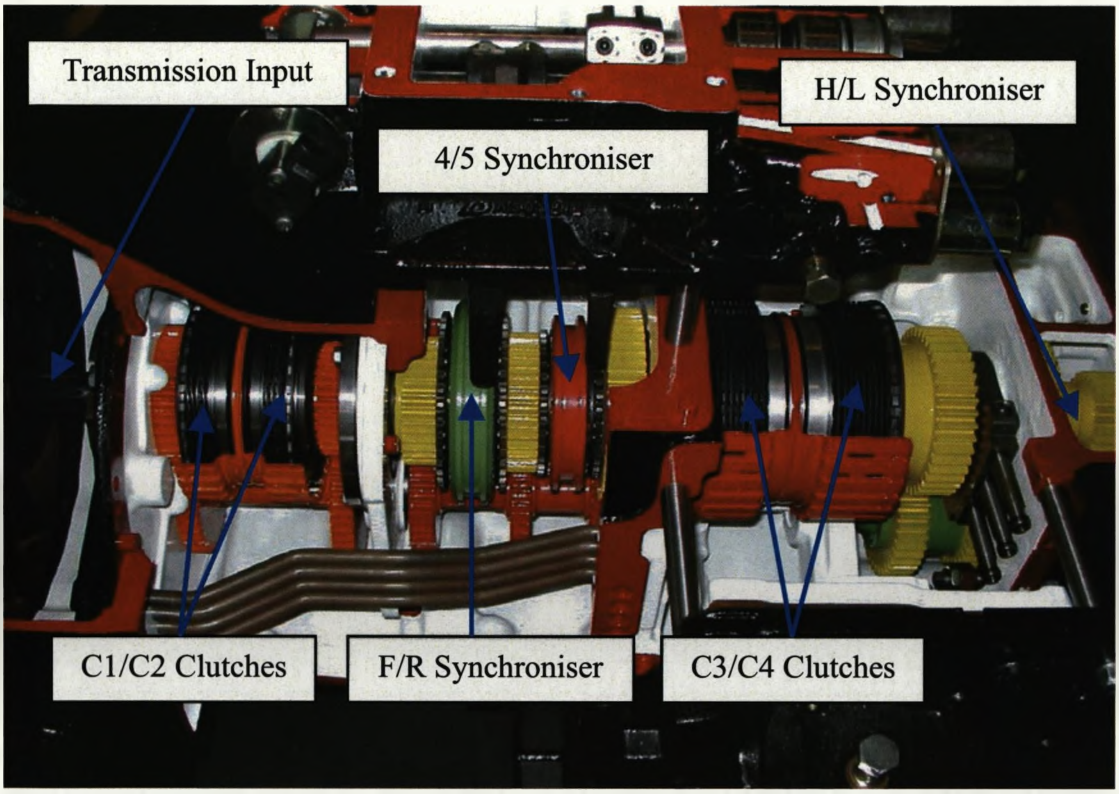



*Figure 3-15 – CNH TSA 16x16 semi-powershift transmission cutaway*

1. The front section contains the C1 and C2 multiplate clutches and their associated gears, these work in conjunction with C3 and C4 clutches. At any time, one of each pair is active to permit selection of one of the four powershift gear ratios.
2. The middle section contains the Forward/Reverse (F/R) gears and synchroniser together with the gears for the 4/5 range change and its synchronizer.
3. The rear section contains the C3 and C4 multiplate clutches and their associated gears, together with the High/Low (H/L) gears and synchronizer.

The four powershift gear ratios are doubled by the action of the 4/5 synchroniser, and then again by the H/L synchroniser, giving a total of 16 forward gear ratios. By moving the F/R synchroniser to reverse, all transmission gear ratios are also available in reverse, although they are not identical to the forward ratios.

The four powershift clutches (C1, C2, C3 and C4) are all pressure lubricated wet multi-plate constant running clutches, engaged by oil pressure (via Pulse Width Modulated (PWM) control valves) and spring released. C1 and C2 have four friction and four steel plates, C3 and C4 have nine of each and are of a larger diameter, reflecting the fact that they are further along the transmission path and therefore subject to higher torque loadings, in addition to being responsible for driveline engagement and vehicle inching. The F/R synchroniser and the 4/5 synchroniser are both electro-hydraulically operated by PWM solenoid valves.

The combinations of clutches, and range selections required to give each of the 16 forward speeds are shown, together with the transmission ratio and the total engine to rear wheel hub ratio, in Table 3.1. The difference between these ratios is a result of the differential (5.22:1) and the rear axle epicyclic (6.75:1) reduction ratios.

*Table 3.1 – Gear selection and ratios - 16x16 transmission*

| Gear | Powershift Clutches | | | | 4/5 Synchro | | Range | | Transmission | Total Driveline Ratio |
|---|---|---|---|---|---|---|---|---|---|---|
| (G) | C1 | C2 | C3 | C4 | 1-4 | 5-8 | L | H | Ratio ($r_t$) | ($r_d$) |
| 1 | | X | | X | X | | X | | 8.51 | 300.12 |
| 2 | | X | X | | X | | X | | 6.94 | 244.66 |
| 3 | X | | | X | X | | X | | 5.70 | 200.93 |
| 4 | X | | X | | X | | X | | 4.65 | 163.80 |
| 5 | | X | | X | | X | X | | 3.62 | 127.76 |
| 6 | | X | X | | | X | X | | 2.95 | 104.15 |
| 7 | X | | | X | | X | X | | 2.43 | 85.53 |
| 8 | X | | X | | | X | X | | 1.98 | 69.73 |
| 9 | | X | | X | X | | | X | 2.18 | 76.75 |
| 10 | | X | X | | X | | | X | 1.77 | 62.57 |
| 11 | X | | | X | X | | | X | 1.46 | 51.38 |
| 12 | X | | X | | X | | | X | 1.19 | 41.89 |
| 13 | | X | | X | | X | | X | 0.93 | 32.67 |
| 14 | | X | X | | | X | | X | 0.76 | 26.63 |
| 15 | X | | | X | | X | | X | 0.62 | 21.87 |
| 16 | X | | X | | | X | | X | 0.51 | 17.83 |

n.b. 'X' denotes engagement

Table 3.1 demonstrates the overdrive nature of the top four transmission gear ratios (13-16), engine speed almost being doubled through the transmission in the case of gear 16, admittedly to then be reduced by the differential and rear axle. This transmission design reduces the size of the transmission components as, for a given power level, lower torques need be transmitted but speed-related efficiency losses tend to be greater.

The power flows for three example transmission gear ratios are shown in Figures 3-16, 3-17 and 3-18. The transmission design causes all components to rotate at all times regardless of the gear ratio engaged. The power flow paths are shown as a green line, with components responsible for power transfer shaded red. The selection of powershift and synchromesh gears are shown with blue boxes or shading.

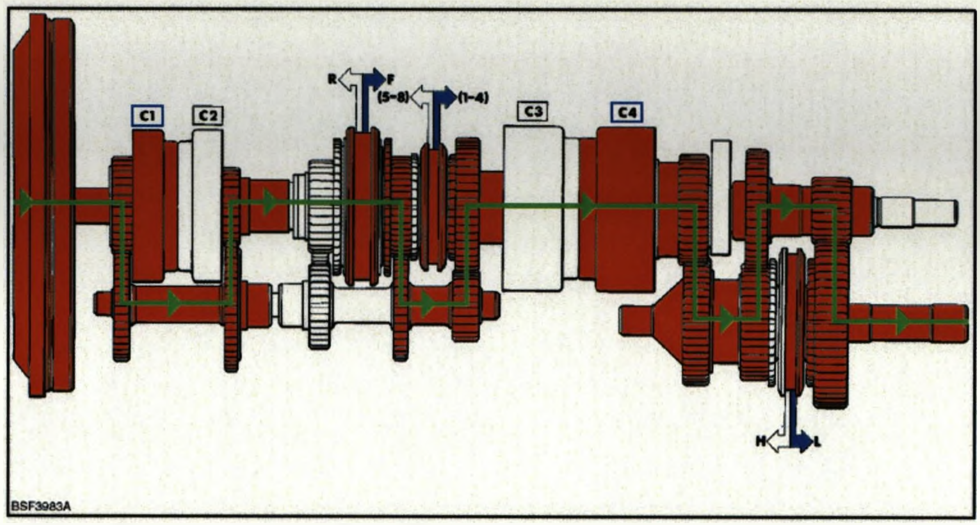



*Figure 3-16 – Transmission path (green line) and powered components (shaded red) in gear 3*

The complexity of the paths through the transmission varies dramatically between gear ratios. Figure 3-16 depicts gear 3, one of the most complex routes. In contrast gear 6 (see Figure 3-17) is simple, virtually passing straight through the transmission with just a single range reduction. Gears 13 and 14 (high range ratios) are similarly simple, each passing through one of the other gear pairs in the rear of the transmission, as demonstrated for gear 16 (see Figure 3-18).

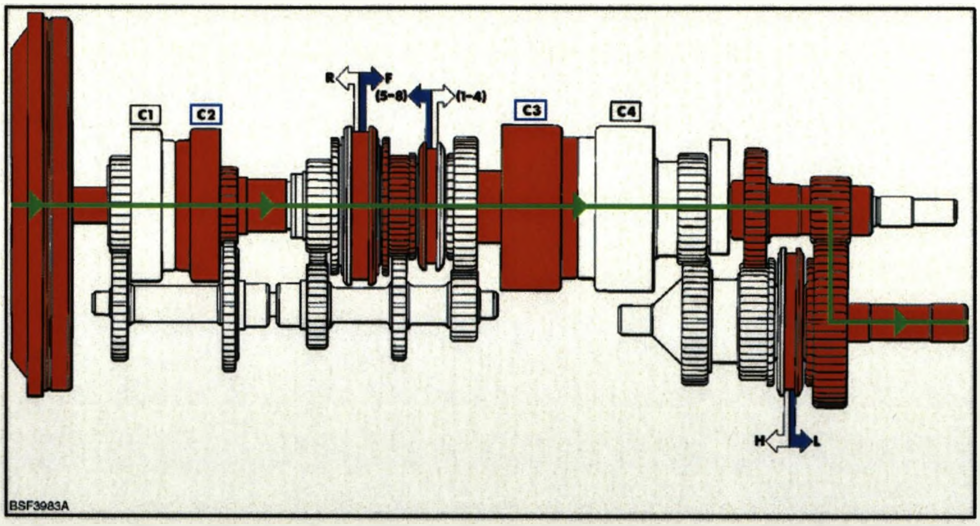



*Figure 3-17 – Transmission path (green line) and powered components (shaded red) in gear 6*

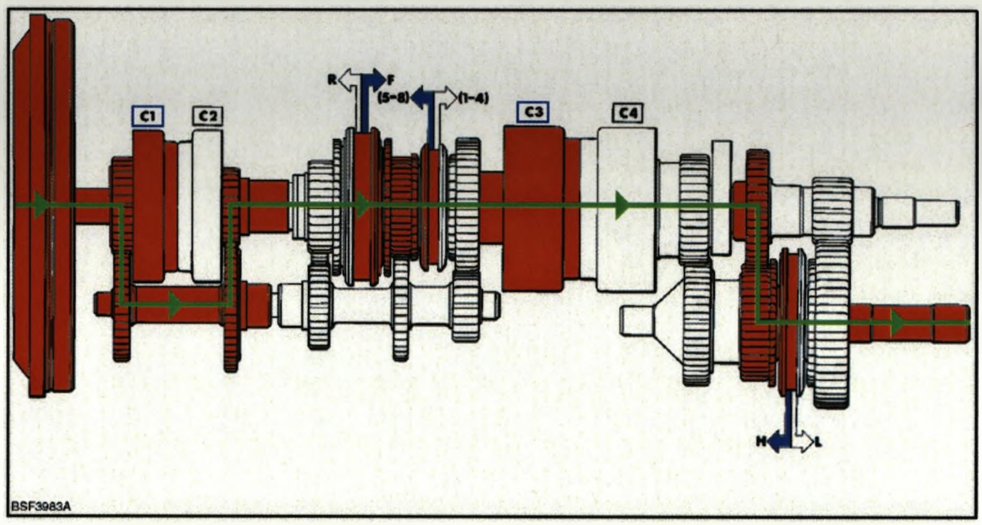



*Figure 3-18 – Transmission path (green line) and powered components (shaded red) in gear 16*

### 3.4.2 Transmission Operation

Gearshift control is undertaken by the vehicle electronic control module, which utilises information from a number of sensors to assist control of the PWM valves as well as providing status information to other control modules. The four powershift gear ratios are selected by the operator by means of push buttons on the side of the gear lever. PWM valve operation, and therefore powershift clutch engagement, is then controlled by the module with respect to time with compensation being made for transmission oil temperature. To make a 4/5 change, a second button on the gear lever must be held whilst pressing either the up or down gear shift buttons. Again, once instigated by the operator, the electronic module controls the actual gearshift. The range-change between the upper eight and lower eight gears is the only change which requires the clutch pedal to be depressed and the gear lever moved manually, the H/L synchroniser then being moved by a Bowden cable. However, when this change is made, the control module also changes the engaged clutches and the position of the 4/5 synchroniser, ready for the clutch pedal to be released in the new gear. In addition to the oil temperature, the transmission control module utilises information regarding a number of transmission parameters:

- clutch pedal position (potentiometer);
- clutch disconnect status (switch);
- shuttle lever position (switches);
- F/R synchroniser position (potentiometer);
- 4/5 synchroniser position (potentiometer);
- gear lever position (switches);
- gear selector switches (momentary, upshift or downshift);
- parking brake status (switch);
- operator presence on the seat (switch).

The test tractor transmission also featured a user-selectable 'autoshift' feature, whereby the controller automatically shifted between the powershift gear ratios to maintain a consistent engine load (as indicated by a range of engine speeds). In the lower gear range, the driver had to reselect autoshift after making the 4/5 change in the usual manner. In the upper range the synchromesh change (12/13) was made automatically by initially pressing the autoshift button twice. This feature was linked to 3pt. hitch operation. Use of the fast raise/lower switches put the feature into standby upon raising and then reactivated it, upon lowering, to ease implement headland manoeuvres. To suit different applications the variation in engine speed required to instigate the shift could be adjusted by the operator, a 20% speed variation being the default, but equally this could be changed to 5, 10, 15 or 25% (for the gears 13-16 a minimum of 20% was always set).

# 4 Vehicle Model

## 4.1 Introduction & Objective

As has been described previously (see Section 1.1), the variable nature of agricultural conditions, together with the wide variety of operations undertaken by a tractor, serves to create highly variable imposed loads. These often make it difficult to fully understand the dynamic characteristics of the tractor. Mathematical models assist by allowing the effects of each parameter on the overall system to be determined, provided the model is an accurate representation of the physical system. Mathematical models also allow the tractor-implement system to be investigated and allow the development of control strategies in a controlled environment, and far quicker than would be possible with prototype systems on the tractor. If a modular modelling approach is used, the effect of substituting different vehicle sub-systems can also be determined, for example a different engine or transmission design.

Mathematical simulation models can, according to Law and Kelton (1982), be classified into two main types: static and dynamic. Static models represent a system at a particular time and therefore consider force, mass and velocity. Dynamic models represent a system as it evolves over time and therefore also considers inertias and accelerations. For this investigation it was desirable to develop a dynamic model from first principles. However, it was identified that, due to time and information limitations, it would also be necessary to use empirically determined model data as required.

The objective of this part of the investigation was to develop a dynamic mathematical model of the test tractor-implement system, to aid understanding of the tractor powertrain characteristics and to allow the development of improved control strategies.

## 4.2 Previous Work

### 4.2.1 Tractor Performance Models

As the tractor is the prime power source for most agricultural applications, the vehicle has been subject to a number of previous investigations for a wide variety of purposes and to differing levels of complexity.

Early steady state work (Zoz, 1970; Wismer and Luth, 1972; Brixius, 1987; Zoz, 1987) used graphical curve-fitting techniques, field data and fundamental equations to predict drawbar performance and travel speed under varying field conditions. The focus of these investigations was the soil-tyre interface. The remainder of the vehicle was considered in very simple terms. These models however are still relevant today, forming the basis of an interactive PC-based performance model by Al-Hamed and Al-Janobi (2001). An alternative approach to basing tyre performance on the empirical relationships of Wismer and Luth (1972), instead using mobility numbers, was developed by Dwyer (1984) to predict power output and workrate under steady state conditions.

Jahns and Steinkampf (1983) developed a steady state model for the purposes of analysing the effects of a number of tractor and implement parameters on tractor operational efficiency. The aim was either maximising output or reducing fuel consumption; a modular format was used for the engine, driveline, soil-tyre interface and implement forming sub-models. A modular approach has been used by many other workers, including Scarlett (1995) to consider the impact of machine and soil parameters on steady state performance, for the purposes of control system development. Both these investigations utilised experimental relationships between drawbar pull and slip in the soil-tyre interaction parts of the model.

One of the earliest dynamic models found in the literature was developed by Crolla (1975) to predict the performance of a tractor operating cultivation equipment. Comparisons to previous-steady state models showed that tractive efficiency and therefore performance were reduced under dynamic conditions. The validated model was used to determine the effects of different parameters, particularly those relating to draught control on dynamic tractor-implement performance. McMullan (1981) described a family of dynamic simulation programmes, including the aforementioned work of Crolla (1975), to investigate implement control systems, including an experimental electro-hydraulic hitch control system. A dynamic model for similar purposes was developed by Olson and Cornell (1987), although some parts have been simplified, such as the engine to a steady state torque-speed curve, and the transmission model ignores any driveline inertia effects.

### 4.2.2 Other Vehicle Models

A number of other non-tractor performance models are worthy of mention here, and the principles involved have been taken into consideration during the development of the model for this investigation. Hohl (1990) developed a dynamic model to predict the acceleration and maximum velocity of wheeled, off-road vehicles. The system was reduced to two lumped rotational inertias, connected by the master clutch, to which the equations of motion were applied. The interaction with the ground surface was accounted for with a slip-coefficient of traction curve, although little is discussed about this.

The same approach of lumping components before and after the clutch was taken by Phillips *et al* (1990) for their simulation model to predict fuel economy and performance of a 5 tonne truck. Equations of motion were used to describe the relationships between components for each half and to account for rolling resistance, gradient and aerodynamic losses. As the performance being considered was on-road, no complex soil-wheel interactions needed to be determined. The model also simplified driveline losses to a single efficiency value.

## 4.3 Modelling Approach

### 4.3.1 Overview

On the basis of previous work, a dynamic model based on fundamental Newtonian principles was still deemed to be the most appropriate approach to take. The tractor-implement system was considered as five inter-connected sub-systems (engine, driveline, traction interface, vehicle and implement), which together form the overall system model (see Figure 4-1). This approach not only simplified the modelling task, but also allowed for the easy replacement of one or more sub-models. For example it would be possible to replace the driveline model to investigate the performance of an alternative transmission system. In this investigation, the use of sub-models allowed different implements to be considered; namely a power harrow, plough or trailer.

The representation was simplified by only considering one traction interface point at which the entire vehicle mass is applied. The hitch system was also omitted as field experiments (see Section 5) were undertaken with the 3 pt. hitch operating in position control only. This resulted in changes in working depth being limited to those made by the operator during transient experiments. As the soil engaging implements were operated either in semi-mounted form (plough) or, in the case of the power harrow, with depth controlled by the rear roller rather than the 3pt. hitch, the weight transfer to the tractor was relatively small and constant: it was therefore not considered in the simulation., this simplification is validated by the field data (see Appendix A4). A number of specific sub-system simplifications were made, each of which is discussed within the appropriate description.

### 4.3.2 Operation

The model sub-systems are linked in a loop (see Figure 4-1) with forces and torques being transferred from the implement, through each sub-system, to the engine. The determined engine speed from this torque load is then transferred back through each sub-system to the traction interface, where the vehicle forward speed is obtained. This forward speed is also passed to the implement to allow calculation of the draught force (and P.T.O. torque) requirement for the next time step.

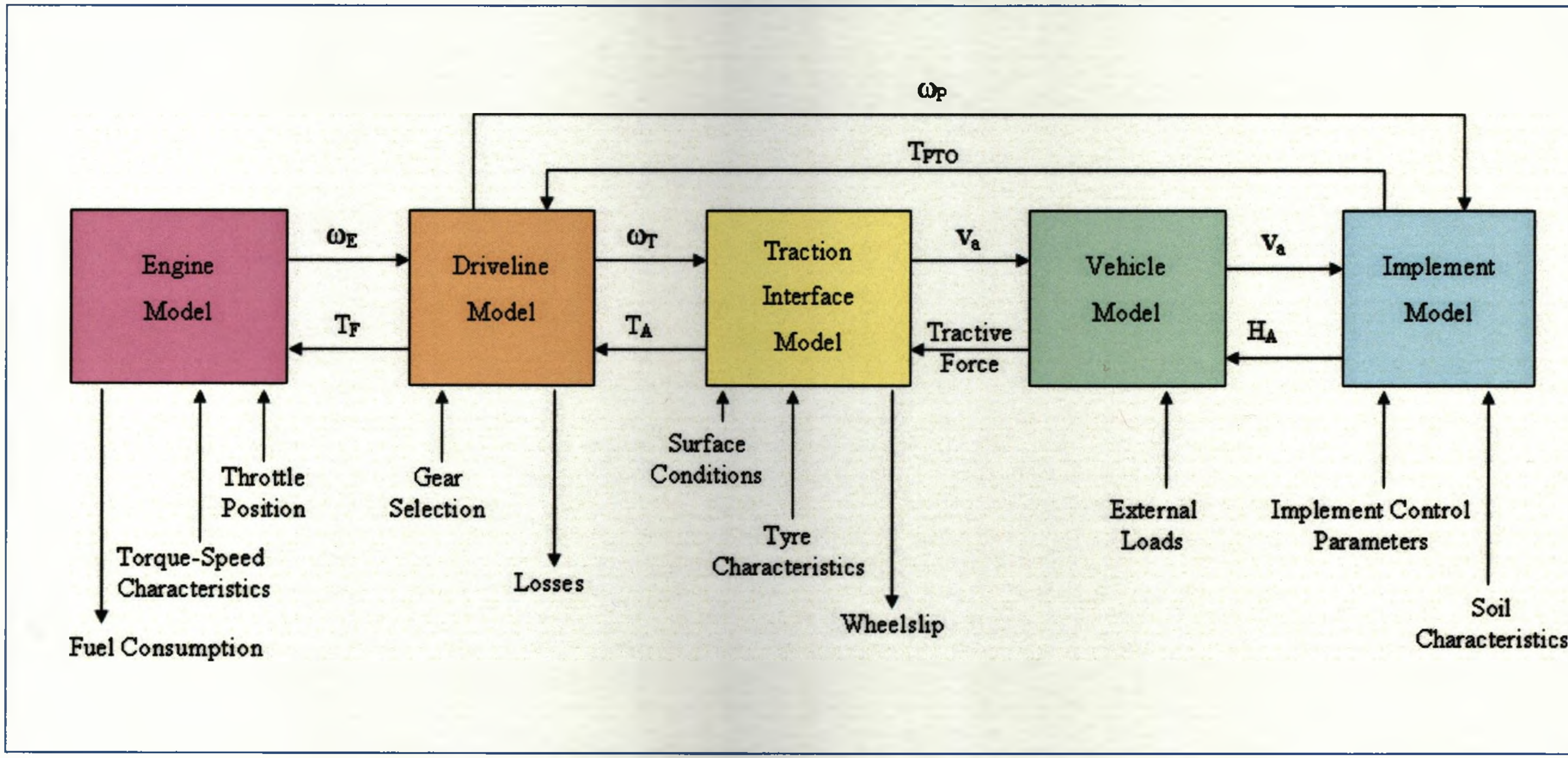



*Figure 4-1 – Modular representation of the tractor-implement combination*

### 4.3.3 Modelling Software

The model was developed using Simulink, part of the MATLAB® family of programmes. Simulink is a software package for modelling, simulating and analysing dynamic systems. A graphical user interface allowed models to be constructed using block diagram representations of the underlying equations for each sub-system and the overall model.

A limitation of Simulink is that all parameters used in the model must be represented with standard roman alpha-numeric characters. No provision is made to allow the use of Greek or subscript symbols. Table 4.1 shows the model parameters and the equivalent notation used in the remainder of this investigation. A number of additional parameters were also used in the model; these are shown in Table 4.2. Any model parameters not detailed in these two tables can be found in the M-file (see Appendix A2.1).

*Table 4.1 – Model parameters and their equivalent notation*

| Source sub-system | Model Parameter | Term Description | Units | Equivalent Parameter: |
|---|---|---|---|---|
| Test Inputs | foot_th | foot throttle | % | [illegible] |
| Test Inputs | gear | gear specify | - | G |
| Test Inputs | pld | plough working depth | mm | $D_P$ |
| Test Inputs | plw | plough working width | mm | $W_P$ |
| Test Inputs | phd | power harrow working depth | mm | $D_{HA}$ |
| Engine | fuel_q | actual fuel quantity injected | mg/stroke | Q |
| Engine | erpm | engine speed | rpm | $\omega_E$ |
| Engine | fly_tq | flywheel torque | Nm | $T_F$ |
| Engine | accel_tq | torque to accelerate | Nm | $v_a$ |
| Engine (Governor Droop) | droop | Engine droop mode:<br>0 = 0%    1 = 5%    2 = 12% | - | $\delta l$ |
| Engine (Full Load Curves) | bst_stat | boost status:<br>2 = Standby    3 = Control Initiated<br>4 = non-static boost | - | $B_s$ |
| Engine (P&FC) | fly_pwr | flywheel power | kW | $P_F$ |
| Driveline | PTO_tq | PTO torque at flywheel | Nm | $T_{PTO}$ |
| Traction Interface | slip | wheelslip | % | S |
| Vehicle | for_spd | vehicle forward speed | km/h | $v_a$ |
| Vehicle | pull | drawbar pull | kN | $H_A$ |

*Table 4.2 – Additional model parameters*

| Source sub-system | Model Parameter | Term Description | Units |
|---|---|---|---|
| Top Level | imp_spec | Identifies current implement selection:<br>1 = power harrow 2 = plough<br>3 = trailer 4 = none | - |
| Test Inputs | hand_th | hand throttle | % |
| Test Inputs | Fi | Soil texture adjustment<br>1=fine; 0.7=medium; 0.45=coarse | - |
| Test Inputs | ST | Soil type: 1=sandy; 2 = clay | - |
| Test Inputs | grade | field/road gradient | % |
| Test Inputs | ISO | Selection of isochronous governor:<br>0=ISO; 1 = Speed dependant droop | - |
| Engine | sps0 | set point speed under no load | rpm |
| Engine | sps | set point speed | rpm |
| Engine | s_error | speed error | rpm |
| Engine | fpi | fuel quantity from PI controller | mg/stroke |
| Engine | fuel_q | actual fuel quantity injected | mg/stroke |
| Engine | brake_tq | brake torque (torque produced less friction) | Nm |
| Engine | fan_tq | fan torque | Nm |
| Engine | ip_tq | total input torque (from vehicle) | Nm |
| Engine (P&FC) | sfc | specific fuel consumption | g/kw.hr |
| Engine (P&FC) | afc | actual fuel consumption | litres/hr |

The source of the parameters detailed in Tables 4.1 and 4.2 are highlighted by orange shading in the appropriate model sub-system. Green shading has been used to show where these parameters are used in the model. Light blue shading is used to show the in-ports and out-ports from each sub-system. Where a sub-system has additional sub-models, these are shaded in yellow with the name inside the box. Additional model data, primarily for use in lookup tables, was written as a MATLAB M-file and is shown in Appendix A2.1.

## *4.4 Model Descriptions*

### 4.4.1 Overall Model

The overall model in the Simulink environment is shown in Figure 4-2. The main difference to the generic representation shown in Figure 4-1 is the inclusion of the different implements. The user selects an implement prior to commencing the simulation by entering its number in the 'implement selector' box. The multiport switch then connects that implement to the tractor sub-systems.

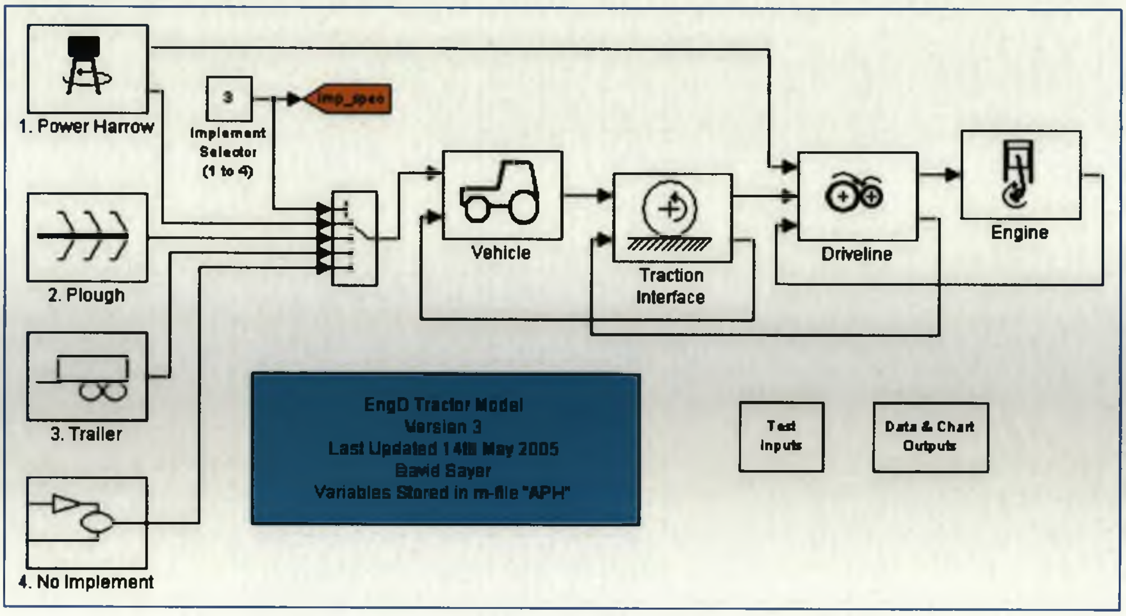



*Figure 4-2 – Overall tractor-implement block diagram*

In addition to the tractor-implement sub-systems, the overall model also contains the provision for the user to enter additional test parameters ('Test Inputs') and to output data from the model in numerical or graphical format ('Data & Chart Outputs').

## 4.4.2 Engine

### *4.4.2.1 Outline Engine Model*

This sub-system is the most complex to represent, primarily as a result of the engine control features and characteristics previously described (see Section 3.2). This complexity resulted in the engine model being formed with a number of additional sub-models operating within it. Figure 4-3 shows the engine model block diagram structure, including the sub-models (shaded yellow).

Returning to the main engine model, the overlying principle is that at each time step the torque available to accelerate the engine is calculated. The rate of engine acceleration is then determined by the engine inertia:

$$\dot{\omega}_E = \frac{T_{ACC}}{I_E}$$ *Equation 4-1*

The engine acceleration is then integrated to give the engine speed. At the next time step, this engine speed is compared to the set point engine speed and the speed error determined. The set point engine speed is determined from the hand and foot throttle settings, using lookup tables, together with any reductions from the governor (see Section 4.4.2.2). This speed error is used by the fuel controller to determine the new fuel quantity. The actual fuel quantity delivered is limited by the active full-load fuel curve (see Section 4.4.2.4).

Given the fuel quantity injected and the engine speed, the engine brake torque (the engine output torque with no accessories or load) is determined from a 2-dimensional lookup table. This data was determined from a bare engine on a dynamometer and gives a linear relationship between torque and fuel quantity injected, at a constant engine speed (see Appendix A2.2). The torque required to drive the engine cooling fan is then subtracted from the brake torque to give the flywheel torque. Finally, the torque required from the driveline sub-system at the new time step is subtracted from the flywheel torque to give the new torque available for engine acceleration.

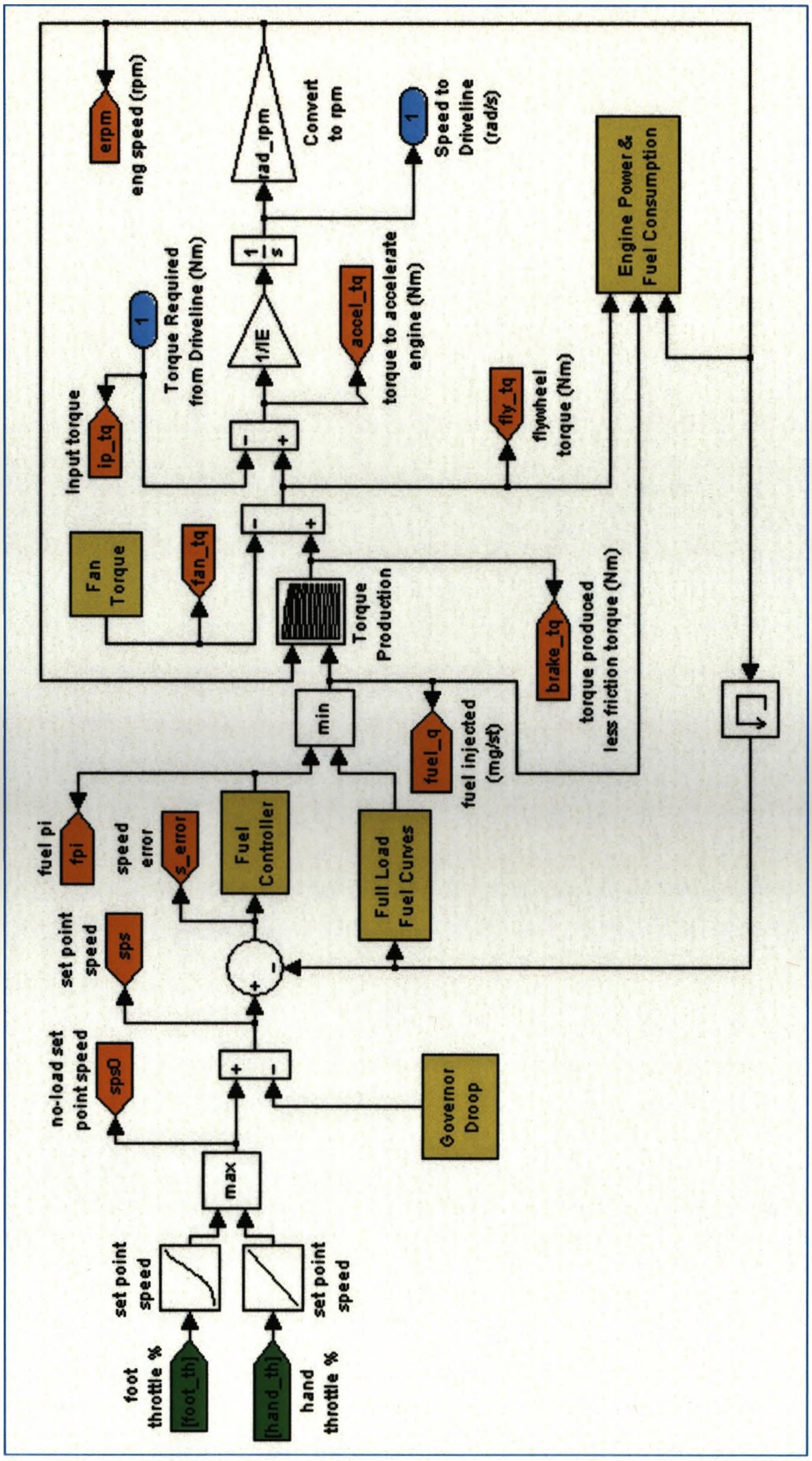



*Figure 4-3 – Engine sub-system block diagram*

#### *4.4.2.2 Governor Droop*

As previously described in Section 3.2, the test tractor's engine featured three droop settings ($\delta_I$). Selection of 5% or 12% droop is made according to tractor forward speed, with 0% (Isochronous) being selected by the user. To account for the effect of droop, the set point engine speed is reduced according to the throttle position and the engine load.

The linear relationship between fuel quantity and torque allows the engine full-load performance curve to be considered as a fuel-speed relationship (see Figure 4-4). If the engine is being operated at the point indicated in Figure 4-4, engine speed is reduced from the no-load speed ($\omega_{E2}$) to $\omega_{E1}$, at the same time the quantity of fuel injected increases from $Q_{NL}$ to Q.

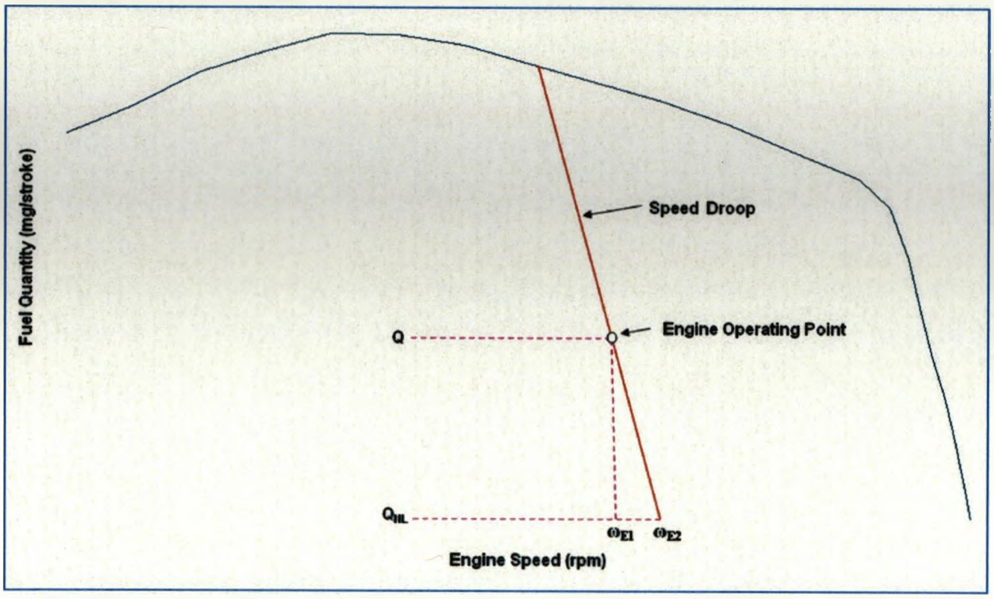



*Figure 4-4 – The relationship between fuel quantity and engine speed with speed droop*

The feedback factor (FBF) gives the reduction in engine speed for a given increase in fuel quantity and is determined by:

$$FBF = \frac{\omega_{E2} - \omega_{E1}}{Q - Q_{NL}} \qquad \textit{Equation 4-2}$$

Equation 4-2 can be re-arranged to determine the loaded engine speed ($\omega_{E1}$):

$$\omega_{E1} = \omega_{E2} - FBF(Q - Q_{NL}) \qquad \textit{Equation 4-3}$$

The feedback factor for each droop setting is detailed in the M-file (Appendix A2.1). The governor droop sub-model (see Figure 4-5) therefore determines the speed to be subtracted from the set-point speed in the manner described above.

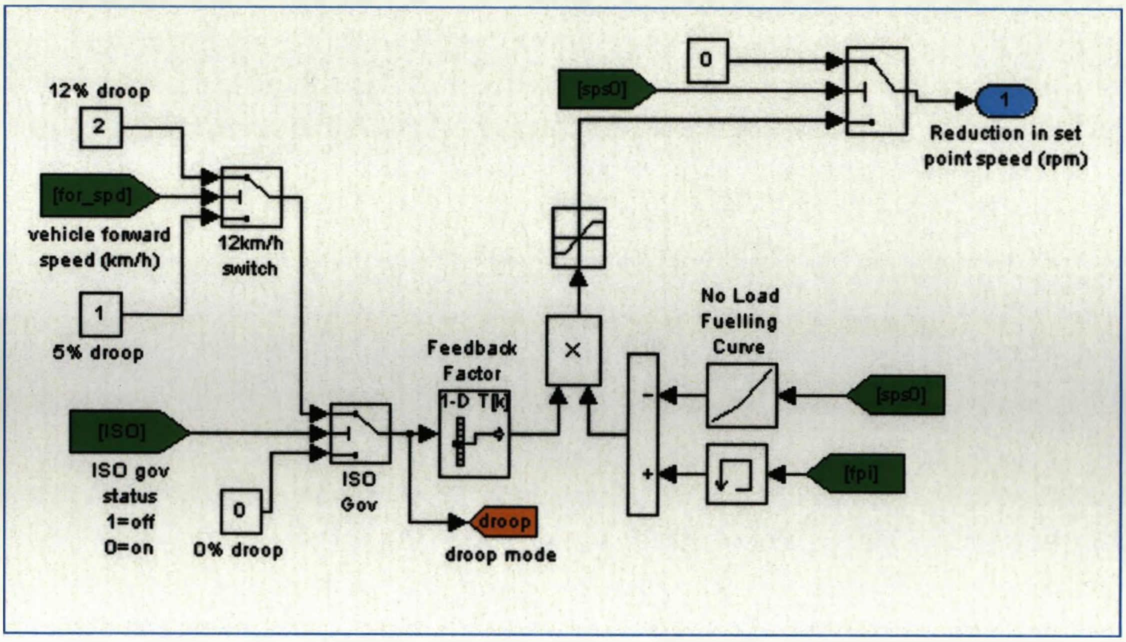



*Figure 4-5 – Governor droop sub-model block diagram*

#### 4.4.2.3 *Fuel Controller*

The fuel controller sub-model determines the fuel quantity at each time step from the speed error. The Controller used a standard Proportional + Integral (PI) controller. To determine the optimum PI setting, the engine sub-system response to changes in set-point speed and load were compared with P.T.O. dynamometer data. The fuel controller block diagram is shown in Figure 4-6.

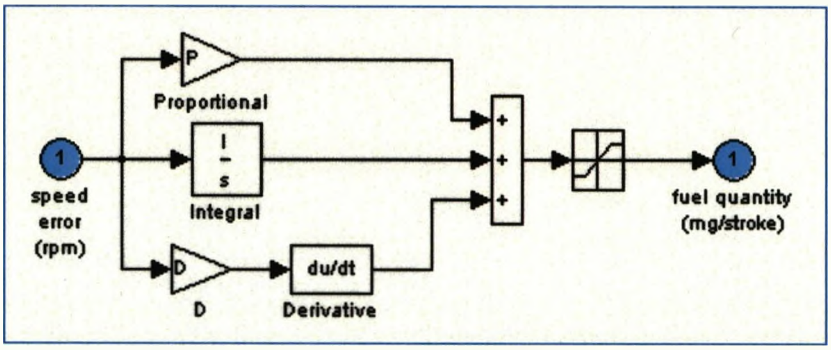



*Figure 4-6 – Fuel controller block diagram*

### 4.4.2.4 Full-Load Fuel Curves

When operating under governor control, the fuel controller sub-model determines the fuelling based on speed error. However, the engine torque output is limited by the full-load fuel curve. In the case of the test tractor engine there are two full-load curves, relating to the tractor operating in boosted and unboosted mode (see Section 3.2). This sub-model uses engine speed to determine the maximum fuel quantity allowable. The power boost logic is also included to determine which curve can be used. As described in Section 3.2, power boost can only be implemented for non-static P.T.O. operation provided there is a minimum torque of 250Nm attributed to the P.T.O. driveline and engine speed is above the minimum threshold. Logic is then used to determine which maximum fuelling curve is used. Figure 4-7 shows the block diagram for this sub-model, the output of which is then compared to the fuel controller output, and the minimum fuel quantity is used to determine engine torque.

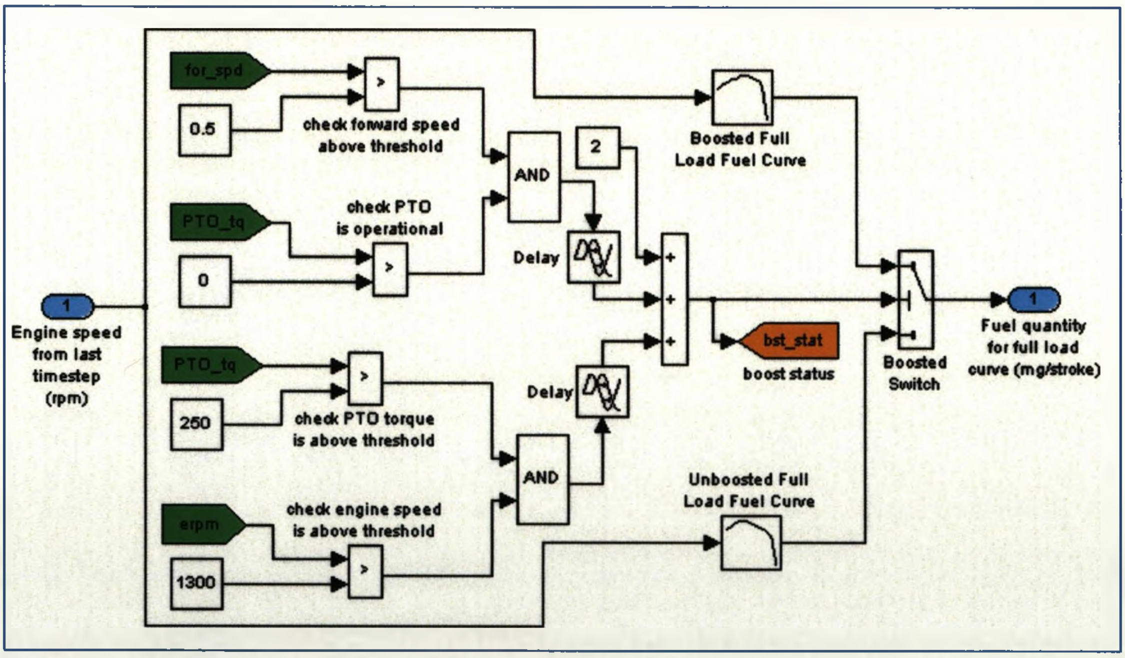



*Figure 4-7 – Full-load boosted fuel curves block diagram*

### *4.4.2.5 Cooling Fan Torque*

The test tractor engine uses a viscous fan to provide cooling. The power requirements of the fan and coupling vary as a function of fan speed (see Figure 4-8).

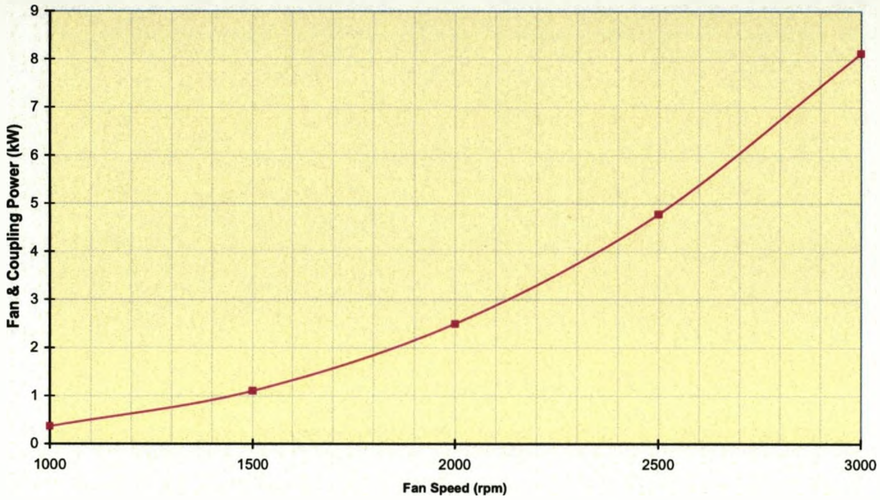



*Figure 4-8 – Fan power consumption as a function of fan speed*

Unfortunately, the viscous fan coupling also allows the fan speed to vary as a result of variations in engine cooling requirements. Steady-state P.T.O. dynamometer test data was used to determine fan speed ($\omega_F$) from engine speed and flywheel power, using multiple regression techniques (Equation 4-4). The R.M.S. error of prediction was 94rpm.

$$\omega_F = (0.9604 \times \omega_E) + (7.012 \times P_F) \qquad \textit{Equation 4-4}$$

Using the estimate for fan speed, the resultant fan power requirement was determined from the relationship in Figure 4-8. This was then divided by engine speed to give the additional torque load at the engine flywheel. The block diagram of the fan sub-model is shown in Figure 4-9.

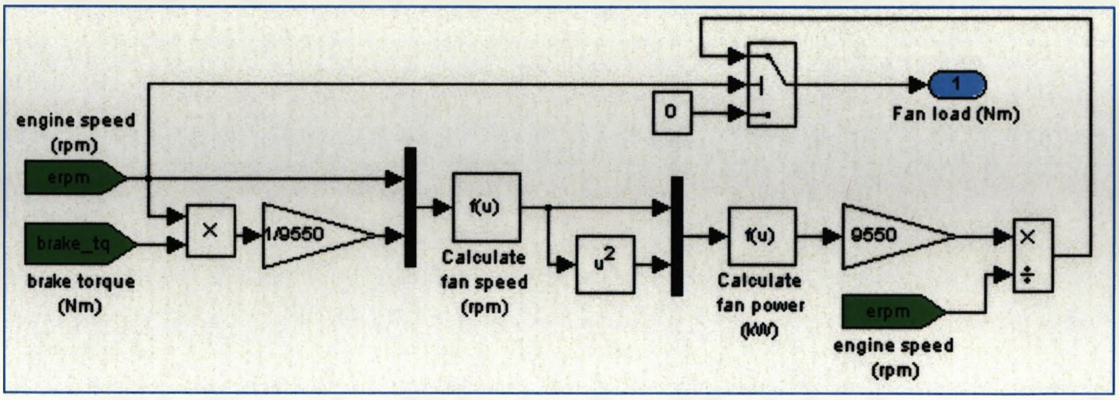



*Figure 4-9 – Viscous fan torque requirement block diagram*

### *4.4.2.6 Engine Power and Fuel Consumption*

The engine power is calculated from engine speed and flywheel torque. Actual fuel consumption is determined from the fuel quantity used, together with the engine speed, with a conversion to give a quantity in litres per hour. Specific fuel consumption is then calculated, taking into account the engine power developed. The block diagram for this sub-model is shown in Appendix A2.3.

### 4.4.3 Driveline

#### *4.4.3.1 Outline Driveline Model*

The driveline sub-system has both torque and speed paths connecting it to the engine and the traction interface (see Figure 4-10). The torque-related part of the model receives the torque requirement at the axle ends and reduces it by the rear axle ratio and the appropriate transmission gear ratio. The torque required to overcome driveline losses is then added to the flywheel equivalent axle torque (see Section 4.4.3.2). Flywheel equivalent P.T.O. torque from the implement sub-system is added, as is the torque to overcome driveline inertia during acceleration. The equivalent driveline inertia in each transmission gear ratio was determined separately during this investigation (see Section 6.6).

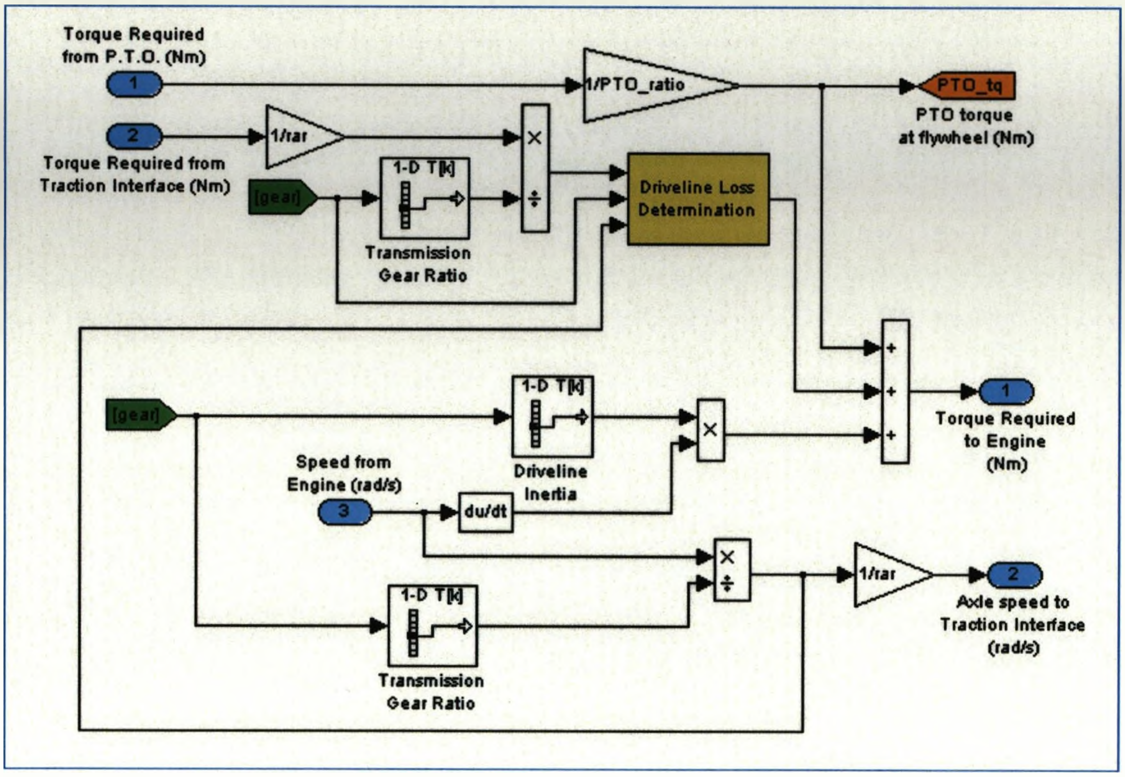



*Figure 4-10 - Driveline sub-system block diagram*

The speed-related part of the model takes the engine speed and reduces it by both the transmission gear ratio and the rear axle ratio, before outputting it to the traction interface model. The transmission gear and rear axle ratios were previously defined in Table 3.1; the P.T.O. ratio was 2.12:1. This data, together with the equivalent driveline inertia in each gear, was obtained from the M-file.

The driveline sub-system did not take into account the disruption in torque transfer during a gearshift. This should be included in subsequent model development if higher speed operation is to be considered.

#### *4.4.3.2 Driveline Torque Losses*

A thorough investigation of the torque losses through the driveline was conducted as part of this investigation (see Section 6.5). The outcome was a driveline model (Equation 4-5) to predict the torque at the flywheel from the flywheel equivalent axle torque ($T_{EF}$), the transmission output speed ($\omega_T$) and the number of active gear meshes in the selected gear ratio ($G_m$).

$$T_F = (1.01 \times T_{EF}) + (0.016 \times \omega_T) + (1.7 \times G_m) \qquad \textit{Equation 4-5}$$

The block diagram representing this equation is shown in Figure 4-11.

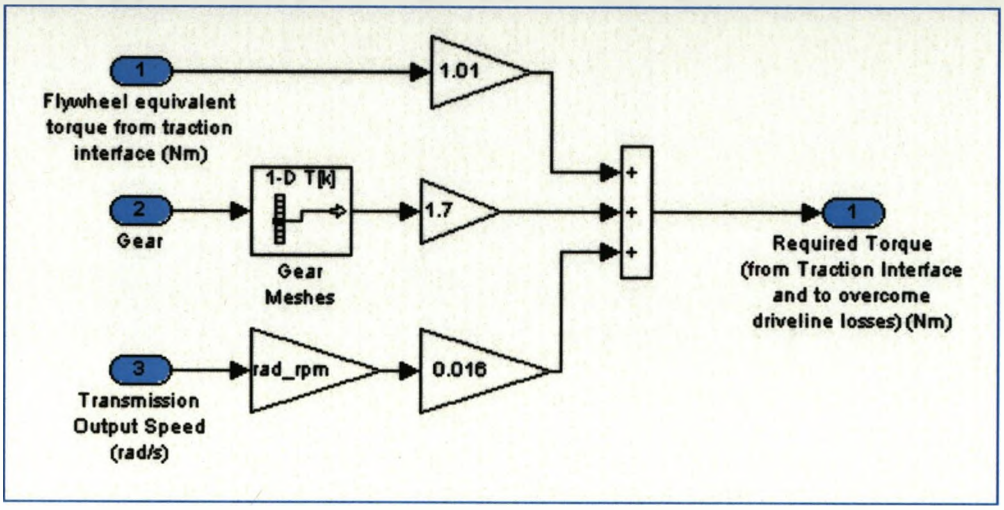



*Figure 4-11 – Driveline torque loss sub-model block diagram*

### 4.4.4 Traction Interface

This sub-system (see Figure 4-12) transforms the force from the vehicle sub-system into a torque at the axle end, by multiplying by the loaded wheel radius.

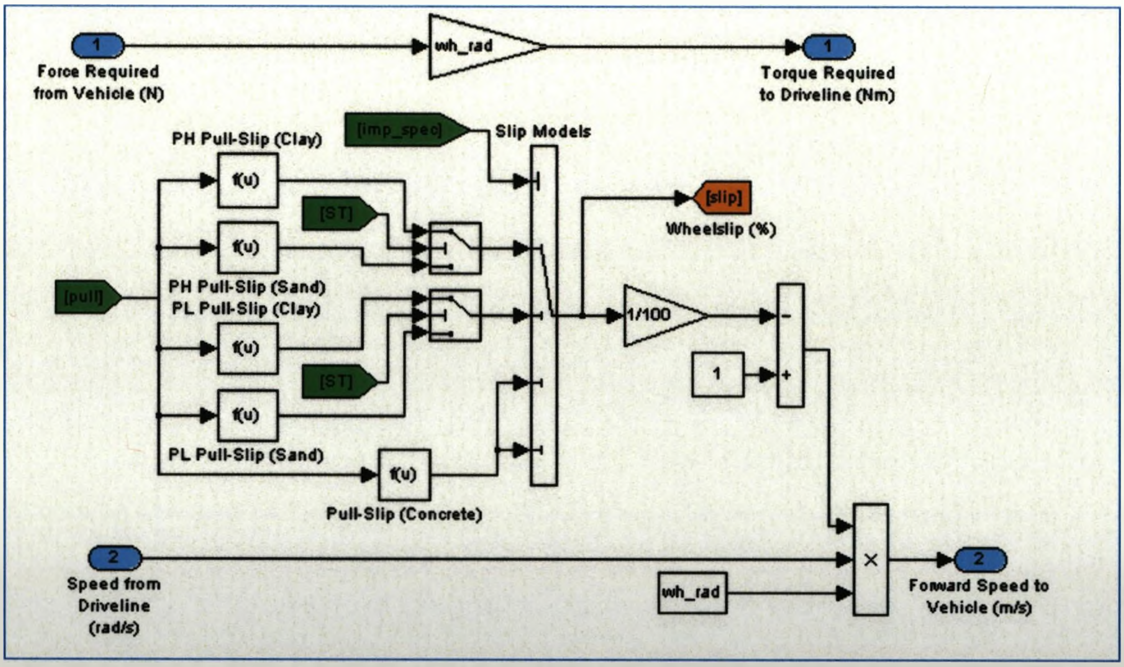



*Figure 4-12 – Traction interface sub-system block diagram*

The speed-related part of this sub-system considers the traction (slip-pull) characteristics experienced on each surface. The traction characteristics for plough and power harrow operation were derived from field data obtained during the field and road investigation (see Section 5). Ploughing slip-pull characteristics are shown in Figure 4-13 and power harrowing slip-pull characteristics are shown in Figure 4-14. Using field data restricts the operating range of the model, however, as the field data was collected across a wide operating envelope the majority of practical operating scenarios are accounted for. For on-road applications, slip-pull characteristics were determined from OECD test data on the equivalent production version of the test tractor (see Figure 4-15).

Axle speed from the driveline is multiplied by the wheel loaded radius to give a theoretical forward speed, which is then reduced according to the level of wheelslip.

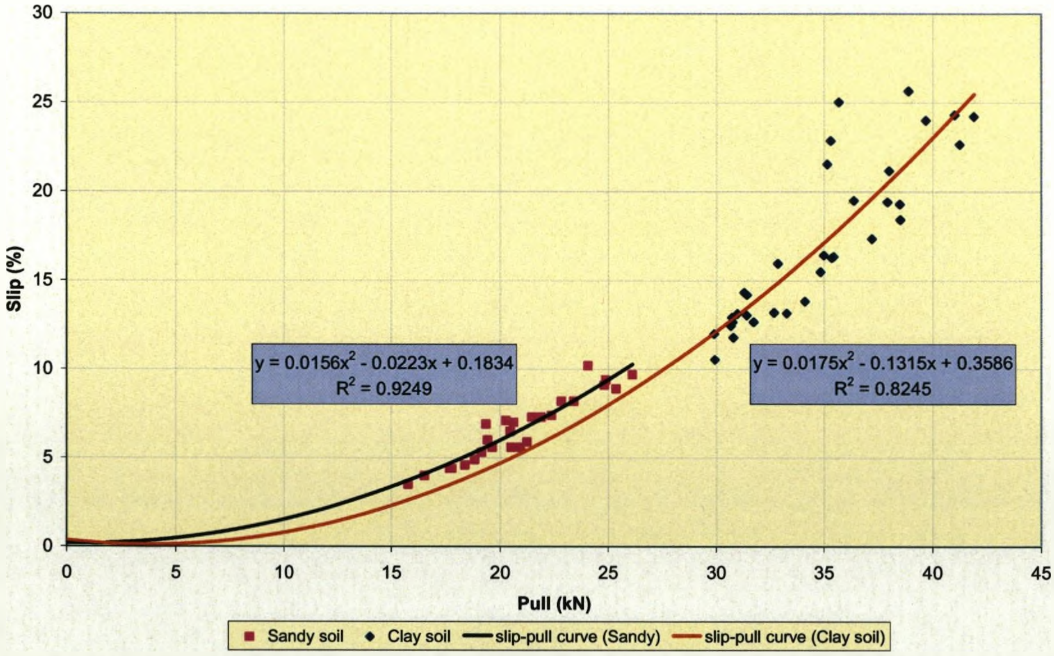



*Figure 4-13 – Tractor slip-pull characteristics whilst ploughing*

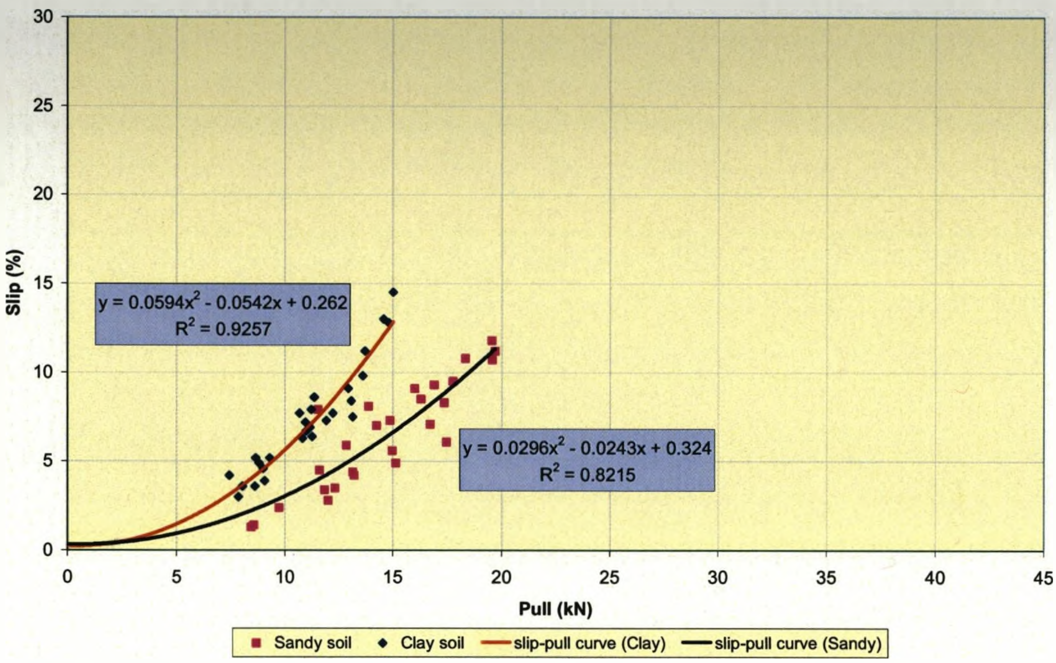



*Figure 4-14 – Tractor slip-pull characteristics whilst power harrowing*

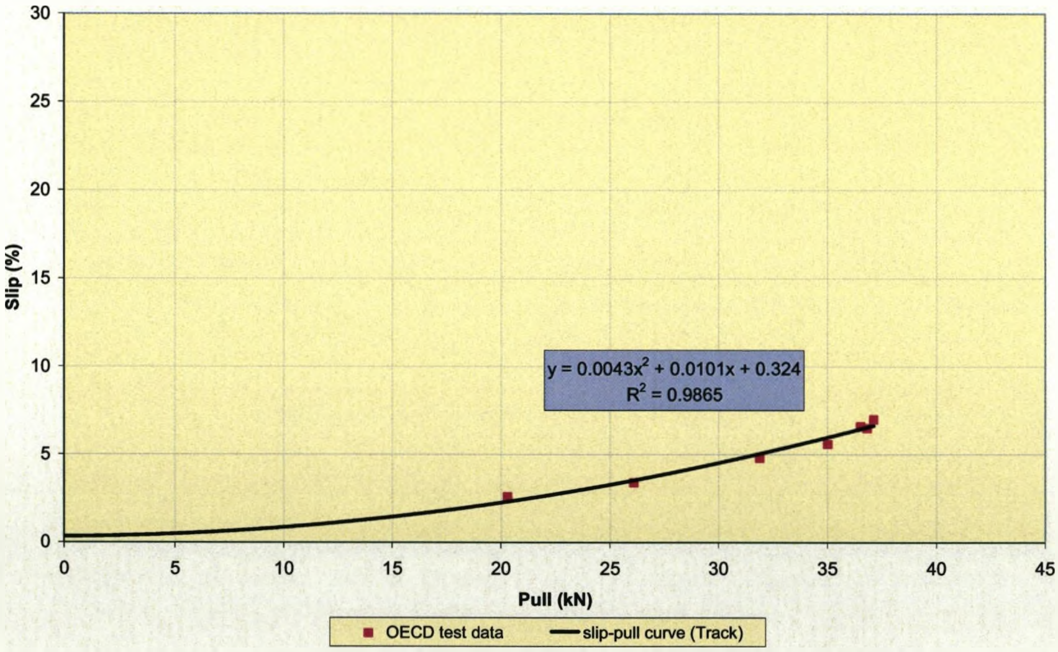



*Figure 4-15 – Tractor slip-pull characteristics on concrete*

## 4.4.5 Vehicle

### *4.4.5.1 Outline Vehicle Model*

This sub-system considered the additional forces required to overcome rolling resistance, vehicle acceleration and slopes. These forces are added to the drawbar pull requirement from the implement. The forward speed from the traction interface is also transformed into the vehicle forward speed (km/h).

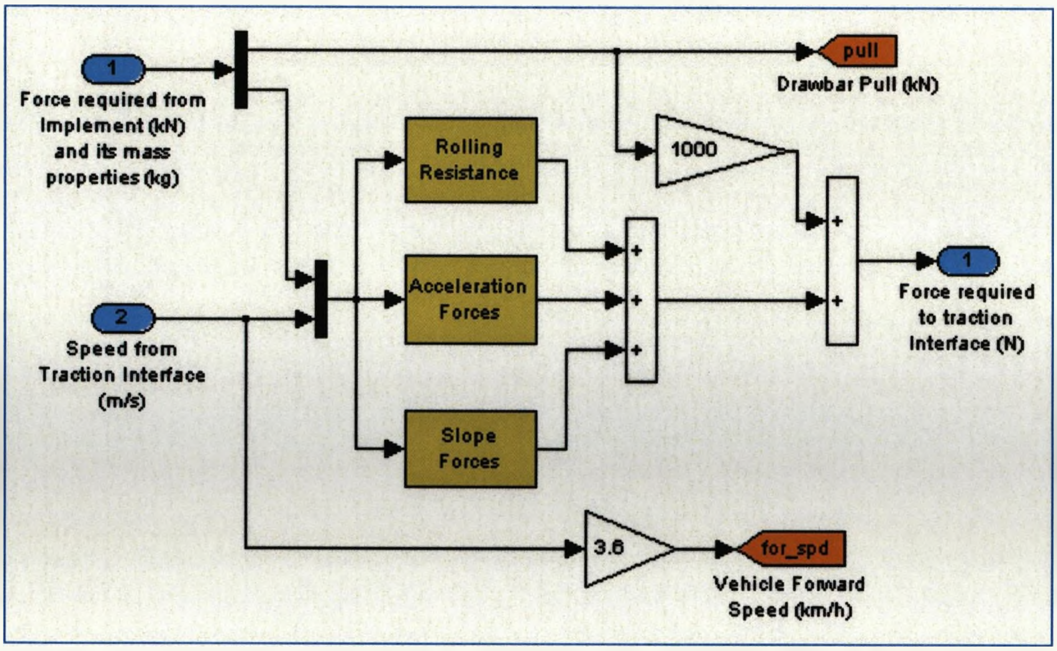



*Figure 4-16 – Vehicle sub-system block diagram*

### *4.4.5.2 Rolling Resistance*

Rolling resistance is non-recoverable energy expended in deforming the tyre, and for off road applications, the surface. Losses due to rolling resistance are highly complex with many factors needing to be taken into account. Primarily, rolling resistance losses for off-highway vehicles are a function of the weight on the wheel, wheel diameter and soil conditions. Secondary factors which influence rolling resistance include vehicle forward speed, tyre temperature, inflation pressure, tyre tread and the level of torque transmitted.

Many workers, including Gillespie (1992) and Lucas (1986) have simplified the problem of accounting for rolling resistance by considering the vehicle weight multiplied by a coefficient of rolling resistance ($C_{RR}$) which varies according to the tyre dimensions and the surface. An alternative approach is to calculate mobility numbers for each surface. This method, based on the work of Frietag (1965) calculates a mobility number from surface resistance (cone index), tyre mass and dimensional properties. This number is then used to calculate various performance values, including rolling resistance.

For this part of the investigation, coefficients of rolling resistance based on tyre size and surface type were used with values being those stated by Macmillan (2002) for a comparable tyre size: 0.02 for operating on a hard concrete (or tarmac) surface; 0.095 for ploughing on dry stubble and 0.15 for power harrowing on ploughed soil (see Appendix A2.3 for the block diagram). Mobility numbers were not considered as cone index data was not available for the surfaces investigated, therefore an approximation would have been required, hence making the methodology no more accurate than the one chosen.

#### *4.4.5.3 Acceleration*

The force required to accelerate the tractor and implement is determined, from Newton's second law, by their mass multiplied by the rate of vehicle acceleration (see Appendix A2.3 for the block diagram).

#### *4.4.5.4 Slope*

The additional force required to operate on a slope is determined by the component of the combined tractor and implement weight acting perpendicular to the slope (see Appendix A2.3 for the block diagram):

$$F = (M_T + M_I) \times g \times Sin\theta \qquad \textit{Equation 4-6}$$

### 4.4.6 Implement

#### *4.4.6.1 Power Harrow*

During power harrowing, the total load imposed on the tractor is determined by the draught force and P.T.O. torque requirement. Bentley (2000) identified that for a constant rotor speed, power harrow draught force and P.T.O. torque were proportional to forward speed and working depth. In the absence of a suitable theoretical model, power harrow steady state experimental data (see Section 5.5) was used to generate 2-dimensional lookup tables from which the model could determine draught force and P.T.O. torque for either a sandy soil or a clay soil.

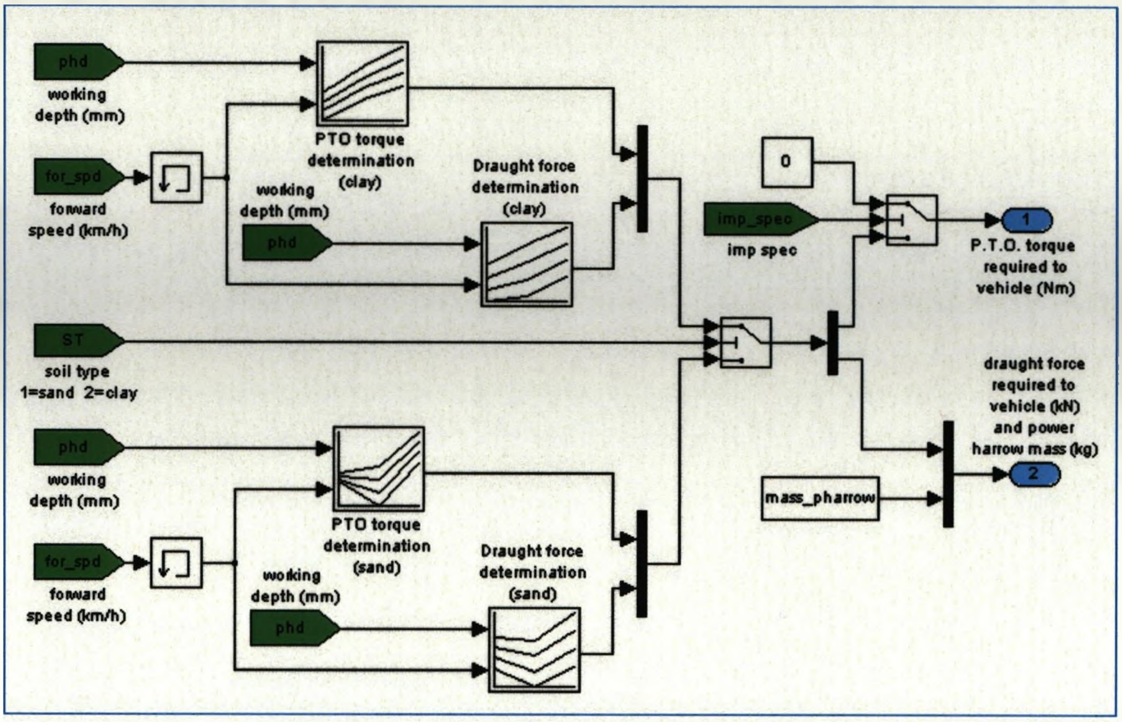



*Figure 4-17 – Power harrow sub-system block diagram*

#### *4.4.6.2 Plough*

The draught force requirement of a mouldboard plough is primarily a function of working depth, plough width, forward speed and soil texture. Plough draught was calculated by use of a standard equation for a mouldboard plough, obtained from

ASAE agricultural machinery management data (American Society of Agricultural Engineers, 2003):

$$H_A = 0.001 \times Fi \times \left[A + (B \times v_a) + (C \times v_a^{\,2})\right] \times \frac{W_P}{1000} \times \frac{D_P}{10} \qquad \textit{Equation 4-7}$$

where A, B and C are plough-specific parameters and Fi is a dimensionless parameter based on soil texture (see Table 4.2). The output of Equation 4-7 for the plough used in this investigation for three different soil conditions is shown in Figure 4-18:

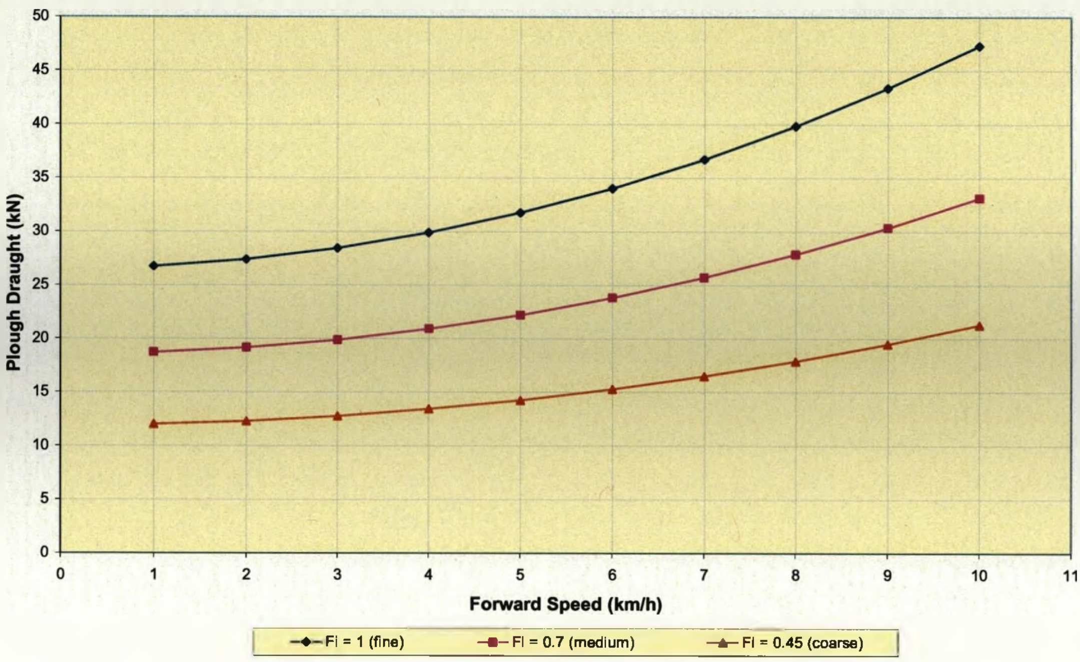



*Figure 4-18 – Typical output of the plough draught equation: (DP = 229m, WP = 1778mm)*

The Simulink block diagram for the plough sub-system is shown in Figure 4-19.

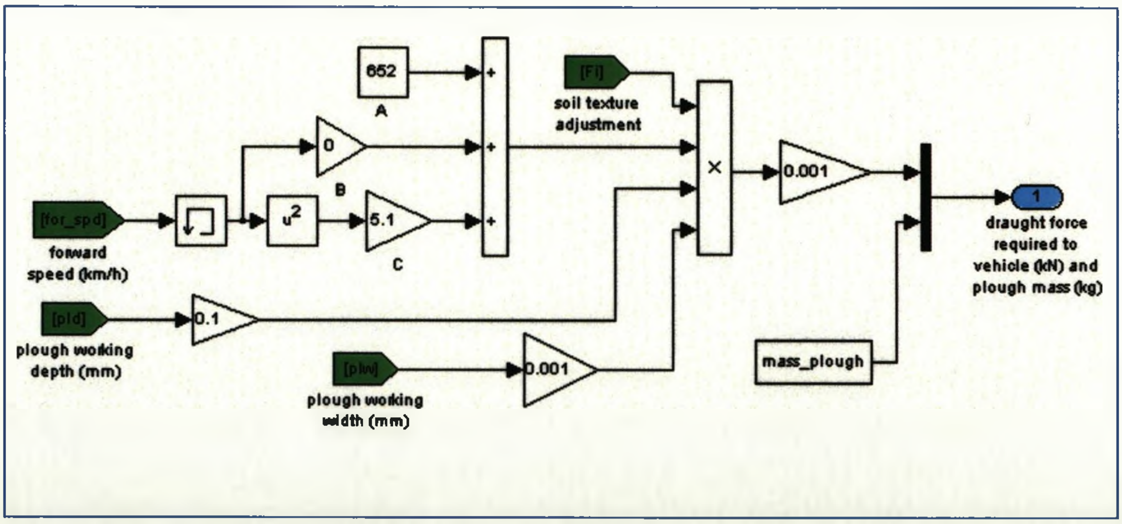



*Figure 4-19 – Plough sub-system block diagram*

#### 4.4.6.3 Trailer

During transport operations, the mass of the trailer must be considered in addition to tractor mass, to determine rolling resistance and slope-related forces. This sub-system was used to bring the trailer mass into the model (from the M-file).

## 4.5 Model Inputs

The tractor-implement model can be used in a variety of different ways, and therefore the type of inputs required also differs. In most situations the model requires the throttle and gear to be specified, together with implement and soil parameters. These can either be single values, or as an input file with respect to time. In the case of the soil parameters, a random noise signal within limits could be used to define the soil texture coefficient (Fi). Alternatively the implement torque and force requirements as measured in the field could be substituted in place of the implement sub-system.

## 4.6 Model Validation

The model was validated using data from the field investigations; the validation procedure is presented in Section 7.

# 5 Field & Road Investigation

## *5.1 Introduction & Overall Objectives*

An extensive field test programme was devised to provide data for use in development and validation of powertrain models and the axle dynamometer experiments. In addition, the field test programme potentially allowed powertrain characteristics and trends to be identified, as well as areas for further investigation and improvement. Whilst there is some historical data relating to agricultural tractor powertrain loading in existence (Scarlett *et al*, 1998; Hansen *et al*, 1986; Kim *et al*, 2000), powertrain characteristics are directly related to the component design and configuration of the vehicle under investigation.

In order to improve the operational characteristics of tractors, it is essential not only to understand the dynamic variation in loads due to changing external conditions (e.g. soil strength) for a defined tractor-implement configuration, but also to determine the effects of specific load transients (e.g. gearshifts) on the vehicle powertrain. Consequently the objectives of this section of the investigation were:

- to quantify the dynamic variation in powertrain loading encountered whilst performing a range of agricultural operations with a number of pre-determined tractor-implement configurations (steady state tests); and
- to determine the effects of specific user-induced transients on the test vehicle powertrain (transient tests).

A range of operations were selected to reflect not only typical usage for a tractor of this engine power, but also to load the tractor in the different ways that would be experienced during practical farm work, these being:

- mouldboard ploughing (a low-medium speed operation with a high draught requirement);
- power harrowing (a low speed operation with a high P.T.O. power demand); and
- trailer transport (a high speed operation with a high drawbar power requirement).

Mouldboard ploughing and power harrowing operations were undertaken on two soil types, a sandy loam and a clay soil, to determine the effect soil characteristics could have upon the tractor driveline loads. Operation-dependent parameters were recorded simultaneously through tractor-implement mounted instrumentation (see Section 5.2). To minimise driving style variation the same individual operated the test vehicle at all times.

## 5.2 *Instrumentation*

### 5.2.1 Tractor-Based Transducers & Indicators

The tractor CAN-bus and diagnostic plug (see Section 3.1) facilitated the acquisition of a number of tractor-based parameters. The process was simplified as the structure of most of the desired CAN-bus messages adhered to the J1939-71 standard format, the recommended practice for surface vehicles (Society of Automotive Engineers, 2002). In addition to the standard messages, the electronic control modules transmitted a number of vehicle specific messages including those relating to engine power boost, which were also acquired. Minor modifications to the vehicle software were necessary to increase the transmission rate of some messages to provide adequate data update rates. The CAN-bus messages acquired during each operation are outlined in Table 5.1. Table 5.2 details the source of each parameter message, the output format to the CAN-bus and indicates the J1939 standard messages. Additional information on tractor-based sensors and their calibration (excluding flywheel torque) is contained in Appendix A3.

*Table 5.1 – Tractor-based (CAN-bus) parameters acquired during each operation*

| No | Parameter | Symbol | Ploughing | Power Harrowing | Transportation |
|---|---|---|---|---|---|
| | Time | | x | x | x |
| T1 | Engine Speed | $\omega_E$ | x | x | x |
| T2 | Flywheel Torque | $T_F$ | x | x | x |
| T3 | Engine Torque | $T_E$ | x | x | x |
| T4 | Gear | G | x | x | x |
| T5 | Transmission Output Speed | $\omega_T$ | x | x | x |
| T6 | Theoretical Forward Speed | $v_t$ | x | x | x |
| T7 | True Forward Speed | $v_a$ | x | x | x |
| T8 | Wheelslip | S | x | x | |
| T9 | Throttle Position | $\varepsilon_t$ | | | x |
| T10 | Rockshaft Position | $\varepsilon_r$ | x | x | |
| T11 | Boost Percent | $B_{\%}$ | | x | |
| T12 | Boost Status | $B_S$ | | x | |
| T13 | Vehicle Torque Demand | $T_{DV}$ | | x | |
| T14 | P.T.O. Torque (Vehicle) | $T_{PTO}$ | | x | |
| T15 | Engine Speed Droop | $\delta_l$ | | x | x |

*Table 5.2 – Tractor-based parameter details*

| No | Parameter | Data Source | | Output | J1939 Message |
|---|---|---|---|---|---|
| | | Transducer Type | Transducer Location | | |
| T1 | Engine Speed | variable reluctance | crankshaft tone wheel | rpm | Yes |
| T2 | Flywheel Torque | Flywheel Dampter & Hall-effect | flywheel | % | No |
| T3 | Engine Torque | calculated (engine module) | | % | Yes |
| T4 | Gear | calculated from (1) and (5) | | integer | Yes |
| T5 | Transmission Output Speed | Hall-effect | transmission output shaft | rpm | Yes |
| T6 | Theoretical Forward Speed | calculated from (5) | | km/h | No |
| T7 | True Forward Speed | radar | under vehicle | km/h | Yes |
| T8 | Wheelslip | calculated from (6 & 7) | | % | No |
| T9 | Foot Throttle Position | potentiometer | foot throttle assembly | % | Yes |
| T10 | Rockshaft Position | potentiometer | 3pt hitch rockshaft | % | No |
| T11 | Boost Percent | calculated from (1, 2 and 14) | | % | No |
| T12 | Boost Status | calculated from (1, 2 and 14) | | Integer | No |
| T13 | Vehicle Torque Demand | calculated from (1), (3) and (11) | | % | Yes |
| T14 | P.T.O. Torque (Vehicle) | 2 x Hall-effect & PTO Shaft | flywheel damper & transmission output shaft | Nm | No |
| T15 | Engine Speed Droop | calculated from (7) | | integer | No |

### 5.2.2 Implement Transducers

In addition to the tractor-based information, a number of implement parameters were recorded depending on the operation being undertaken (see Table 5.3).

*Table 5.3 – Acquired implement parameters for each operation*

| No | Parameter | Symbol | Ploughing | Power Harrow | Transportation |
|---|---|---|---|---|---|
| I1 | Draught Force (Left, Right, Top) | $H_L$, $H_R$, $H_T$ | x | x | |
| I2 | Vertical Force (Left, Right) | $V_L$, $V_R$ | x | x | |
| I3, I5 | Working Depth | $D_P$, $D_H$ | x | x | |
| I4 | Working Width | $W_P$ | x | | |
| I6 | P.T.O. Torque (External) | $T_{BH}$ | | x | |

Table 5.4 details the source of each parameter and the calibrated format of the recorded data. All external sensors were calibrated. Further information and calibration results are presented in Appendix A3

*Table 5.4 – Implement acquired parameter details*

| No | Parameter | Data Source | | Output |
|---|---|---|---|---|
| | | Transducer Type | Transducer Location | |
| I1 | Draught Force (L,R,T) | Strain Gauge | Scholtz Linkage | kN |
| I2 | Vertical Forces (L,R) | Strain Gauge | Scholtz Linkage | kN |
| I3 | Plough Depth | Linear Potentiometer | Depth Skid | mm |
| I4 | Plough Width | LVDT | Plough Frame | mm |
| I5 | Power Harrow Depth | Rotary Potentiometer | Depth Adjustment Hyd.Cylinder | mm |
| I6 | P.T.O. Torque | Strain Gauge (Shaft) | British Hovercraft Meter | Nm |

#### *5.2.2.1 Implement Forces*

Draught and vertical forces, generated by soil-engaging implements, were measured using a heavy-duty 3pt. linkage dynamometer, based on a design originally proposed by Scholtz (1966), which fitted between the tractor and implement (see Figure 5-1). This design enabled the measurement of both horizontal and vertical forces from the implement. The sign convention is shown in Figure 5-2.

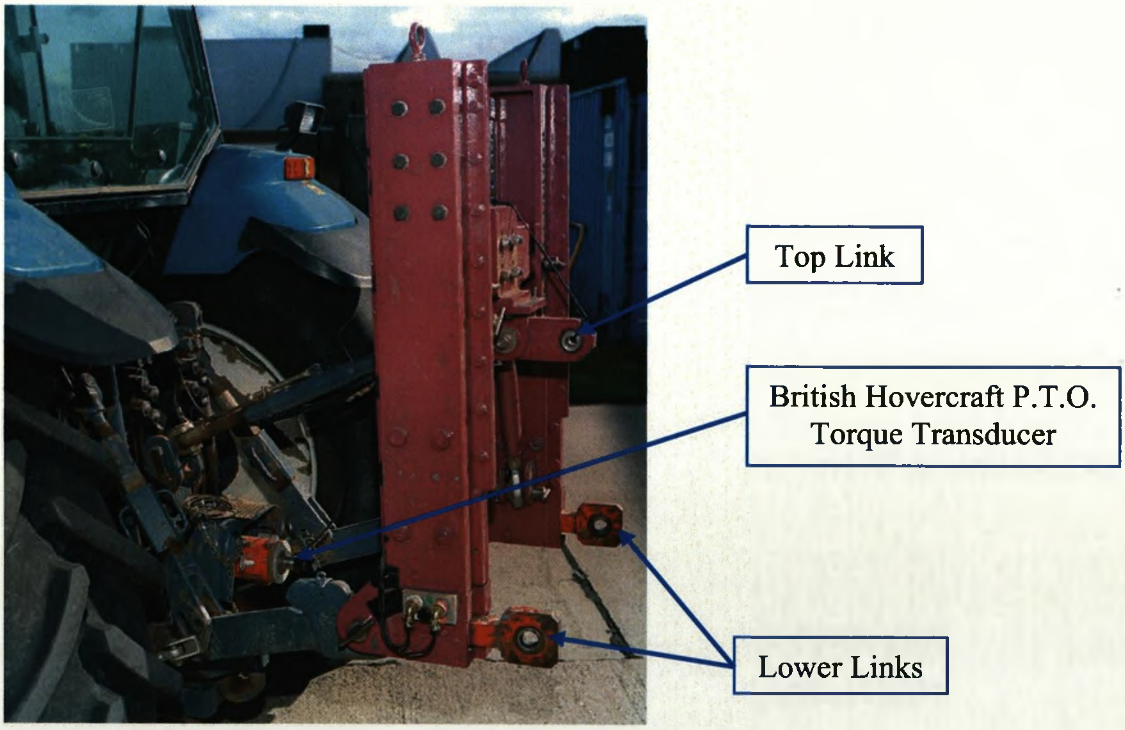



*Figure 5-1 – Scholtz linkage dynamometer & British Hovercraft torque transducer*

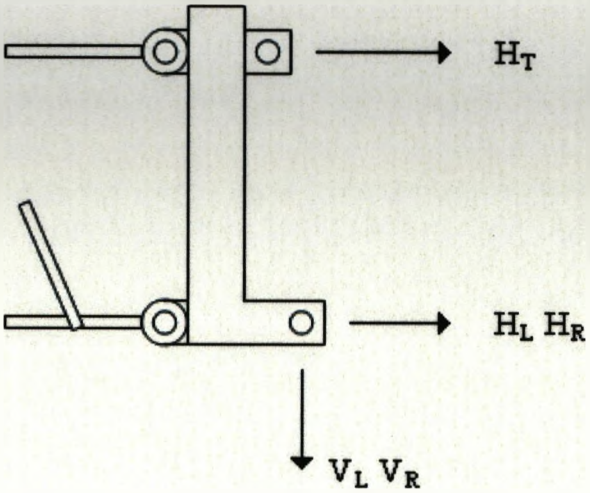



*Figure 5-2 – Force conventions used for Scholtz linkage dynamometer*

| | | | |
|---|---|---|---|
| Where: | $H_T$ | = | Top link horizontal force |
| | $H_L$ | = | Lower left link horizontal force |
| | $H_R$ | = | Lower right link horizontal force |
| | $V_L$ | = | Lower left link vertical force |
| | $V_R$ | = | Lower right link vertical force |

### *5.2.2.2 Plough Working Depth & Width*

A spring-loaded depth skid, used to monitor plough working depth ($D_P$), was mounted to the plough beam to run along the unploughed surface in front of the centre mouldboard (see Figure 5-3). The movement of the skid, relative to the beam, was the basis for depth measurement. Plough furrow width ($W_P$) was monitored with a Linear Voltage Differential Transformer (LVDT) mounted across the plough working width adjustment mechanism (see Figure 5-3).

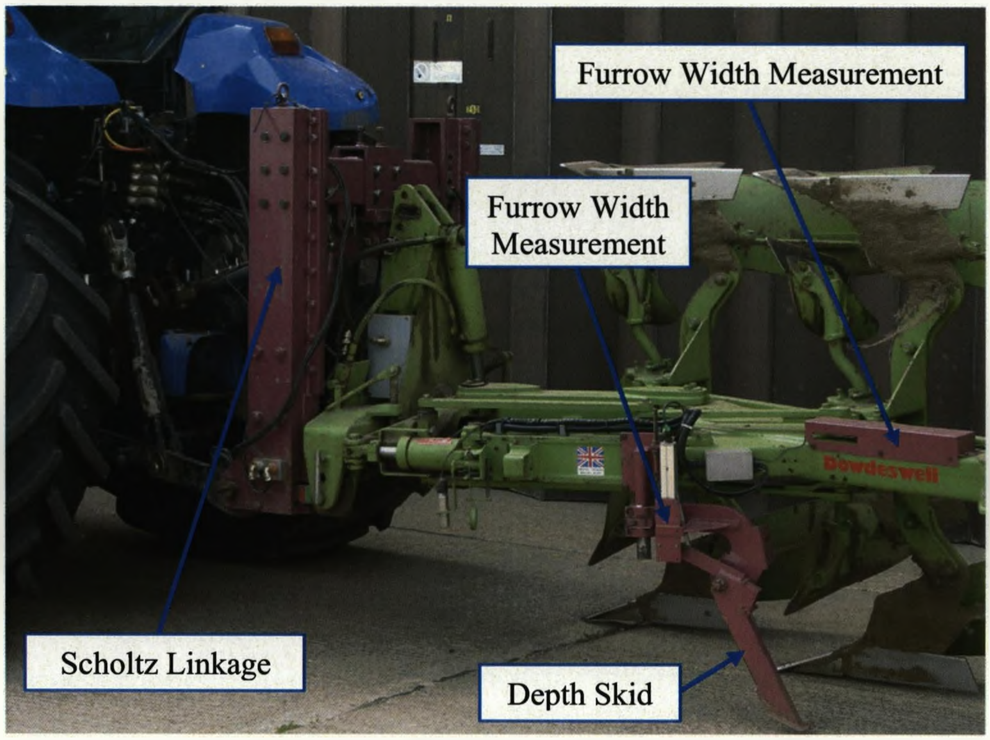



*Figure 5-3 – Plough depth and width measuring methods*

### *5.2.2.3 P.T.O. Torque*

The dynamic performance of the test tractor's internal P.T.O. torque measurement system was deemed to be questionable following preliminary experiments. Therefore an external 'British Hovercraft' P.T.O. torque transducer was fitted between the tractor and the implement to give a more reliable assessment of implement P.T.O. torque ($T_{BH}$) (see Figure 5-1). The value of $T_{BH}$ is referenced at the P.T.O. The 'equivalent' torque seen at the flywheel reduced by the flywheel : P.T.O. speed ratio of 2.12:1 ($r_{pe}$).

### *5.2.2.4 Power Harrow Working Depth*

Power harrow depth was measured using a pair of drawstring actuated, spring-loaded rotary potentiometers. The potentiometers determined extension of the hydraulic cylinders, mounted at each side of the implement, which controlled the working depth of the power harrow by varying the vertical position of the rear consolidation roller ($D_{HL}$, $D_{HR}$), (see Figure 5-4).

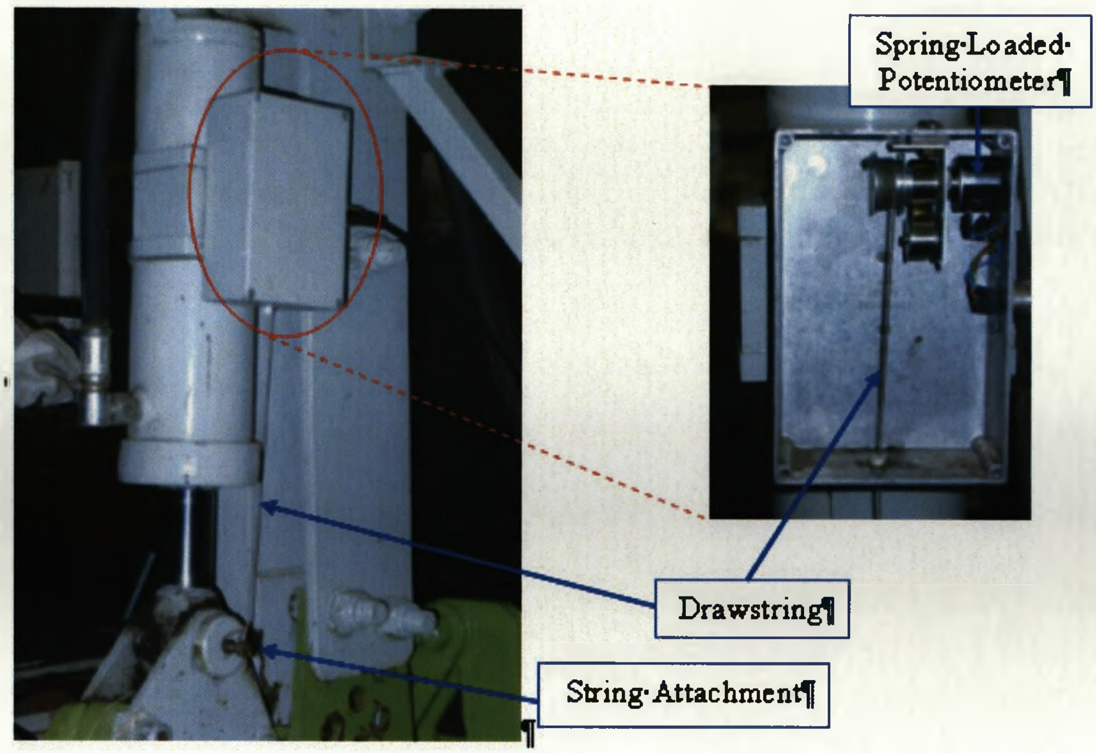



*Figure 5-4 – Power harrow working depth measurement*

### 5.2.3 Data Acquisition Equipment & Processing

The data acquisition system (see Figure 5-5) was based on a Gridcase 1520 ruggedised lap-top computer. The internal input/output (I/O) bus was connected into an interface unit beneath the computer.

The interface unit was equipped with:

1. DAS-8 analogue-digital converter (ADC) card converting analogue inputs to digital form at 12 bit resolution. This card was connected to two EXP-16 multiplex cards allowing a total of 32 analogue inputs to be accommodated; and
2. CAN-bus interface card, based on an Intel 82527 chip: this card allowed the data acquisition system to acquire CAN-bus messages via the tractor diagnostic plug.

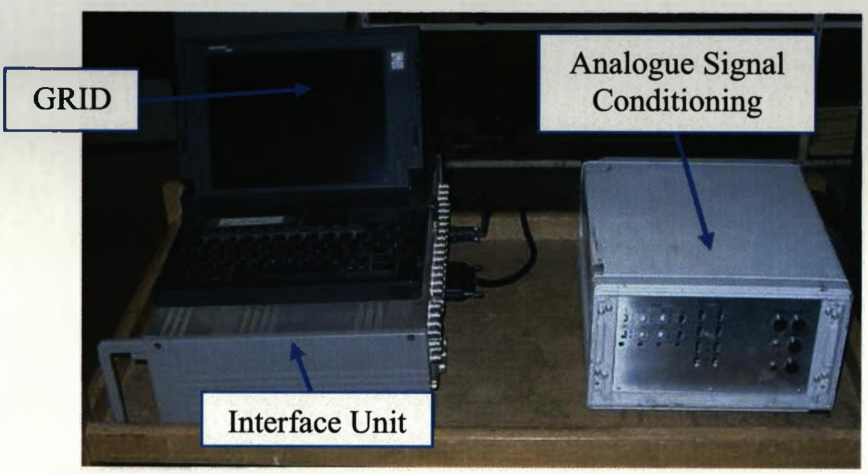



*Figure 5-5 – Data acquisition equipment*

An in-house designed signal conditioning unit, which contained the signal conditioning circuits for all the analogue transducers, was used. The unit was used to power the transducers as well as ensure all outputs were ±5V DC for subsequent use conversion by the ADC in the interface unit. The interface unit also allowed gain and offset calibration adjustments to be made and low-pass filtering to be implemented.

The computer ran bespoke software allowing a number of configuration files to be stored. A file was then selected to configure the system for the data acquisition task to be performed. The file implemented instructions as to which analogue and CAN-bus messages should be sampled, any calibration calculations to be implemented, the sampling rate and the number of samples per stored data point. The output file was in standard comma-separated-variable text format. A bespoke processing programme was executed on each output file to allow the calculation of additional derived parameters (e.g. flywheel power) prior to data analysis. These derived parameters have been defined in Appendix A3.21 to A3.23. Each type of field experiment had a unique processing file to meet the requirements of that part of the work. Typical data acquisition rates of 10Hz and 50Hz were used for steady state and transient tests (see individual sections for more details). Kocker and Summers (1985) determined that the natural frequency of draught force signals was 2Hz, therefore these acquisition rates were sufficiently high to avoid aliasing errors by ensuring sampling was undertaken at more than twice the signal frequency, as defined by Shannon (1949). The CAN-bus parameters were subject to sampling and averaging rates within each vehicle microprocessor controller.

## 5.3 *Ploughing - Steady State*

### 5.3.1 Objective

The objective of this part of the experimental programme was to determine the influence of different tractor-implement parameters on powertrain loading, whilst mouldboard ploughing in clay and sandy loam soil, and to determine the load variation experienced due to soil changes across the field.

### 5.3.2 Experimental Equipment

The test tractor was operated with a Dowdeswell Delta Furrow 100HA five furrow semi-mounted reversible plough (see Figure 5-6), an adequate size to impose a sufficient draught on the test tractor. The plough had the provision to change the draught requirement by varying the furrow width between 305mm and 457mm (12" to 18") hydraulically (see Appendix A1.2). This plough was used for the steady state and the subsequent transient ploughing experiments (see Section 5.4).

*Figure 5-6 – Test tractor and mouldboard plough during experimental trials*

### 5.3.3 Experimental Design & Procedure

The experiments were undertaken on a 'light' sandy loam Cottenham series soil and a 'heavy' clay Evesham (formally Wicken) series soil (King, 1969), these being representative of two different operating conditions likely to be encountered in practical farm work. The original intention was to investigate three furrow width settings of 305mm, 356mm and 406mm and three transmission gear ratios reflecting the range of forward speeds used in practice. However, the difference in strength and consequently powertrain loading between the two soils resulted in a varied range of transmission gear ratios being used and it was not possible to use the widest furrow width setting and highest chosen forward speed in the clay soil. Nonetheless, the range of gear ratios and furrow widths used (see Table 5.5) allowed a wide variation in powertrain load levels to be investigated, whilst allowing some overlap in forward speed between the two soil types. A mean constant loaded engine speed (2200rpm) and a ploughing depth of 225mm were maintained where possible and three replicates of each configuration were performed in each soil. All ploughing was undertaken in one direction across the field to remove any potential variation between the left and right sides of the plough. Parameters relating to each tractor-implement configuration were recorded for a period of 120 seconds in the middle of a pass across the field, with data sampled at 100Hz and subsequently averaged to 10Hz prior to recording to avoid aliasing errors.

*Table 5.5 – Tractor-implement configurations used for steady state ploughing experiments*

| Soil Type | Forward speed (km/h) | Gear | Furrow width (mm) | Total runs |
|---|---|---|---|---|
| Clay | 4.3, 5.6, 6.8 and 8.3 | 4, 5, 6 and 7 | 305, 356 and 406* | 33 |
| Sandy loam | 5.6, 8.3 and 10.2 | 5, 7 and 8 | 305, 356 and 406 | 27 |
| | * not in gear 7 | | | |

### 5.3.4 Parameters

The acquired tractor-implement parameters for this part of the investigation are outlined in Tables 5.1 and 5.3. In addition, a number of secondary parameters including flywheel power, drawbar power, tractive efficiency, theoretical ploughed area and flywheel draught torque (the component of flywheel torque due to implement draught) were calculated for each time point during data processing (see Appendix A3.21).

### 5.3.5 Results

Absolute mean values (both within each trial and across the replications) of the key data for each tractor-implement configuration are presented in Table 5.6 and additional mean and standard deviation data for all parameters in each experimental run are presented in Appendix A4.1.

Figures 5-7 and 5-8 present combined engine torque-speed scatter data arising from all transmission ratios used in either sandy loam or clay soil types respectively. These are for a 356mm furrow width setting. Additional distributions for 305mm and 406mm furrow widths are presented in Appendix A4.1. Two datasets are also presented as time histories for gear 7 at a furrow width of 356mm, in both sandy loam soil and clay soil (see Figures 5-9 and 5-10 respectively).

*Table 5.6 – Ploughing (steady state) results summary*

| | Draught Force $H_A$ (kN) | | Flywheel Torque $T_F$ (Nm) | | Engine Speed $\omega_E$ (rpm) | | True Speed $v_B$ (km/h) | | Flywheel Power $P_F$ (kW) | | Slip S (%) | | Workrate $A_{PT}$ (ha/hour) | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Gear | Sandy | Clay | Sandy | Clay | Sandy | Clay | Sandy | Clay | Sandy | Clay | Sandy | Clay | Sandy | Clay |
| **305mm Furrow Width** | | | | | | | | | | | | | | |
| 4 | | 30.7 | | 207 | | 2209 | | 3.6 | | 47.9 | | 12.7 | | 0.55 |
| 5 | 16.5 | 30.8 | 175 | 275 | 2195 | 2190 | 5.1 | 4.6 | 40.1 | 62.9 | 3.9 | 12.7 | 0.78 | 0.70 |
| 6 | | 33.9 | | 359 | | 2176 | | 5.5 | | 81.6 | | 14.5 | | 0.84 |
| 7 | 17.9 | 32.2 | 291 | 427 | 2197 | 2074 | 7.6 | 6.5 | 67.0 | 92.4 | 4.5 | 12.9 | 1.16 | 0.99 |
| 8 | 20.9 | | 396 | | 2197 | | 9.2 | | 90.9 | | 5.7 | | 1.40 | |
| **356mm Furrow Width** | | | | | | | | | | | | | | |
| 4 | | 32.5 | | 223 | | 2193 | | 3.6 | | 51.2 | | 14.1 | | 0.62 |
| 5 | 19.8 | 38.4 | 192 | 334 | 2198 | 2145 | 5.0 | 4.2 | 44.1 | 74.9 | 5.8 | 19.4 | 0.89 | 0.73 |
| 6 | | 40.3 | | 413 | | 2073 | | 4.8 | | 89.2 | | 22.6 | | 0.83 |
| 7 | 20.8 | 36.3 | 324 | 471 | 2189 | 1914 | 7.4 | 5.8 | 74.4 | 93.6 | 6.5 | 16.3 | 1.31 | 1.01 |
| 8 | 23.6 | | 434 | | 2098 | | 8.5 | | 95.1 | | 8.3 | | 1.51 | |
| **406mm Furrow Width** | | | | | | | | | | | | | | |
| 4 | | 35.9 | | 259 | | 2217 | | 3.3 | | 59.9 | | 21.7 | | 0.66 |
| 5 | 19.7 | 34.3 | 201 | 312 | 2201 | 2196 | 5.0 | 4.3 | 46.4 | 71.6 | 6.7 | 18.9 | 1.01 | 0.87 |
| 6 | | 37.5 | | 411 | 2195 | 2105 | | 4.7 | | 90.2 | | 25.6 | | 0.94 |
| 7 | 22.7 | | 348 | | 2195 | | 7.3 | | 80.0 | | 7.9 | | 1.48 | |
| 8 | 25.0 | | 456 | | 2042 | | 8.2 | | 97.3 | | 9.8 | | 1.66 | |

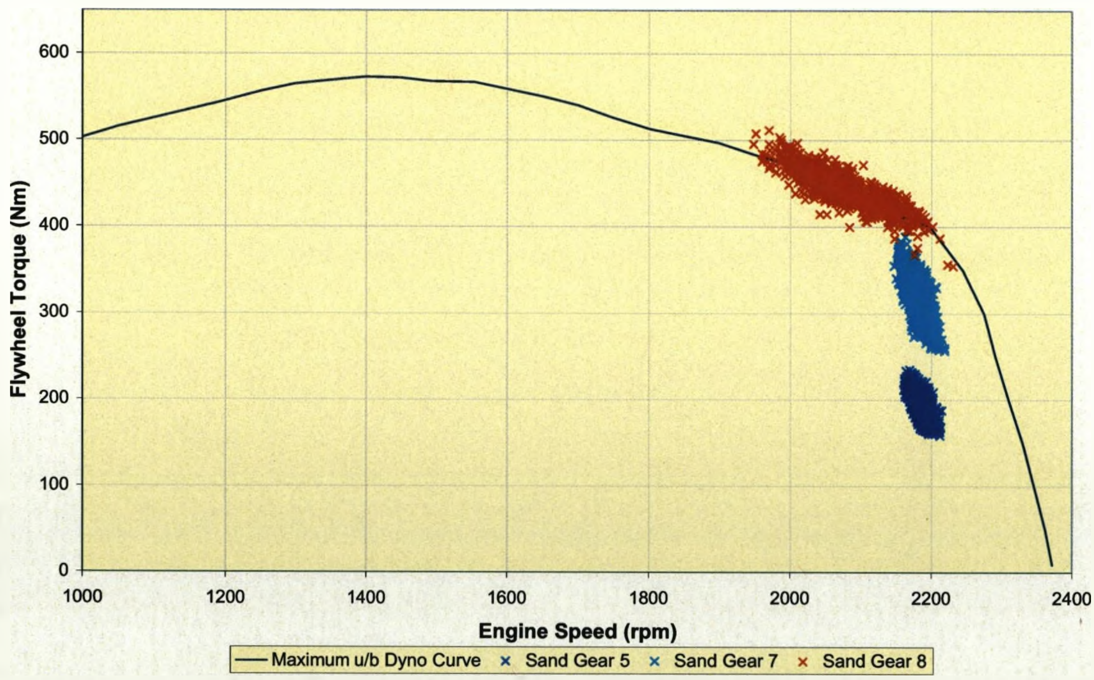



*Figure 5-7 – The effect of gear selection on dynamic loading whilst ploughing sandy loam soil (356mm furrow width)*

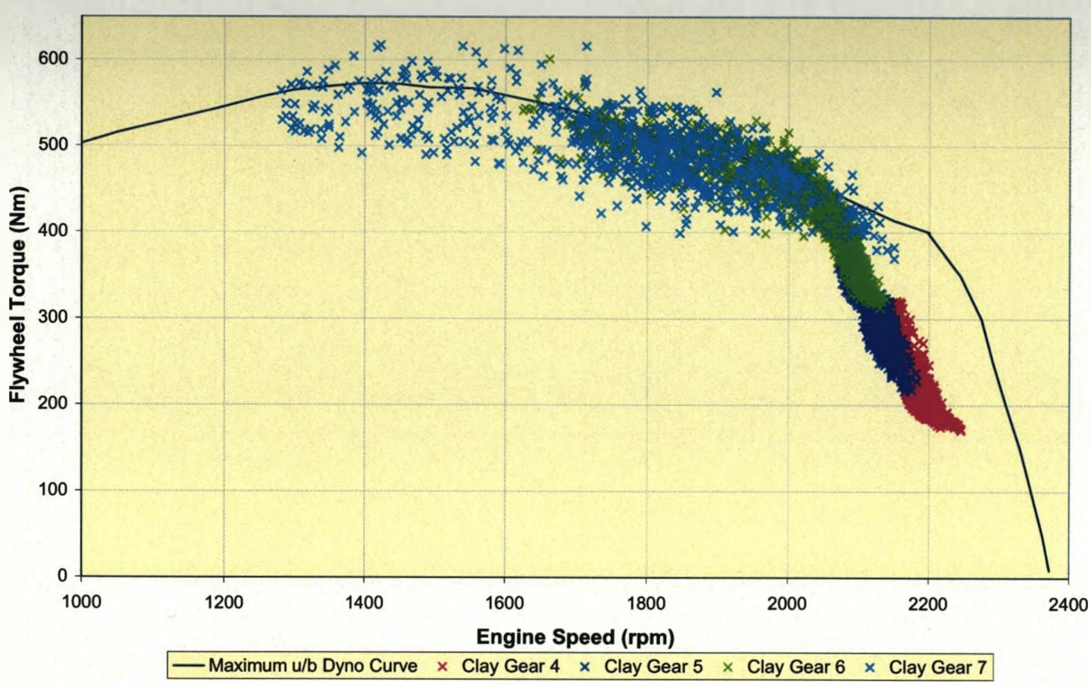



*Figure 5-8 – The effect of gear selection on dynamic loading whilst ploughing clay soil (356mm furrow width)*

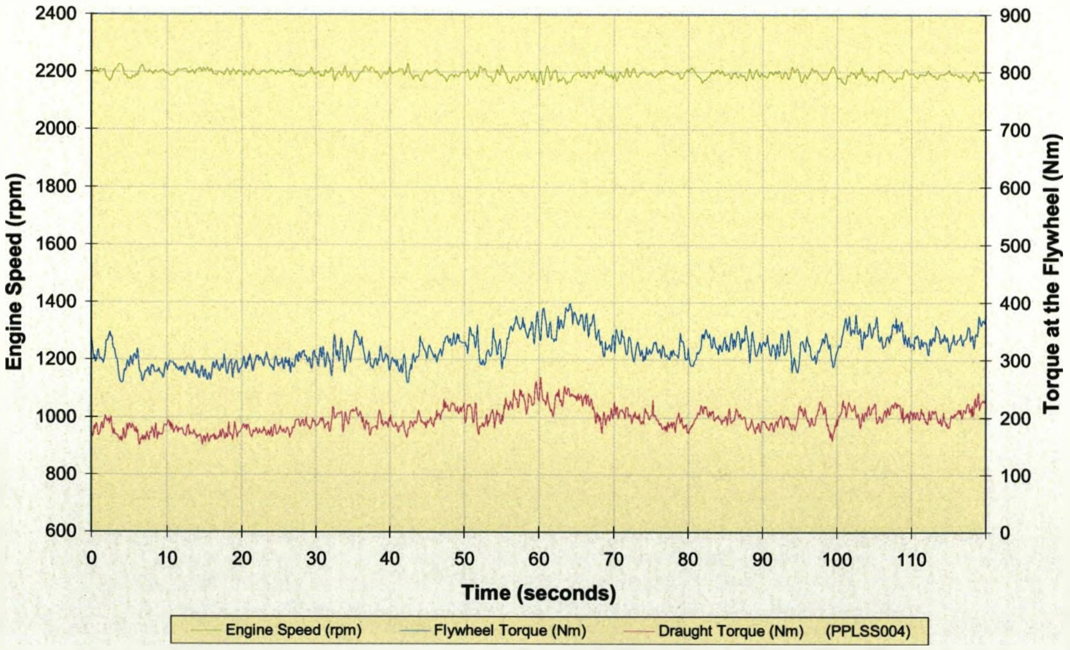



*Figure 5-9 – Example test data time history - sandy loam soil (furrow width 356mm, gear 7)*

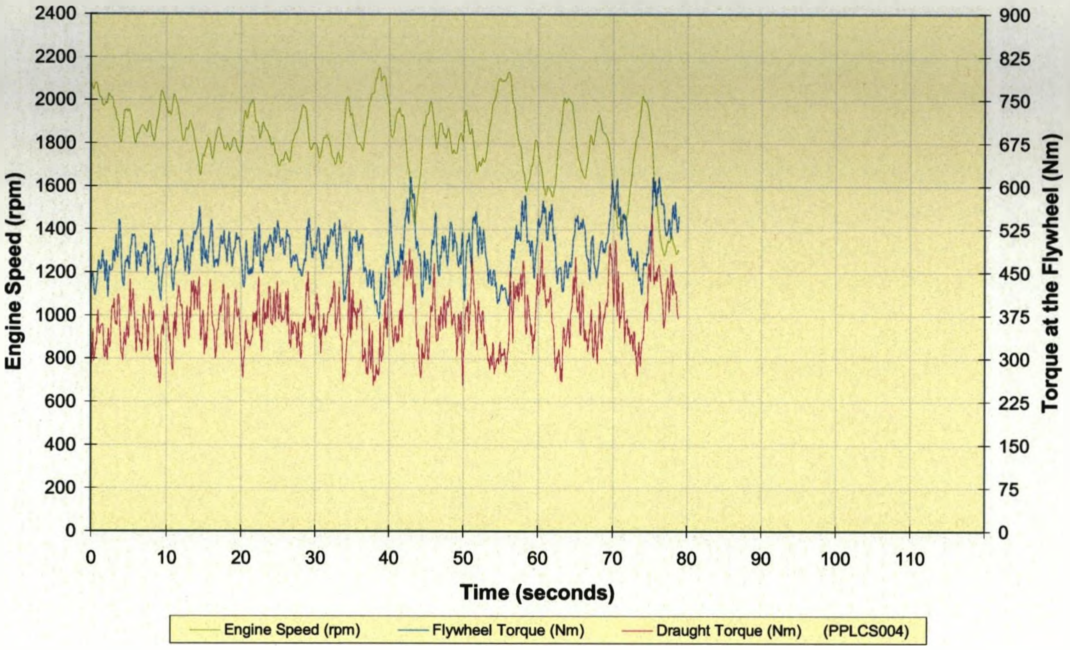



*Figure 5-10 – Example test data time history - clay soil (furrow width 356mm, gear 7)*

### 5.3.6 Discussion

The structurally weaker sandy loam soil imposed lower draught forces upon the tractor and consequently placed a lower power demand on the tractor powertrain compared to the same tractor-implement configuration in clay soil (see Table 5.6, Figure 5-7 and Figure 5-8). Sandy loam soil draught requirement was significantly lower (52-57%) than in clay soil, irrespective of transmission gear ratio or furrow width. The clay soil draught translated to a proportionally higher drawbar pull and flywheel torque requirement from the test tractor. Wheelslip, a function of drawbar pull, was also significantly higher (more than double) than in the sandy loam soil. The higher clay soil wheelslip reduced true forward speed and resulted in a lower theoretical workrate with the same tractor-implement configuration. The theoretical workrate takes no account of headland turns or other time losses and is purely for comparative purposes. Witney (1988) suggests a field efficiency of 75%-85% for mouldboard ploughing to give a true rate of work, but cites many factors including turning techniques, field size and shape, and fieldwork patterns as affecting this rate.

Within the same soil type, the effect of the different tractor-implement operational parameters was less clear due to the interactions between them. By increasing the transmission gear ratio, and therefore forward speed, draught load and flywheel torque were increased (see Figure 5-7 or Figure 5-8). The higher draught, and therefore drawbar pull requirement, meant slip increased with transmission gear ratio. These trends did not hold true for gear 7 in clay soil where, despite an increase in true forward speed, the mean draught reduced from the previous gear. This tractor-implement configuration was at, and slightly beyond, the capabilities of the test tractor in the clay soil. Flywheel torque demand exceeded the region under governor control, and as such the engine was operating on the full-load curve. The result was that mean engine speed was reduced significantly from the desired setting (1914rpm with a 356mm furrow width), and perhaps more importantly operated in an erratic manner, demonstrated by the large spread of torque-speed data (see Figure 5-8).

In Figure 5-8 a small number of points were significantly above the measured dynamometer unboosted curve. Data would not be expected to exclusively fall below this curve as it was developed from the mean of 30 seconds of data at each point under steady state, controlled test cell conditions. The field data was collected under dynamic conditions at 10Hz and was subject to variations in atmospheric conditions. In addition, there are 3600 data points for gear 7 shown, with no averaging. Those falling outside ±5% of the dynamometer curve would be few in number and were generated during the period of erratic engine operation. As no calibration was performed in this range, it is difficult to state the performance of the engine, or the torque transducer, these points are therefore shown to highlight the extreme engine operation experienced. Indeed the configuration was not sustainable as operator intervention to halt the test was required to prevent the engine stalling.

In sandy loam soil increasing furrow width correspondingly increased implement draught requirement and therefore resulted in a higher flywheel torque demand. Higher levels of wheelslip, from the higher draught force, resulted in a reduced forward speed. This working width increase resulted in a larger theoretical workrate despite the forward speed reduction. In clay soil the effect of furrow width on vehicle performance was not as clear. Increasing furrow width from 305mm to 356mm followed the trends described above, as it did for gear 4 with the widest furrow width. For gears 5 and 6, at this width, draught actually reduced whilst workrate increased. The reason for this was the extremely dry summer of 2003 which resulted in the clay soil fracturing on impact with the plough share (meaning not all the clay soil material made contact with the mouldboard and therefore limited the increase in draught with a wider plough width) whereas increasing width in the sandy soil increased the volume of material 'flowing' over the mouldboard and therefore the draught force.

Example dynamic variation in powertrain loading during an experiment in sandy loam soil (see Figure 5-9) shows the changes in flywheel torque can be attributed to implement draught variation. The correlation coefficient between draught and torque was 0.76. This coefficient increased to 0.87 if a 200ms time delay to the torque signal was introduced to account for time-lags between the two measurement points. Draught force was measured at the hitch, whereas torque was measured at the flywheel. Compliance in both the tyres and traction driveline would result in a time delay between the two measurement points. The resultant engine speed fluctuations were small, with a standard deviation of 12rpm around a mean of 2194rpm. In contrast, the equivalent tractor-implement configuration in clay soil (see Figure 5-10) required the operator to intervene to prevent the tractor stalling. There was a great deal of variation in draught and flywheel torque levels, resulting in severe engine speed fluctuations (s.d. 171rpm). This was accentuated by the high engine loading levels, causing engine operation on the full-load curve, where the torque-speed characteristics are such that small changes in engine loading cause significantly greater changes in engine speed (see Figure 5-8).

## 5.4 Ploughing - Transient

### 5.4.1 Objective

The objective was to determine the test tractor powertrain response to:

1. a change in powershift gear (increase and decrease);
2. a change in working depth (increase and decrease); and
3. a change in engine speed (increase and decrease).

### 5.4.2 Experimental Design & Procedure

Each transient configuration was replicated three times at each working width, previously used and for both soil types, at a nominal working depth of 225mm (except during depth change experiments). Recording started prior to the transient and continued for sufficient time to allow the effects of the transient to dissipate, typically between 15 seconds and 30 seconds. A data sampling rate of 100Hz was averaged to 50Hz for recording. Details relating to individual transient experiments follow.

#### 5.4.2.1 Gearshift

Gearshifts were only undertaken between powershift gears across the range of ratios used in the steady state experiments, with upshifts and downshifts investigated as separate trials. Although trials were undertaken in both soils, sensor issues with the sandy loam soil trials resulted in only the clay soil experiments being analysed - the 3-4, 5-6 and 6-7 shifts, and their downshifts, being considered. The shift characteristics were related to vehicle speed and engine load rather than soil type, which merely provided more variation. Gearshift experiments commenced at a loaded engine speed of 2200rpm.

#### 5.4.2.2 Change in Working Depth

Through operator adjustment of the 3pt. hitch position, plough depth was increased or decreased during an experimental run thereby increasing or decreasing the draught load on the tractor. Tractor draught sensitivity was minimised, allowing working depth to be dictated by 3 pt. hitch rockshaft position setting. Target working depth adjustment was between 140mm and 200mm, and vice versa, in each combination of transmission gear ratio and furrow width used for the steady state experiments.

#### *5.4.2.3 Change in Engine Speed*

Engine speed was rapidly adjusted between 1700rpm and maximum (and vice versa) by an operator change in throttle position. Each combination of transmission gear ratio and furrow width used for the steady state experiments were used for each change in engine speed.

### 5.4.3 Parameters

In addition to those used in steady state work (see Section 5.3.4), the ratio of transmission to engine speed ($r_{te}$) was considered during gearshift experiments. This allowed the relationships between the request for a gearshift, the shift timing relative to the request and the shift duration to be investigated.

### 5.4.4 Results

#### *5.4.4.1 Gearshift*

A summary of the key engine speed and torque data during the different gearshifts and different plough widths is presented in Table 5.7. Graphical representations are presented for the 5-6 and 6-5 gearshifts in Figures 5-11 and 5-12 respectively. Gearshifts are not presented for the 3-4 or 4-3 shifts, as the profile is very similar to the 5-6 and 6-5 shifts, albeit at slightly different levels of flywheel torque. A double-swap upshift (6-7) is shown in Figure 5-13. The effect of changes in loading for the same shift is shown in Figure 5-14, for which the plough furrow width was increased by 100mm. An example double-swap downshift is presented in Figure 5-15.

Table 5.7 – Key ploughing gearshift data (clay soil)

| Shift | Clutches | $W_P$ (mm) | $r_{te}$ profile | $T_F$ profile | Pre-shift $\omega_E$ (rpm) | Pre-shift $T_F$ (Nm) | Pre-shift $V_a$ (km/h) | Shift delay* (sec) | Total time** (sec) | During the Shift $T_F$ peak | During the Shift peak time*** | During the Shift $\omega_E$ drop | During the Shift $T_F$ trough | During the Shift trough time*** | During the Shift $\omega_E$ rise | Post-shift $\omega_E$ (rpm) | Post-shift $T_F$ (Nm) | Post-shift $V_a$ (km/h) |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 3-4 | | 1520 | | | 2256 | 172 | 3.14 | 0.62 | 0.88 | 323 | 0.70 | 2175 | - | - | - | 2189 | 191 | 3.62 |
| 3-4 | C4>C3 | 1725 | | | 2188 | 262 | 2.34 | 0.74 | 1.00 | 407 | 0.76 | 2126 | - | - | - | 2086 | 331 | 2.50 |
| 3-4 | | 2010 | | | 2191 | 190 | 2.73 | 0.66 | 0.90 | 361 | 0.68 | 2120 | - | - | - | 2091 | 270 | 3.05 |
| 4-3 | | 1520 | | | 2287 | 185 | 3.95 | 0.24 | 0.72 | 260 | 0.38 | 2245 | - | - | - | 2325 | 174 | 3.42 |
| 4-3 | C3>C4 | 1730 | | | 2202 | 313 | 3.02 | 0.20 | 0.68 | 356 | 0.30 | 2162 | - | - | - | 2297 | 215 | 2.90 |
| 4-3 | | 2010 | | | 2265 | 245 | 3.27 | 0.22 | 0.66 | 294 | 0.32 | 2237 | - | - | - | 2319 | 182 | 3.16 |
| 5-6 | | 1520 | | | 2146 | 238 | 4.69 | 0.68 | 1.02 | 432 | 0.80 | 2051 | - | - | - | 2124 | 321 | 5.57 |
| 5-6 | C4>C3 | 1735 | | | 2200 | 371 | 3.93 | 0.80 | 1.06 | 542 | 0.86 | 1964 | - | - | - | 1860 | 504 | 3.51 |
| 5-6 | | 2005 | | | 2191 | 335 | 4.08 | 0.70 | 0.94 | 448 | 0.76 | 2122 | - | - | - | 2106 | 419 | 4.57 |
| 6-5 | | 1520 | | | 2190 | 342 | 5.59 | 0.20 | 0.62 | 435 | 0.22 | 2154 | - | - | - | 2216 | 264 | 4.85 |
| 6-5 | C3>C4 | 1740 | | | 2120 | 420 | 5.10 | 0.24 | 0.66 | 539 | 0.28 | 2067 | - | - | - | 2167 | 325 | 4.32 |
| 6-5 | | 2005 | | | 2207 | 365 | 5.37 | 0.26 | 0.62 | 543 | 0.34 | 1988 | - | - | - | 2224 | 342 | 4.01 |
| 6-7 | | 1520 | | | 2174 | 324 | 5.73 | 0.30 | 0.68 | 602 | 0.46 | 1901 | - | - | - | 2089 | 436 | 6.49 |
| 6-7 | C2>C1 C3>C4 | 1725 | | | 2154 | 409 | 4.87 | 0.30 | 0.66 | 631 | 0.50 | 1849 | - | - | - | 1966 | 470 | 5.91 |
| 6-7 | | 2010 | | | 2151 | 385 | 5.07 | 0.26 | 0.66 | 640 | 0.44 | 1756 | - | - | - | 1778 | 471 | 5.05 |
| 7-6 | | 1520 | | | 2063 | 448 | 6.47 | 0.12 | 0.82 | 578 | 0.30 | 1922 | 117 | 0.42 | 2162 | 2196 | 331 | 5.73 |
| 7-6 | C1>C2 C4>C3 | 1740 | | | 1818 | 504 | 5.17 | 0.18 | 0.84 | 587 | 0.32 | 1727 | 347 | 0.44 | 2023 | 2114 | 429 | 4.89 |
| 7-6 | | 2005 | | | 2049 | 434 | 5.99 | 0.20 | 0.92 | 555 | 0.36 | 1920 | 211 | 0.5 | 2226 | 2146 | 398 | 5.00 |

* Time from gear selection to $\omega_{te}$ starting to change

** Time from gear selection to ωte stabilising post-shift

*** Time from gear selection to torque peak (or trough)

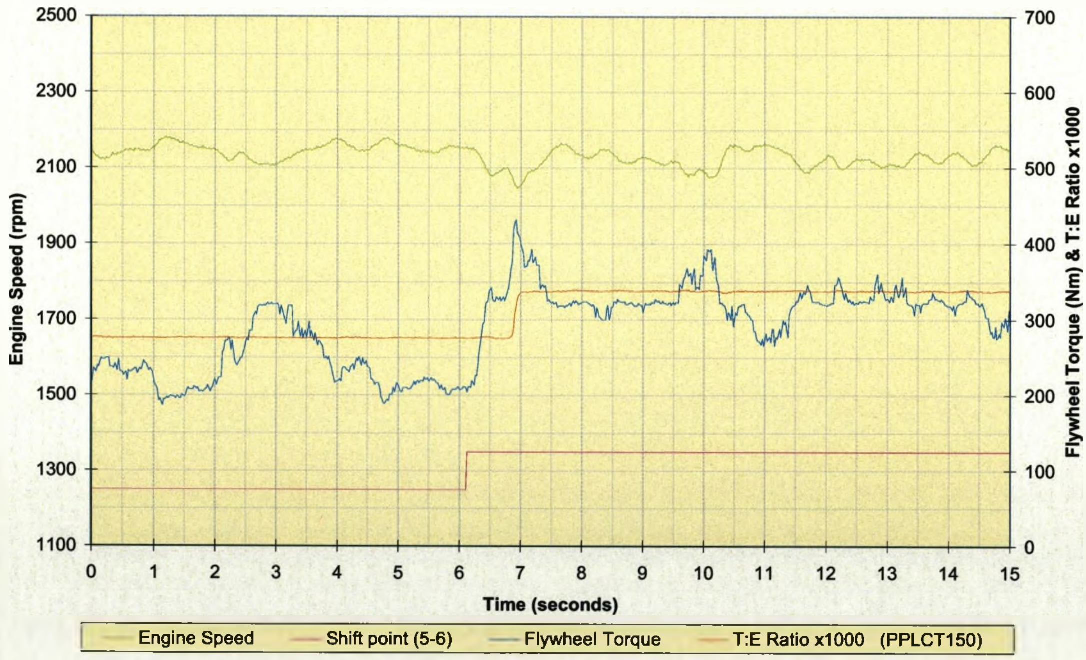



*Figure 5-11 – The effect of a 5-6 gearshift whilst ploughing clay soil (305mm furrow width)*

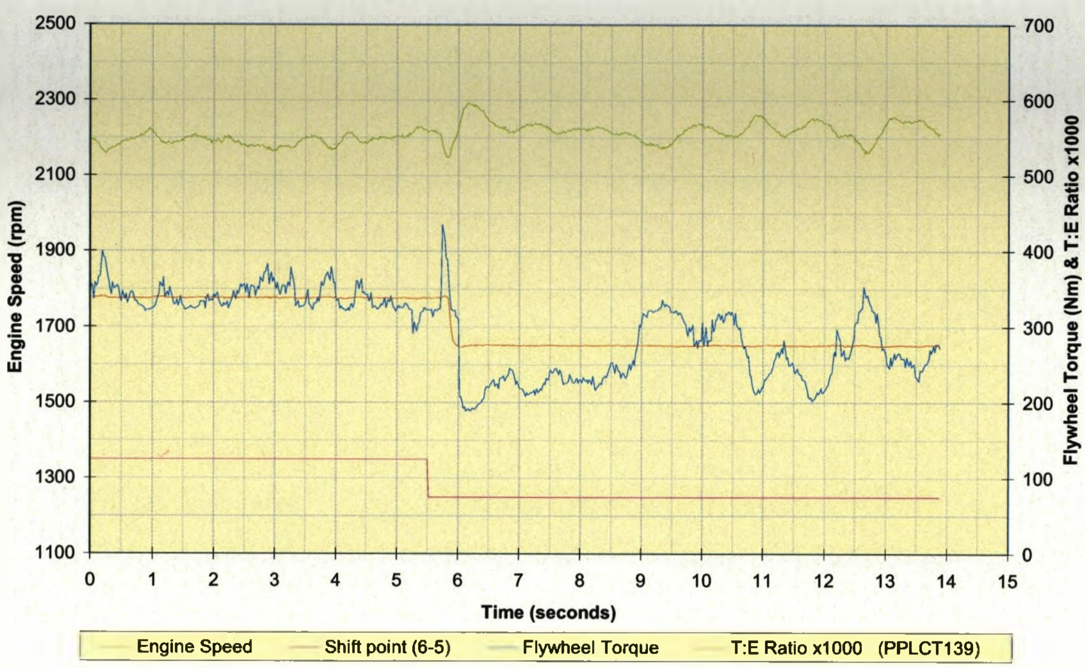



*Figure 5-12 – The effect of a 6-5 gearshift whilst ploughing clay soil (305mm furrow width)*

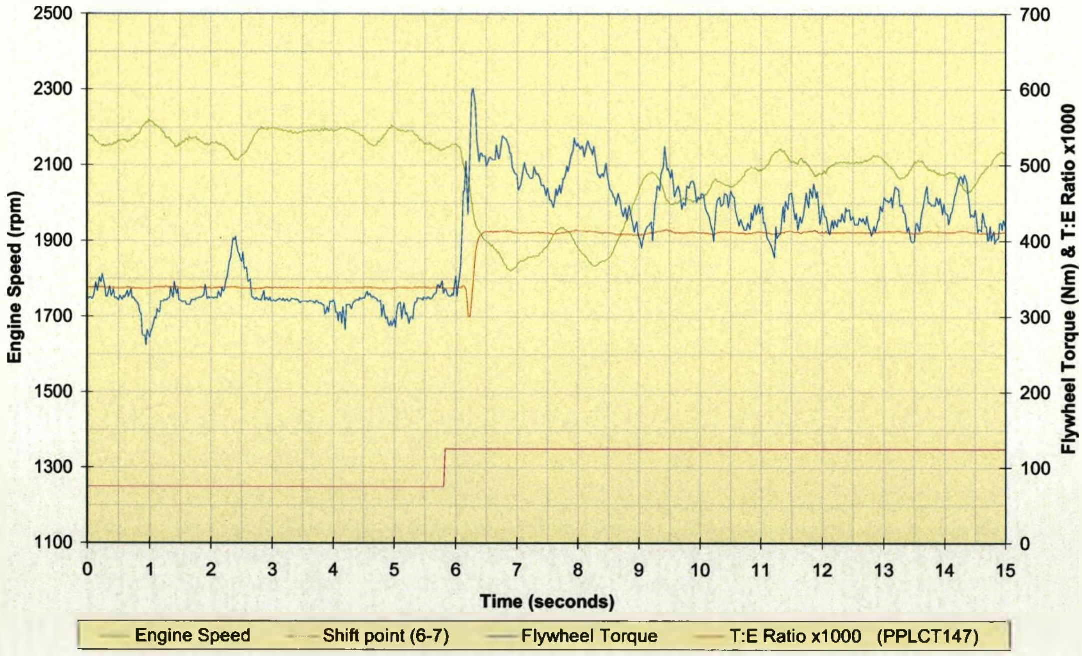



*Figure 5-13 – The effect of a 6-7 gearshift whilst ploughing clay soil (305mm furrow width)*

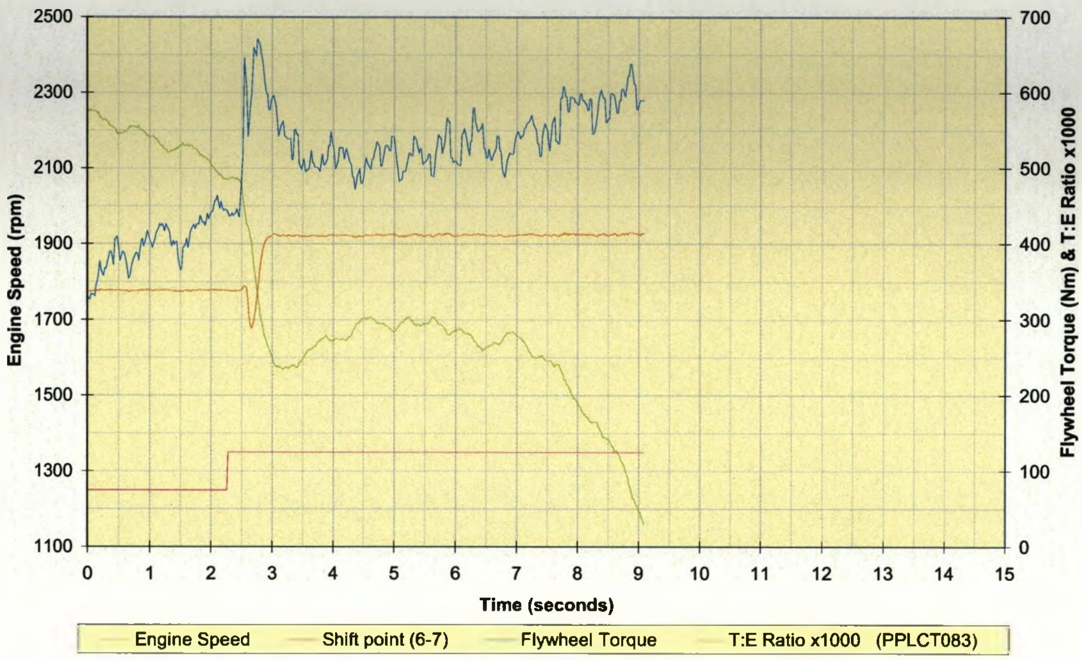



*Figure 5-14 – The effect of a 6-7 gearshift whilst ploughing clay soil (406mm furrow width)*

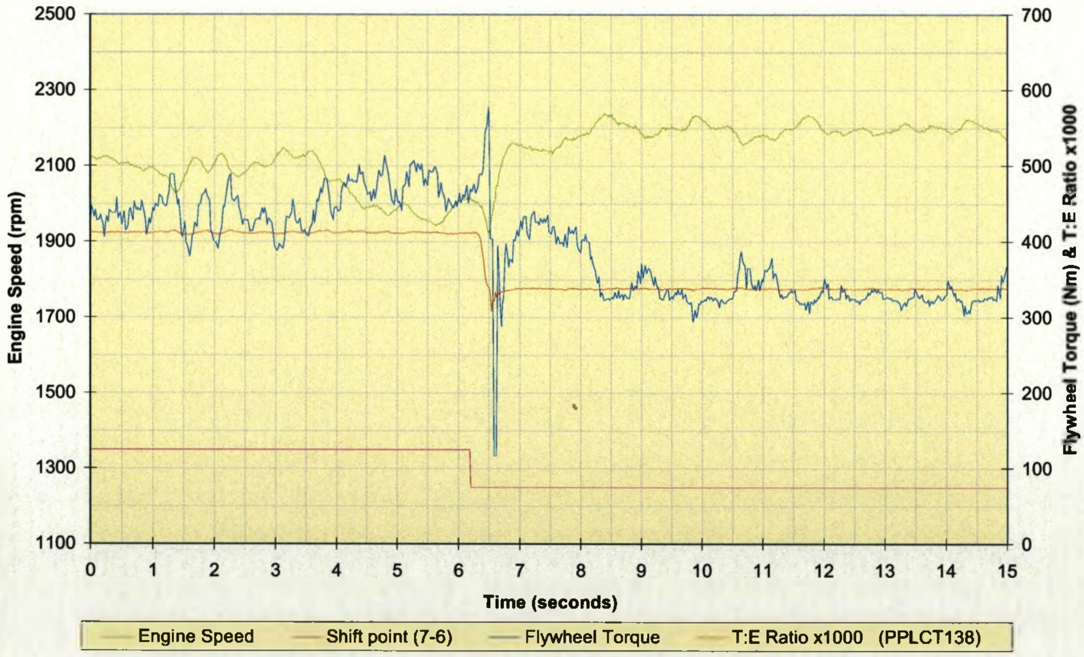



*Figure 5-15 – The effect of a 7-6 gearshift whilst ploughing clay soil (305mm furrow width)*

### 5.4.4.2 Change in Plough Working Depth

Three examples are presented showing the effect changing plough working depth has on engine speed and flywheel torque, all undertaken at a furrow width of 356mm. A summary table of key data is presented in Table 5.8 for the three datasets presented graphically. Figure 5-16 shows a depth increase in a clay soil in gear 5. The same transient is then repeated for a higher forward gear in the same soil (see Figure 5-17). The final chart is presented for the same gear (7) for the sandy soil (see Figure 5-18).

*Table 5.8 – Mean data summary for key parameters during plough depth change*

| Soil | Gear | $W_P$ (mm) | Pre-change | | | | | | | Post-change | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | $D_P$ (mm) | $\varepsilon_r$ (%) | $H_A$ (kN) | $\omega_E$ (rpm) | $T_F$ (Nm) | $v_a$ (km/h) | S (%) | $D_P$ (mm) | $\varepsilon_r$ (%) | $H_A$ (kN) | $\omega_E$ (rpm) | $T_F$ (Nm) | $v_a$ (km/h) | S (%) |
| Clay | 5 | 1755 | 124 | 28 | 17.1 | 2225 | 179 | 5.21 | 2.7 | 213 | 14 | 30.8 | 2189 | 272 | 4.60 | 12.9 |
| Clay | 7 | 1760 | 90 | 29 | 15.6 | 2194 | 267 | 7.58 | 3.9 | 194 | 15 | 39.2 | 1847* | 493 | 5.19 | 22.5 |
| Sand | 7 | 1795 | 143 | 28 | 12.2 | 2207 | 231 | 7.92 | 2.5 | 224 | 12 | 22.5 | 2185 | 352 | 7.44 | 7.5 |

* Very unstable

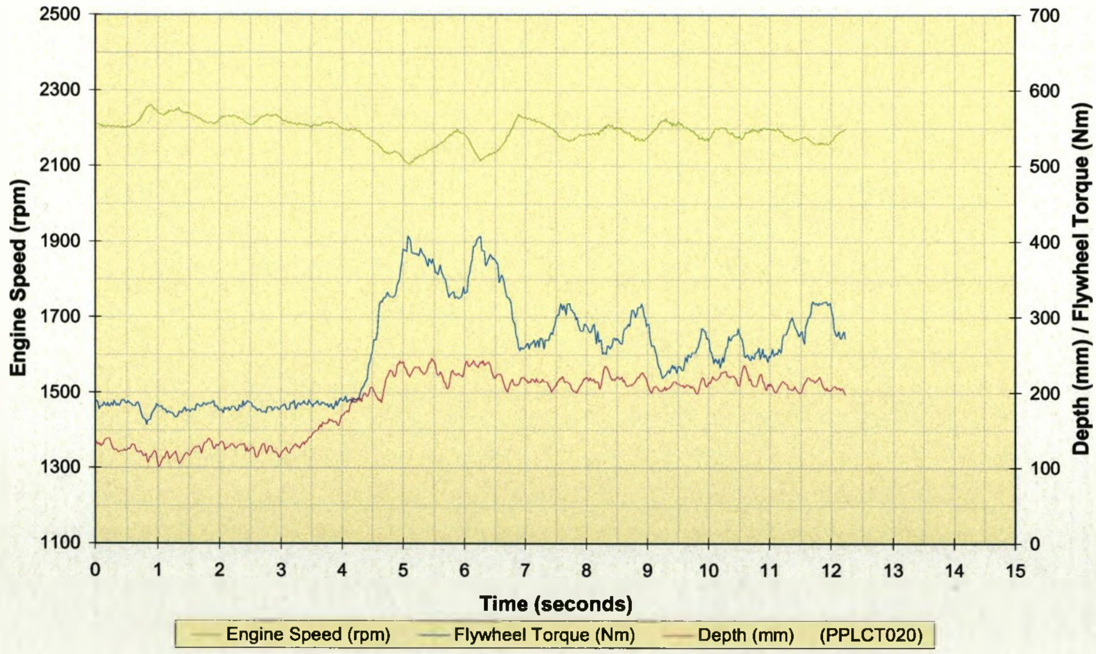



*Figure 5-16 – The effect of a change in working depth on engine speed and flywheel torque whilst ploughing clay soil (356mm furrow, gear 5)*

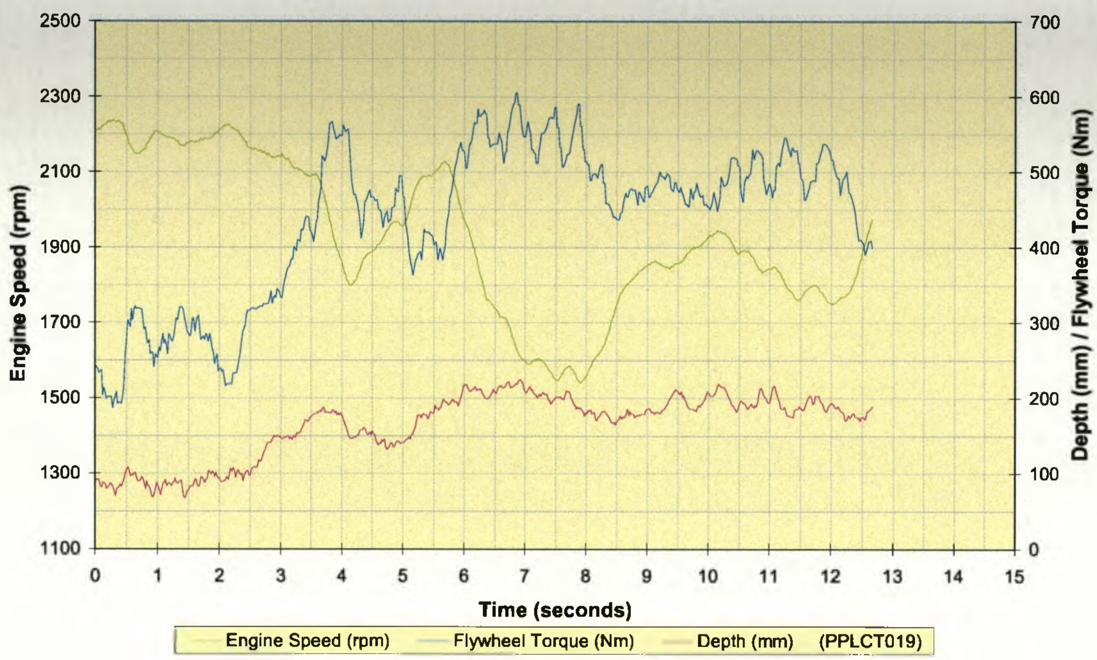



*Figure 5-17 – The effect of a change in working depth on engine speed and flywheel torque whilst ploughing clay soil (356mm furrow, gear 7)*

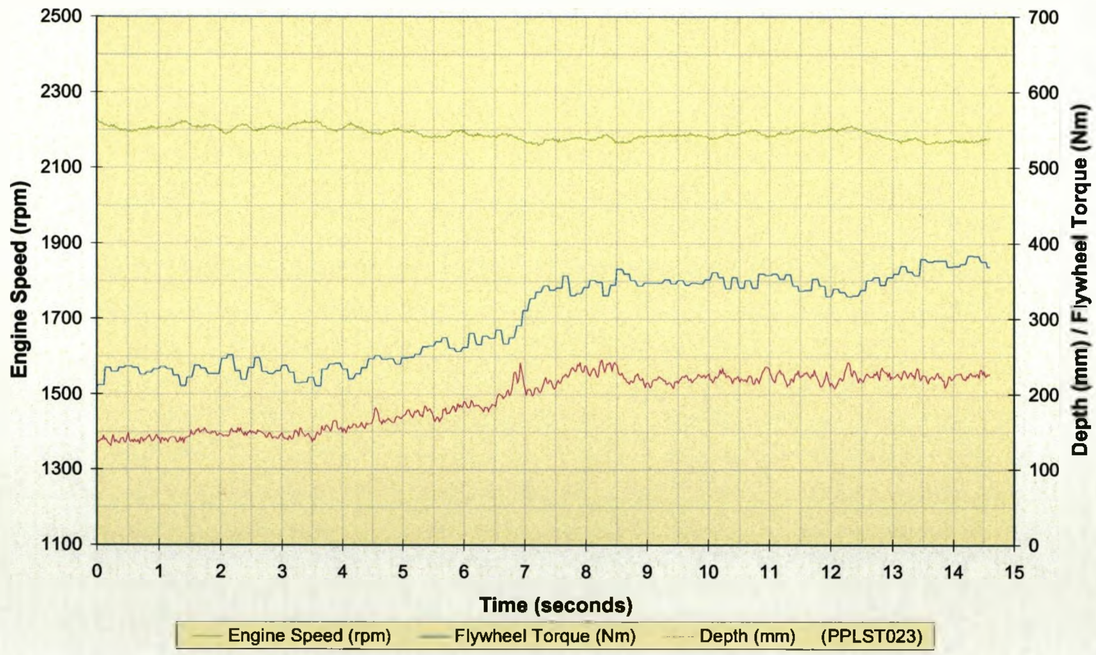



*Figure 5-18 – The effect of a change in working depth on engine speed and flywheel torque whilst ploughing sandy soil (356mm furrow, gear 7)*

### *5.4.4.3 Change in Engine Speed*

One dataset is presented for an increase in engine speed whilst ploughing clay soil (356mm furrow width, gear 5). Figure 5-19 presents the resultant change in engine speed and flywheel torque. Figure 5-20 presents the effects on true forward speed and wheelslip for the same test (engine speed shown for comparison). Similar effects were found for the sandy soil. In higher gears the effects of the change in engine speed are more difficult to identify, especially in the clay soil, due to the soil variability affecting the flywheel torque and therefore engine speed.

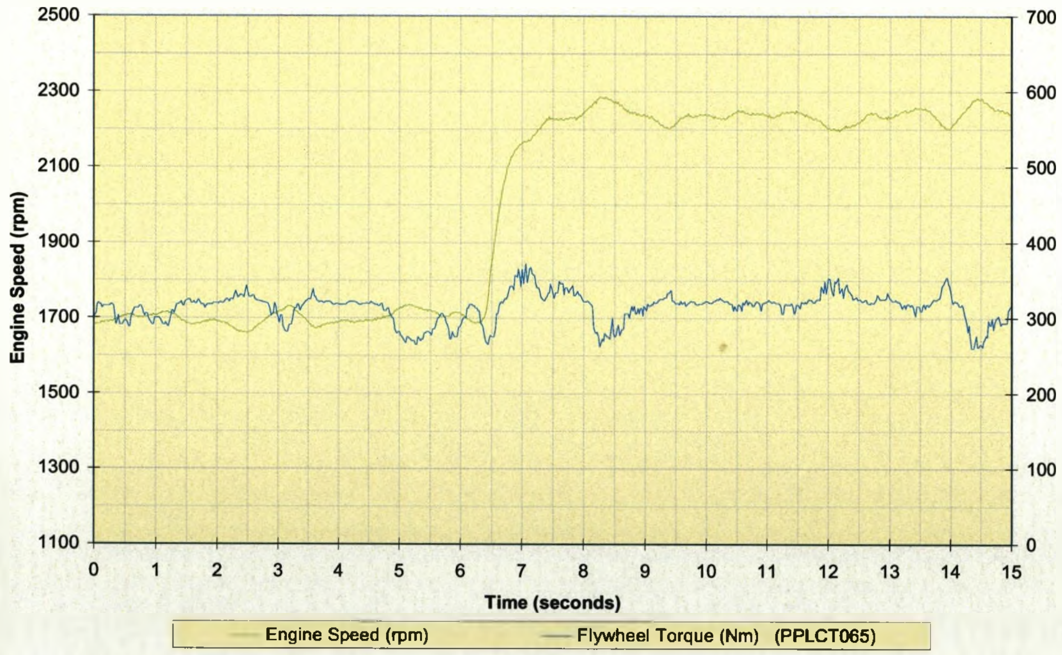



*Figure 5-19 – The effect of a change in engine speed on flywheel torque whilst ploughing clay soil (356mm furrow, gear 5)*

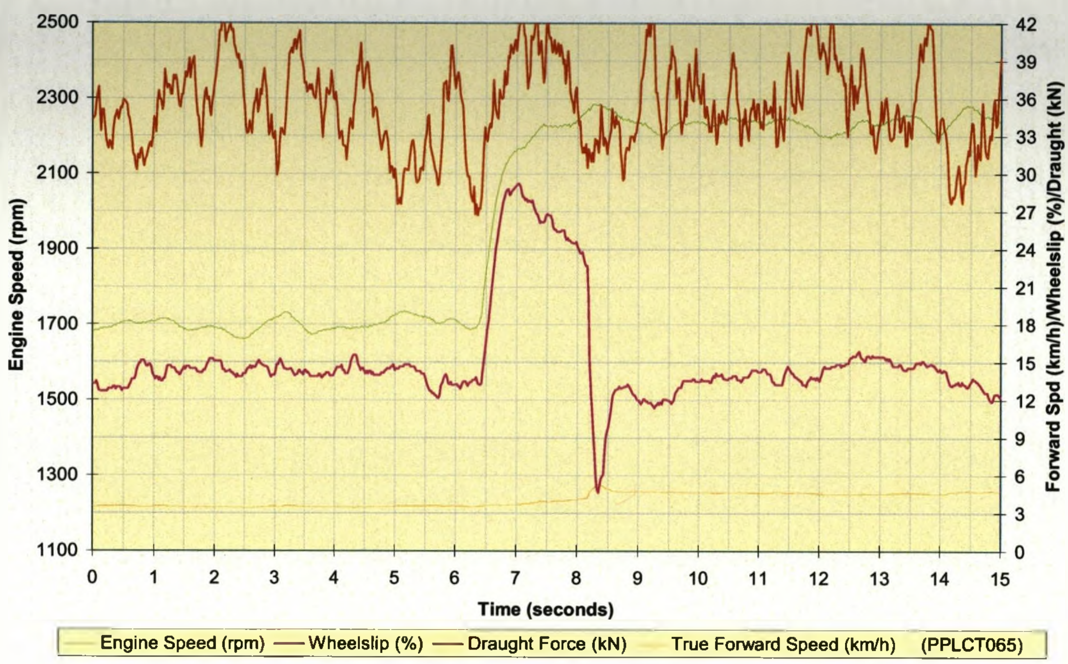



*Figure 5-20 – The effect of a change in engine speed on forward speed, wheelslip and draught (same data) whilst ploughing Clay Soil (356mm Furrow, Gear 5)*

### 5.4.5 Discussion

Problems inherent with undertaking a high draught operation in conditions of natural soil variability often made it difficult to identify the true effects of all transients under investigation, especially at higher forward speeds and engine loading, where the relationships between forward speed, plough draught requirements and engine loading are complex. Nonetheless, the likely impact of each transient is discussed in the following sections.

#### *5.4.5.1 Gearshift*

Table 5.7 demonstrates overall trends from the gearshift experiments. It is difficult to draw definite conclusions due to the variability in the data as a result of the inhomogeneous nature of the clay soil. Gearshifts 3-4, 5-6 and their opposites can be described as single-swap shifts, as only one transmission powershift clutch is released and another engaged to make the change. The characteristics of the 3-4 and 5-6 shifts were very similar to each other both for upshifts and downshifts, and therefore can be considered together. The 6-7 (and opposite) shifts are classified as double-swap shifts, because two pairs of powershift clutches are engaged and released during the shift. The characteristics resulting from a double-swap shift were found to be quite different to the single-swap shift, particularly during a downshift.

During a single-swap upshift, the delay from shift selection to the transmission-engine speed ratio ($r_{te}$) beginning to change was between 0.62 and 0.8 seconds. During this time the torque at the flywheel increased as the oncoming clutch began to engage. As the oncoming clutch pressure ramped up fully and the offgoing clutch disengaged, flywheel torque peaked and $r_{te}$ changed in around 0.3 seconds. Once the ratio had stabilised, the flywheel torque reduced to a post shift level, the total shift time being between 0.9 and 1 second. Total shift time was measured up to $r_{te}$ stabilising and does not include any subsequent settling of the flywheel speed and torque.

During the 6-5 downshift (see Figure 5-12) there was a much shorter delay of between 0.2 and 0.26 seconds from selecting the gear and $r_{te}$ beginning to change. Flywheel torque then peaked just prior to $r_{te}$ reducing as the oncoming clutch pressure was ramped up, whereupon the offgoing clutch was released and flywheel torque reduced rapidly as $r_{te}$ changed. This process took another 0.4 seconds. The overall shift time was between 0.62 and 0.72 seconds.

For the 6-7 upshift (see Figure 5-13) there was a delay of 0.2 seconds before the flywheel torque started to increase as pressure was be ramped up on the C1 clutch. At 0.25 seconds the other oncoming clutch (C4) also began to pressurise, this can be seen as a torque oscillation. As the pressures of both these clutches were still below those required to transmit high levels of torque, $r_{te}$ reduced at 0.3 seconds when the pressure within both offgoing clutches reduced. The oncoming clutch pressures then increased to fully engage the higher gear ratio, resulting in flywheel torque increasing again and $r_{te}$ rising rapidly and then settling at the new ratio. The overall shift time was approximately 0.66 seconds. The subsequent flywheel torque and speed oscillations continued for the remainder of the test, although this was more likely due to soil variations than after-effects of the shift. Whilst the narrow furrow width used during this example test (see Figure 5-13) caused the engine to be heavily loaded and flywheel torque to oscillate, the load variation was accommodated by the engine governor, so progress was maintained. When the widest furrow width setting was used (see Figure 5-14), the upshift was sufficient to heavily load the engine resulting in operation along the full-load curve (see Section 5.3). This caused engine speed to reduce and necessitated operator intervention to prevent stalling. This has to be done quickly as engine speed reduced by over 450rpm, to 1200rpm, in less than 2 seconds.

For the 7-6 downshift (see Figure 5-15) there was a very short delay of around 0.12 seconds prior to $r_{te}$ reducing and the flywheel torque increasing rapidly as both oncoming clutches began to pressurise. Around 0.35 seconds after the shift commenced, pressure to the offgoing clutches rapidly reduced resulting in $r_{te}$ lowering as drive was disengaged. This caused flywheel torque to fall rapidly. At 0.5 seconds the oncoming clutch pressures were rapidly increased. As drive was re-engaged, $r_{te}$ and flywheel torque both increased once more. Total shift time was between 0.8 and 0.9 seconds, although flywheel torque remained higher than the eventual post-shift magnitude for another 1.5 seconds. In both double-swap shifts up and down the rapid reduction in $r_{te}$ as the transmission was disengaged, and the resultant flywheel torque fluctuations, made these shift very noticeable to the operator.

Regardless of shift type, post-shift flywheel torque increased for an upshift and reduced for a downshift from the pre-shift levels. Generally the forward speed of the tractor increased as a result of an upshift. There were two exceptions: a 5-6 shift ($W_P$ = 1735mm) and a 6-7 shift ($W_P$ = 2010mm) where the forward speed reduced or stayed the same. In both these instances this was as a result of the large torque increase reducing engine speed. In the case of the 5-6 shift this resulted in a period of torque-speed oscillation, whereas in the 6-7 situation the test had to be aborted as engine speed continued to be reduced beyond a sustainable level.

The effect of furrow width was not always clear during these tests. Generally the increase in width caused a higher level of flywheel torque loading, although, as can be seen from Figures 5-13 and 5-14, the shift characteristics remained very similar. As with the steady state experiments, a furrow width increase did not always result in a flywheel torque increase.

#### *5.4.5.2 Change in Plough Working Depth*

Plough draught, and therefore flywheel torque, was influenced by plough working depth ($D_P$). Increases in depth resulted in a higher draught force requirement, an increase in flywheel torque and a reduction in engine speed. Table 5.8 demonstrates these trends. Rockshaft position ($\varepsilon_r$) was used as the basis for depth control as it was visible to the operator during data collection. Typical changes in mean working depth were between 80mm and 100mm. Table 5.8 also shows the effect on forward speed of increasing the plough depth, especially in the higher draught requirement of the clay soil. Forward speed was reduced as draught increased due to the increase in wheelslip, as well as the reduction in engine speed.

Figure 5-16 shows the effect of increasing the ploughing depth in a clay soil at a moderate loading (gear 5, 356mm furrow width setting). As $D_P$ increased there was a time delay of just over 1 second before flywheel torque increased and engine speed slightly reduced due to the stiffness and damping in the tyres and driveline, as well as the engine response. Following the depth increase there was a degree of instability as the engine controller attempted to respond to a continually changing draught requirement. As the engine torque load was still less than 300Nm, even at the deeper working depth, the engine and vehicle speed only reduced slightly.

Increasing working depth at a higher forward speed had a greater impact on the draught requirement and therefore flywheel torque demand and engine speed. Figure 5-17 demonstrates this, for the same conditions as previously described, but in gear 7. The soil variation had more effect on flywheel torque demand at this higher forward speed (approximately 7.5km/h), making the influence of plough depth increase more difficult to determine. As working depth increased, engine loading reached the full-load curve after which point engine speed reduced dramatically. In turn this reduced the vehicle forward speed. As was discussed during the steady state work (see Section 5.3), this forward speed reduction would reduce the plough draught allowing the engine to recover speed. That engine (and vehicle) speed recovery would then serve to increase the draught again, putting the engine into relatively unstable operation with continually changing speed. The clay soil, continually providing additional disturbance, would mean engine speed would be unlikely to fully stabilise.

In comparison to Figure 5-17, the same test was performed on sandy soil and is presented in Figure 5-18. The lighter, more uniform loading from the sandy soil resulted in a much more stable operation of the vehicle. It should be noted that a software fault resulted in the flywheel torque signal only being transmitted to the vehicle CAN-bus at 10Hz rather than 100Hz during these experiments. Irrespective of this, even when the working depth was increased, the flywheel torque increase was still within the governed operating range of the engine and therefore engine speed reduction was minimal (20rpm).

#### *5.4.5.3 Change in Engine Speed*

An increase or decrease in engine speed showed few interesting effects on powertrain loading (see Figures 5-19 and 5-20), with the soil variability more influential on loading despite a low gear being used. Despite an overall increase in forward speed (3.5km/h to 4.7km/h), mean draught and flywheel torque showed little increase (neither more than 5%) compared to the slower engine speed, the same being true for all tractor-implement configurations investigated. The small overall change in mean draught force, from before to after the change in engine speed, resulted in wheelslip remaining constant (in this example 14%). However, during the engine speed change itself, the sudden engine acceleration and a relatively stiff driveline resulted in the tractor wheels accelerating rapidly. The high draught force from the plough prevented the vehicle speed from quickly increasing. Therefore instantaneous wheelslip peaked at 30%. As the tyres gained traction and accelerated the vehicle, wheelslip reduced again. Sudden changes in engine speed causing high wheelslip could potentially cause soil damage and so should be avoided.

## 5.5 *Power Harrowing - Steady State*

### 5.5.1 Objective

The objective of this part of the experimental programme was to determine the influence of different tractor-implement parameters on powertrain loading, whilst power harrowing in two soil types, and to determine the typical variation experienced due to soil variation within the field.

### 5.5.2 Experimental Equipment

The test tractor was operated with a Dowdeswell 400S 4m power harrow (see Figure 5-21) capable of loading the tractor powertrain sufficiently through the P.T.O. driveline. The cultivating effort of this machine comes from pairs of contra-rotating vertical blades, each rigidly connected to the next in a gear train across the width of the machine. A hydraulically adjusted packer roller at the rear of the machine consolidates the cultivated soil and controls working depth (see Appendix A1.3).

*Figure 5-21 – Test tractor and power harrow during the field investigation*

### 5.5.3 Experimental Design & Procedure

The two soil types previously ploughed (see Sections 5.3 and 5.4) were used for the power harrow investigation, allowing the effects of soil type variation to be identified. Prior to this investigation the fields were consolidated with a light roller to remove variability in furrow shapes and sizes introduced during ploughing.

Three working depths were investigated - namely 75mm, 100mm and 125mm - in conjunction with three tractor transmission gear ratios across the typical working range (gears 4, 5 and 6) relating to theoretical forward speeds of 4.3km/h, 5.6km/h and 6.8km/h. A mean constant loaded engine speed of 1950rpm was maintained where possible, giving a P.T.O. speed of 920rpm. A constant rotor speed was used, this being 308rpm at the loaded engine speed. Each tractor-implement configuration was replicated three times in both sandy loam and clay soil, resulting in 27 individual tests for each soil type. Once equilibrium conditions had been reached in each experimental run, data was recorded for 120 seconds, at a sampling rate of 100Hz, and subsequently averaged to 10Hz during processing.

### 5.5.4 Parameters

The acquired tractor-implement parameters for this part of the investigation are outlined in Tables 5.1 and 5.3. In addition, a number of secondary parameters including P.T.O. power, flywheel power and drawbar power were calculated at each time step during data processing (see Appendix A3.22).

### 5.5.5 Results

Absolute mean values (both within each trial and across the replications) of the key data for each tractor-implement configuration are presented in Table 5.9. Additional mean and standard deviation data for each experimental run is presented in Appendix A4.2.

Figures 5-22 and 5-23 present combined engine torque-speed scatter data arising from each transmission gear ratio used in the sandy loam and clay soil types respectively, at

100mm working depth. Additional distributions for 75mm and 125mm depths are presented in Appendix A4.2. A distribution showing the effect of different depths in the same gear (6) is shown in Figure 5-24. Two datasets are presented as time histories showing the extremes of operation: gear 4 clay soil at 75mm depth (see Figure 5-25) and gear 6 sandy soil at 125mm depth (see Figure 5-26).

*Table 5.9 – Power harrow (steady state) results summary*

| | | Draught Force $H_A$ (kN) | | P.T.O. Torque $T_{BH}$ (Nm) | | Flywheel Torque $T_F$ (Nm) | | % torque due to P.T.O. | | Engine Speed $\omega_E$ (rpm) | | True Speed $v_a$ (km/h) | | Flywheel Power $P_F$ (kW) | | Slip % S (%) | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Sandy | Clay | Sandy | Clay | Sandy | Clay | Sandy | Clay | Sandy | Clay | Sandy | Clay | Sandy | Clay | Sandy | Clay |
| **75mm Working Depth ($W_d$)** | | | | | | | | | | | | | | | | | |
| Gear | 4 | 12.9 | 7.8 | 513 | 218 | 328 | 204 | 74 | 50 | 1917 | 1913 | 3.4 | 3.5 | 65.7 | 40.8 | 5.8 | 3.6 |
| | 5 | 13.8 | 8.9 | 526 | 259 | 386 | 263 | 64 | 46 | 1877 | 1906 | 4.2 | 4.4 | 75.9 | 52.5 | 8.4 | 5.1 |
| | 6 | 17.0 | 11.0 | 543 | 322 | 446 | 343 | 57 | 44 | 1874 | 1909 | 5.1 | 5.3 | 87.6 | 68.6 | 9.1 | 6.8 |
| **100mm Working Depth ($W_d$)** | | | | | | | | | | | | | | | | | |
| Gear | 4 | 8.9 | 8.9 | 396 | 363 | 278 | 282 | 67 | 61 | 1904 | 1899 | 3.5 | 3.7 | 55.4 | 56.1 | 1.7 | 4.0 |
| | 5 | 12.1 | 11.4 | 471 | 407 | 363 | 328 | 61 | 59 | 1891 | 1913 | 4.4 | 4.3 | 71.9 | 65.6 | 3.2 | 6.8 |
| | 6 | 15.8 | 12.8 | 549 | 460 | 468 | 398 | 55 | 55 | 1913 | 1895 | 5.3 | 5.2 | 93.9 | 78.9 | 6.3 | 8.4 |
| **125mm Working Depth ($W_d$)** | | | | | | | | | | | | | | | | | |
| Gear | 4 | 13.8 | 11.1 | 649 | 426 | 403 | 325 | 76 | 62 | 1911 | 1924 | 3.4 | 3.3 | 80.6 | 65.5 | 5.3 | 8.1 |
| | 5 | 17.5 | 13.5 | 727 | 496 | 502 | 391 | 68 | 60 | 1903 | 1910 | 4.2 | 4.2 | 100.0 | 78.2 | 8.0 | 9.5 |
| | 6 | 19.6 | 14.8 | 731 | 549 | 564 | 463 | 61 | 56 | 1801 | 1902 | 4.7 | 4.9 | 106.1 | 92.2 | 11.2 | 13.4 |

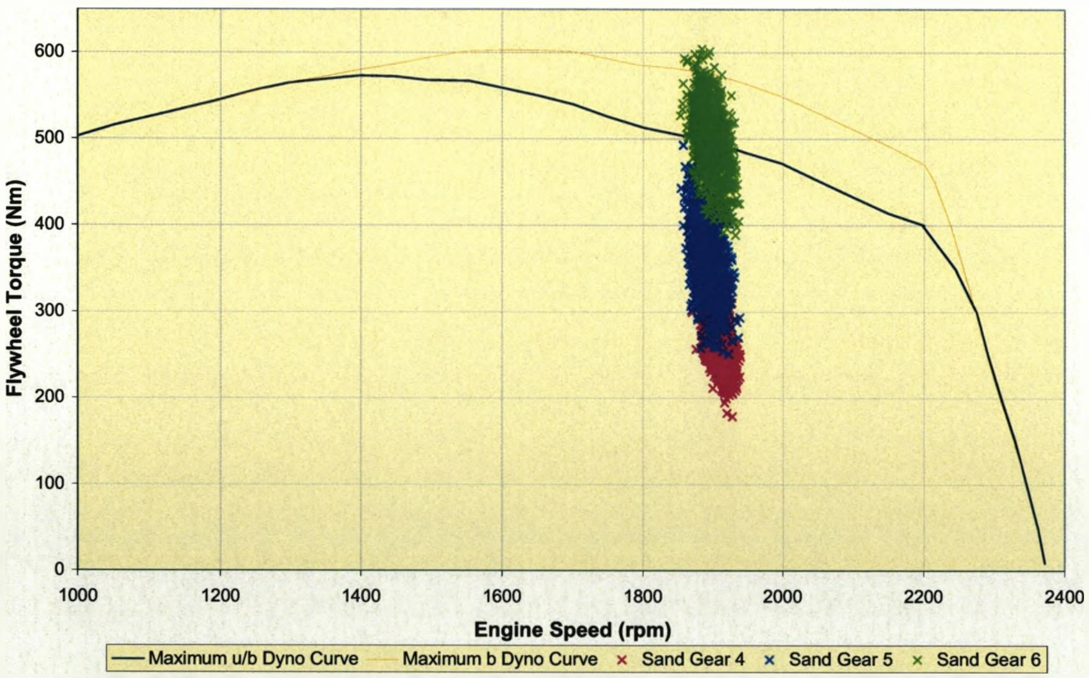



*Figure 5-22 – The effect of gear selection on dynamic loading whilst power harrowing sandy soil (100mm tine depth)*

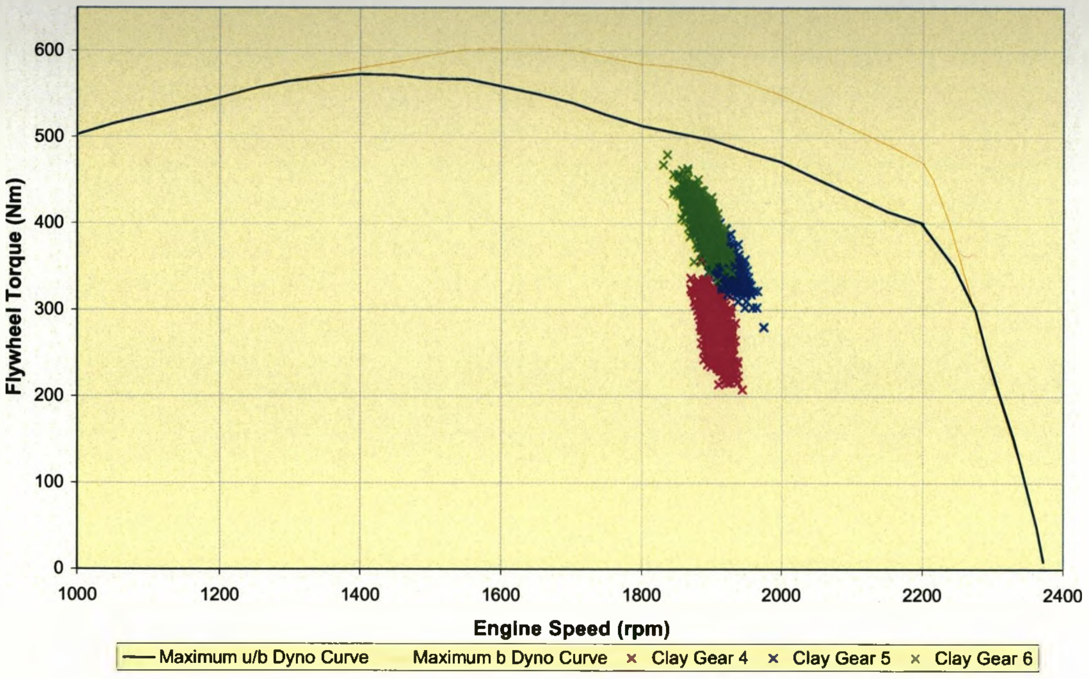



*Figure 5-23 – The effect of gear selection on dynamic loading whilst power harrowing clay soil (100mm tine depth)*

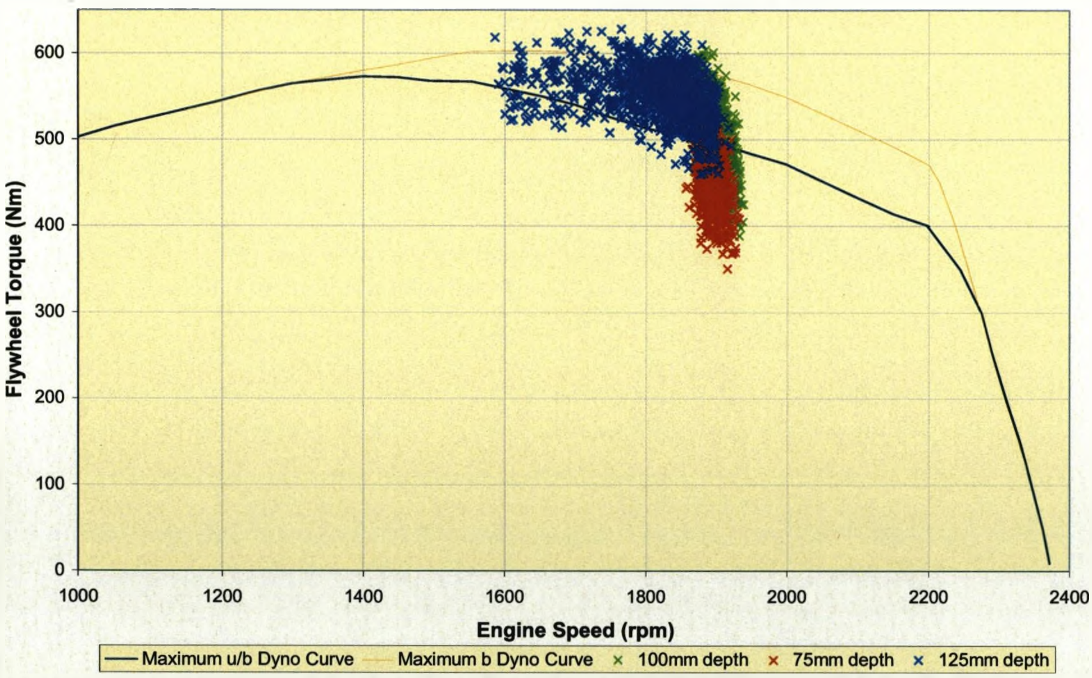



*Figure 5-24 – The effect of working depth on dynamic loading whilst power harrowing sandy soil (gear 6)*

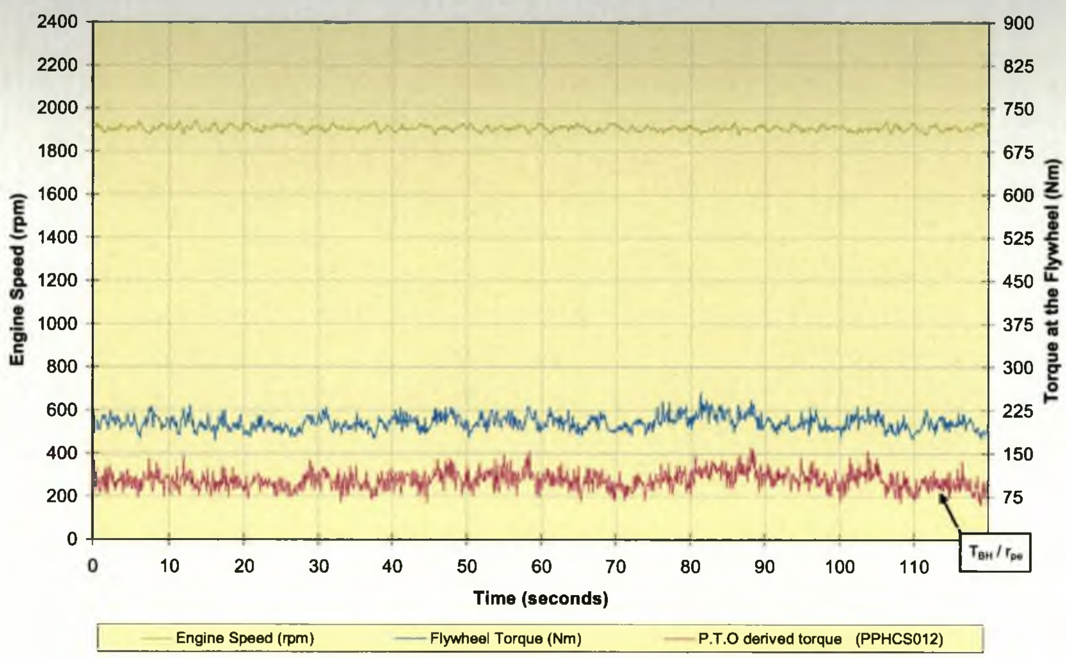



*Figure 5-25 – Example test data time history - clay soil (depth 75mm, gear 4)*

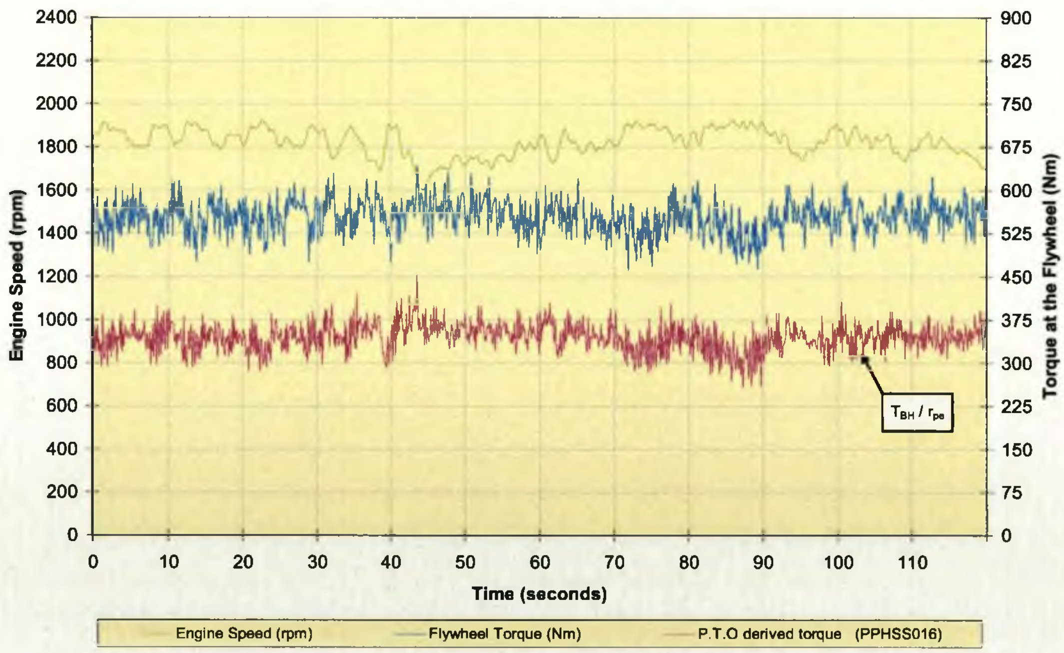



*Figure 5-26 – Example test data time history - sandy soil (depth 125mm, gear 6)*

### 5.5.6 Discussion

As can be seen from the results summary (see Table 5.9) and the torque-speed loading charts (see Figures 5-22 and 5-23), the load arising from the power harrow with the same configuration was generally greater for the sandy soil. This is the opposite of the ploughing data, where the heavier clay soil required more energy. This is most likely a result of the differences between the two soil structures. Clay soil particles tend to be collected in large blocks, whereas sandy soil tends to have a fine grained structure and as a result tends to present a continual force on the power harrow tines, requiring more effort to cultivate. This higher power requirement of sandy soil was also found by previous workers using the same soil types (Scarlett *et al*, 1998; Bentley, 2000). It should be noted that whilst the sandy soil required more power, the power harrow produced a seedbed. In contrast, the clay soil required further tillage prior to planting. It can also be seen from Figures 5-22 and 5-23 that there are differences between the engine torque-speed profiles for the two soil types, this was as a result of a software fault during the clay soil investigation.

During power harrowing, engine loading resulted from both the traction and P.T.O. drivelines. The percentage of flywheel torque from the P.T.O. (shown in Table 5.9) varied according to the soil type, working depth and gear. It should be noted that the non-P.T.O. torque comprises of components both from the traction driveline (implement draught and rolling resistance) and powertrain torque losses. Increasing the transmission gear reduced the percentage of flywheel torque demand, which originated from the P.T.O., as a result of higher draught force. This increased rolling resistance and possibly increased driveline losses.

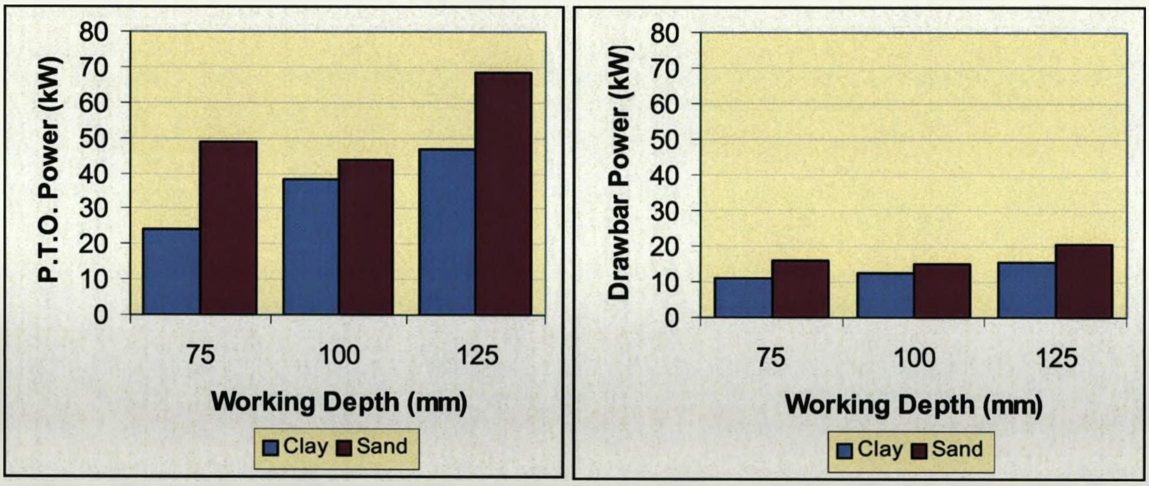



*Figure 5-27 – Comparison of P.T.O. and drawbar power demand when power harrowing at different working depths in gear 5*

The differences in power requirements between the two soil types with respect to working depth is shown in Figure 5-27 for gear 5, in addition to the tabular data (see Table 5.9). As can be seen, at the shallowest operating depth the sandy soil required double the P.T.O. power of the clay soil and 50% more draught (drawbar) power. This resulted in significantly higher flywheel torque loading for the sandy soil seen in (see Table 5.9). The 100mm operating depth provided the least difference between the two soils, both for P.T.O. and draught power requirements. However, as can be seen from Figures 5-22 and 5-23, the operating range in the sandy soil was generally higher. At 125mm working depth the differences between the two soil types are clear again, with both P.T.O. and draught power higher for the sandy soil (giving at least a 22% increase in flywheel torque).

Operating at the deepest working depth in sandy soil, particularly in gears 5 and 6, heavily loaded the tractor powertrain causing an engine speed (and therefore forward speed) reduction, whilst still requiring over 100kW flywheel power. The power boost feature allowed the engine to develop extra torque to maintain progress (see Figure 5-22). Without power boost, given the torque requirement, the engine operating speed would have been reduced further along the full-load curve.

Within the same soil type, increasing the transmission gear ratio increased both the draught force and the P.T.O. torque requirement from the power harrow, resulting in an overall increase in flywheel torque and power demand. During the power harrowing experiments wheelslip was far lower than during ploughing, as the majority of tractor power was transmitted through the P.T.O. driveline rather than the traction driveline. However, slip did become more significant in gear 6 at the deepest working depth used (125mm) as a result of the relatively large draught force encountered (see Table 5.9).

Increasing working depth had the effect of increasing the draught, P.T.O. torque, flywheel torque and power requirements in the clay soil. In the sandy soil increasing the working depth from 75mm to 100mm had the opposite effect, in that the mean draught force, P.T.O. torque and flywheel torque all reduced. Figure 5-24 shows this trend, although it is not really possible to distinguish between the regions of operation for 75mm and 100mm working depth. The reduction in mean loading could be as a result of an optimum depth of operation in those conditions for that soil type. With a different rotor speed this may have not occurred. Increasing working depth from 100mm to 125mm resulted, as would be expected, in all loading parameters rising.

Dynamic variation in powertrain loading at the two extremes of operation, i.e. clay soil in gear 4 at 75mm working depth (see Figure 5-25) and sandy soil in gear 6 at 125mm working depth (see Figure 5-26), show that the overall tractor engine loading was more stable during power harrowing than in ploughing, where large variations in engine torque-speed demand were commonplace. Consequently during the power harrowing investigation no tests needed to be aborted due to overloading and permitted greater engine power utilisation.

In the lowest loading situation (see Figure 5-25) there is very little variation in engine speed (standard deviation 12rpm) or flywheel torque demand (standard deviation 13Nm). Figure 5-26 shows the highest powertrain loading experienced during power harrowing. Despite the high mean flywheel torque demand of 550Nm, the standard deviation was only 29Nm because the engine was able to maintain operation in the governed engine range, thereby reducing engine speed variation. Mean engine speed was 1826rpm, with a standard deviation of 64rpm.

## 5.6 Power Harrowing - Transient

### 5.6.1 Objective

The objective of the transient power harrowing experiments was to determine the variation in powertrain loading which occurs during a step change in one of the key operating parameters. Two specific parameters were investigated:

1. a change in powershift gear (increase and decrease); and
2. a change in working depth (increase and decrease).

### 5.6.2 Experimental Design & Procedure

Each experimental run was recorded prior to making a parameter change and then subsequently recorded for a sufficient time to allow the effects of the transient (step change) to stabilise. This whole process typically took 15 to 20 seconds. A data sampling rate of 100Hz was used, averaged to 50Hz prior to recording.

#### 5.6.2.1 Gearshift

As with ploughing, gearshifts were only undertaken between powershift gears. Four shifts were investigated: upshifts and downshifts between gear 3 and 4 and between gear 5 and 6. The same shifts were used for both soil types although, as with ploughing, only data arising from clay operating conditions were analysed. Each shift was replicated three times.

#### 5.6.2.2 Change in Working Depth

Working depth was varied during a run by adjusting the height of the rear packer roller via the tractor external hydraulics. The limits of adjustment were set at the minimum and maximum depths used in the steady state experiments, i.e. 75mm and 125mm, although this was difficult to achieve as only visual feedback of hydraulic ram extension was available to the operator. Experimental runs were undertaken both increasing and decreasing the depth in each of the transmission gears used in the steady state experiments. Each transient configuration was replicated three times.

### 5.6.3 Parameters

The same parameters (both acquired and derived) as were previously used (see Section 5.5.4) were considered during the transient power harrowing experiments.

### 5.6.4 Results

#### *5.6.4.1 Gearshift*

A summary of the key engine speed and torque demand data experienced during the different gearshifts and different working depths is presented in Table 5.10. Graphical representations are presented for a 5-6 gearshift at 125mm depth (see Figure 5-28) and a 4-3 shift at 100mm working depth (see Figure 5-29). As all the gearshifts investigated were single-swap shifts, 3-4 and 6-5 gearshifts are not presented graphically as their profile is very similar to the 5-6 and 4-3 shifts respectively albeit at slightly different levels of flywheel torque.

*Table 5.10 – Key power harrow gearshift data (clay soil)*

| Shift | Clutches | $D_{HA}$ (mm) | $r_{te}$ profile | $T_F$ profile | Pre-shift $\omega_E$ (rpm) | Pre-shift $T_F$ (Nm) | Pre-shift $v_a$ (km/h) | Shift delay* (sec) | Total time** (sec) | During the Shift $T_F$ peak | During the Shift peak time*** | During the Shift $\omega_E$ drop | Post-shift $\omega_E$ (rpm) | Post-shift $T_F$ (Nm) | Post-shift $v_a$ (km/h) |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 3-4 | C4>C3 | 77 | | | 1922 | 170 | 2.90 | 0.36 | 0.82 | 347 | 0.74 | 1846 | 1914 | 194 | 3.43 |
| 3-4 | C4>C3 | 104 | | | 1906 | 203 | 2.86 | 0.26 | 0.82 | 447 | 0.64 | 1821 | 1891 | 257 | 3.36 |
| 3-4 | C4>C3 | 126 | | | 1919 | 260 | 2.80 | 0.36 | 0.84 | 440 | 0.64 | 1842 | 1910 | 322 | 3.30 |
| 4-3 | C3>C4 | 78 | | | 1920 | 191 | 3.52 | 0.30 | 0.78 | 325 | 0.36 | 1855 | 1920 | 175 | 2.96 |
| 4-3 | C3>C4 | 103 | | | 1893 | 269 | 3.45 | 0.22 | 0.62 | 350 | 0.42 | 1844 | 1900 | 215 | 2.91 |
| 4-3 | C3>C4 | 124 | | | 1916 | 317 | 3.34 | 0.32 | 0.78 | 454 | 0.44 | 1848 | 1930 | 249 | 2.88 |
| 5-6 | C4>C3 | 77 | | | 1914 | 287 | 4.41 | 0.68 | 0.92 | 403 | 0.76 | 1852 | 1892 | 345 | 5.13 |
| 5-6 | C4>C3 | 104 | | | 1898 | 318 | 4.34 | 0.68 | 0.86 | 479 | 0.90 | 1822 | 1871 | 376 | 5.06 |
| 5-6 | C4>C3 | 126 | | | 1893 | 388 | 4.00 | 0.70 | 0.86 | 647 | 0.82 | 1794 | 1866 | 457 | 4.62 |
| 6-5 | C3>C4 | 78 | | | 1886 | 315 | 5.23 | 0.22 | 0.70 | 397 | 0.34 | 1834 | 1899 | 276 | 4.43 |
| 6-5 | C3>C4 | 104 | | | 1868 | 379 | 5.05 | 0.26 | 0.70 | 464 | 0.36 | 1834 | 1894 | 324 | 4.36 |
| 6-5 | C3>C4 | 127 | | | 1895 | 464 | 4.68 | 0.24 | 0.66 | 602 | 0.28 | 1815 | 1920 | 389 | 4.15 |

* Time from gear selection to $\omega_{te}$ starting to change

** Time from gear selection to ωte stabilising post-shift

*** Time from gear selection to torque peak (or trough)

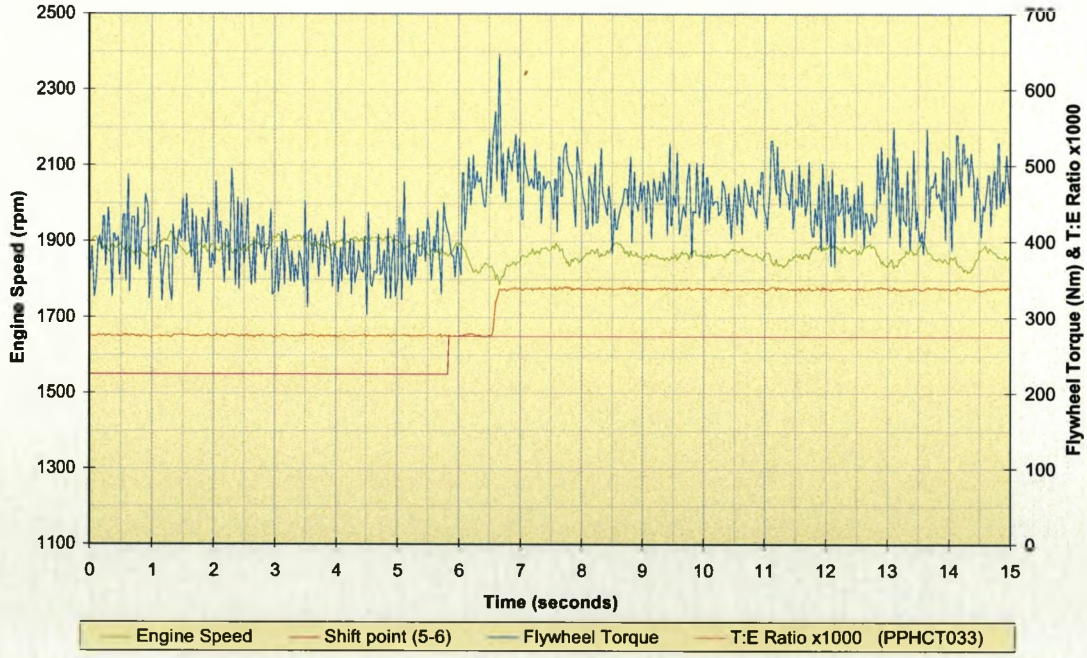



*Figure 5-28 – The effect of a 5-6 gearshift on engine speed and flywheel torque whilst power harrowing in clay soil (125mm depth)*

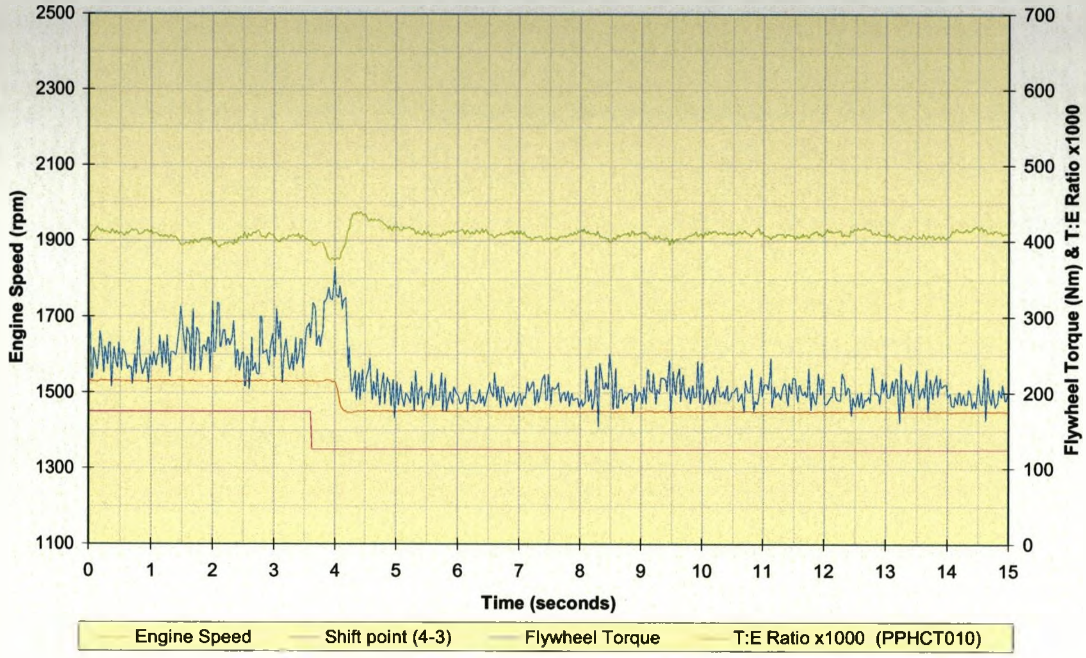



*Figure 5-29 – The effect of a 4-3 gearshift on engine speed and flywheel torque whilst power harrowing in clay soil (100mm depth)*

### 5.6.4.2 Change in Working Depth

A summary table of key depth change data is presented in Table 5.11. One dataset only is presented graphically for an increase in working depth. Figure 5-30 depicts the effects on engine speed and Figure 5-31 presents the resultant P.T.O. torque demand referenced to the flywheel ($T_{BH}/r_{pe}$). The time where power boost activates is shown on both diagrams.

*Table 5.11 – Mean data summary for key parameters during power harrow depth change*

| Soil | Gear | Depth Change | Pre-change | | | | | Post-change | | | | | File Ref |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | $D_{HA}$ (mm) | $\omega_E$ (rpm) | $T_{BH}/r_{pe}$ (Nm) | $T_F$ (Nm) | $v_s$ (km/h) | $D_{HA}$ (mm) | $\omega_E$ (rpm) | $T_{BH}/r_{pe}$ (Nm) | $T_F$ (Nm) | $v_s$ (km/h) | |
| Sandy | 4 | increase | 78.8 | 1936 | 202 | 315 | 3.45 | 126.3 | 1894 | 364 | 520 | 3.22 | PPHSTD49 |
| Sandy | 4 | decrease | 125.8 | 1904 | 327 | 451 | 3.38 | 77.1 | 1938 | 181 | 259 | 3.61 | PPHSTD52 |
| Sandy | 5 | increase | 76.5 | 1932 | 199 | 343 | 4.51 | 116.9 | 1890 | 360 | 553 | 3.97 | PPHSTD41 |
| Sandy | 5 | decrease | 123.9 | 1900 | 357 | 529 | 4.17 | 83.9 | 1922 | 243 | 391 | 4.35 | PPHSTD53 |
| Sandy | 6 | increase | 75.0 | 1914 | 235 | 429 | 5.27 | 116.4 | 1761 | 353 | 587 | 4.63 | PPHSTD47 |
| Sandy | 6 | decrease | 120.2 | 1721 | 370 | 589 | 4.51 | 73.2 | 1912 | 252 | 435 | 5.22 | PPHSTD54 |
| Clay | 4 | increase | 78.0 | 1905 | 98 | 206 | 3.43 | 119.8 | 1889 | 187 | 320 | 3.30 | PPHCTD40 |
| Clay | 4 | decrease | 127.3 | 1964 | 195 | 331 | 3.40 | 89.2 | 1980 | 121 | 233 | 3.53 | PPHCTD43 |
| Clay | 5 | increase | 79.2 | 1918 | 125 | 271 | 4.41 | 118.3 | 1886 | 192 | 357 | 4.18 | PPHCTD54 |
| Clay | 5 | decrease | 127.3 | 1895 | 231 | 410 | 4.11 | 84.0 | 1947 | 131 | 283 | 4.48 | PPHCTD48 |
| Clay | 6 | increase | 74.2 | 1902 | 132 | 311 | 5.38 | 121.7 | 1849 | 225 | 454 | 4.66 | PPHCTD52 |
| Clay | 6 | decrease | 126.9 | 1925 | 227 | 459 | 4.83 | 84.9 | 1969 | 150 | 342 | 5.39 | PPHCTD39 |

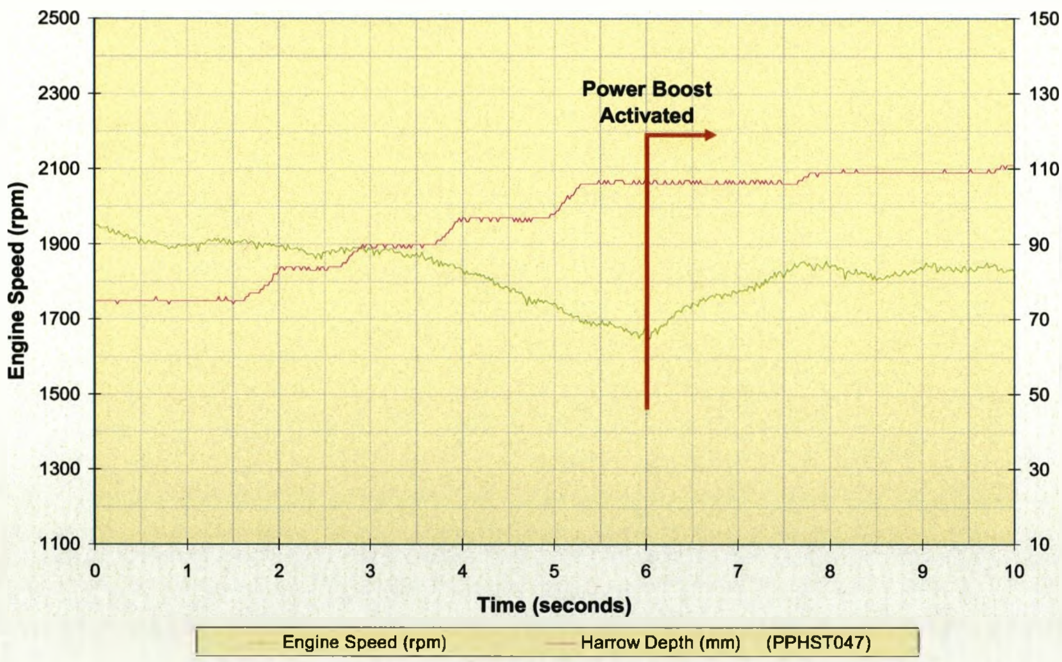



*Figure 5-30 – The effect of increasing working depth on engine speed whilst power harrowing in sandy soil (gear 6)*

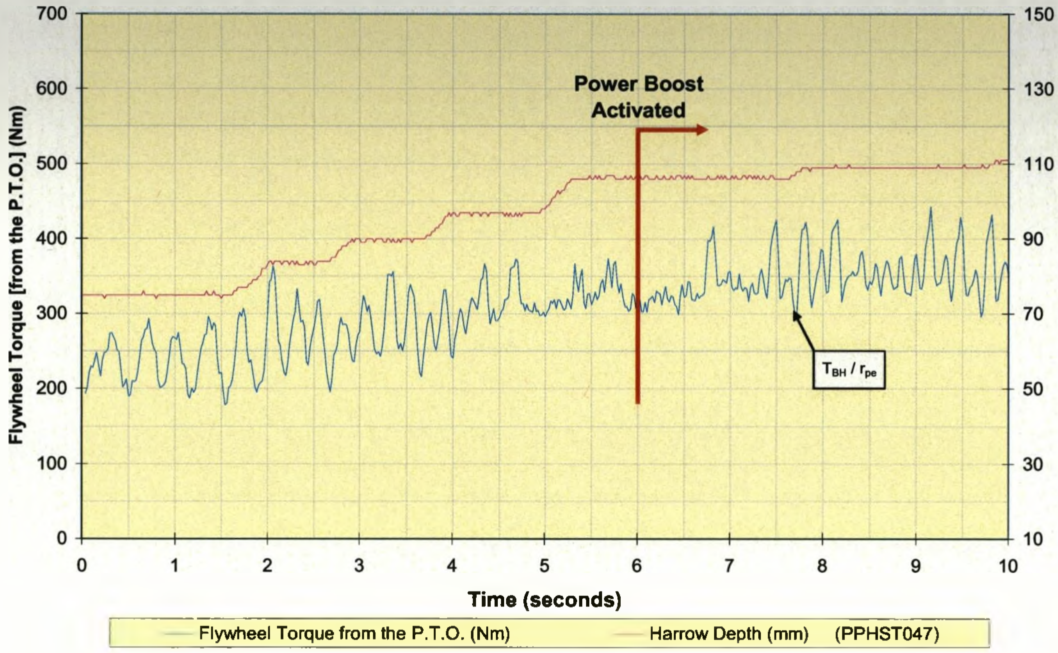



*Figure 5-31 – The effect of increasing working depth on P.T.O. torque at the flywheel whilst power harrowing in sandy soil (gear 6)*

## 5.6.5 Discussion

### *5.6.5.1 Gearshift*

As only single-swap shifts were undertaken (see Section 5.6.2.1), the shift characteristics were similar regardless of gear. This is confirmed by the shape of the $r_{te}$ (ratio of transmission output to engine speed) and the $T_F$ (flywheel torque) profiles in Table 5.10. The two upshifts varied in the length of the shift delay before $r_{te}$ started to change. For the 3-4 shift it was between 0.26 and 0.36 seconds, whereas the 5-6 shift was around 0.7 seconds. However, the overall shift time still remained very similar, around 0.8 seconds, to the point where $r_{te}$ stabilised. During the delay the oncoming clutch pressure was progressively raised, increasing the flywheel torque (see Figure 5-28). After the delay the oncoming clutch was fully pressurised and the offgoing clutch released, resulting in $r_{te}$ changing rapidly and causing a flywheel torque 'spike'.

The two downshifts were of very similar profile, as can be seen in Table 5.10. The delay prior to $r_{te}$ changing was between 0.22 and 0.32 seconds for both the 4-3 and 6-5 shifts. As the oncoming clutch pressure increased the flywheel torque peaked, just prior to $r_{te}$ reducing (see Figure 5-29). Subsequently flywheel torque and $r_{te}$ reduced rapidly as the offgoing clutch was released. This process took a further 0.4 to 0.48 seconds.

All shifts, even at high flywheel torques, were less erratic than those experienced during ploughing as a result of the lower draught requirement of the power harrow. The result was a stable engine speed during all shifts. At the higher engine loadings this can be partly attributed to the extra engine power available as a result of the power boost feature. Power boost allowed the engine to operate under governor control over a greater range of torque output rather than along the full-load curve, where the torque-speed characteristics lead to a greater change in engine speed for a given change in torque demand. Regardless of shift type, mean flywheel torque increased for an upshift and reduced for a downshift from the pre-shift levels. Vehicle forward speed always increased for an upshift and reduced for a downshift as slip, even at higher loads, was significantly lower than during ploughing.

The different power harrow working depths had little influence on the gearshift, other than to increase the mean flywheel torque demand both before and after the shift for a greater working depth. The greater working depths also resulted in a higher peak flywheel torque requirement during the shift (see Table 5.10).

#### *5.6.5.2 Change in Working Depth*

The fundamental power harrow power requirement - working depth relationships were identified during the steady state investigation (see Figure 5-27). The trends were repeated during transient depth change trials (see Table 5.11). Figure 5-30 shows the effect of increasing the power harrow working depth in a sandy soil upon engine speed. As depth increased, the resulting higher torque requirement reduced engine speed. In this example, as depth increased engine speed reduced to around 1700rpm, because the engine was operating on the full-load curve. As the mean P.T.O. torque requirement (see Figure 5-31) increased, the engine controller allowed power boost to operate and the resulting additional torque made available allowed engine speed to increase once again.

This example shows the benefit of the power boost feature and demonstrates how it begins to operate when depth (and P.T.O. torque demand) increases. It should be noted that despite the manufacturers claim of the feature operating on a graduated basis, depending on the proportion of torque division along the traction and P.T.O. drivelines (as discussed in Section 3.2), in reality when P.T.O. torque was sufficient to allow the engine to enter boosted mode, it always operated at the maximum level (or within 1% of it). In addition, the engine didn't always appear to change between the boosted and unboosted modes when expected. A further investigation of this feature's operational characteristics was therefore undertaken (see Section 9).

## 5.7 Transport – Road

### 5.7.1 Objective

The objective of this part of the experimental programme was to determine the typical variation in powertrain loading experienced whilst undertaking transport activities.

### 5.7.2 Experimental Equipment

The road transport investigation was conducted with a Marston 'Ace' 10 tonne capacity, tandem-axle grain trailer (see Figure 5-32), typical of the trailer size used with a tractor of this engine power. The trailer was ballasted to give a total gross train weight (tractor + trailer + load) of 19,164kg (see Appendix A1.4).

*Figure 5-32 – Test tractor and trailer prior to transport experiments*

### 5.7.3 Experimental Design & Procedure

During transport, the road profile, size and surface quality are principle factors in determining powertrain loading and the tractor's ability to maintain forward speed. Four sections of public road were used to provide a range of transport situations, each taking between 4 and 8 minutes to complete. The road sections proved challenging to the test tractor, due to the uphill gradients, the road surface quality or width restricting forward speed. Each section of the test route was replicated three times, avoiding peak commuter times, to ensure the limitations in performance were due to road characteristics rather than interactions with other vehicles. Data was sampled at 100Hz and averaged to 10Hz prior to recording. The sections used were:

1. Barton to Pegsdon (BTPD) – A section of the B655 (Barton – Hitchin road). Road surface and width did not restrict forward speed but featured a number of steep up and down gradients (10%), in some cases severely restricting forward speed (section length: 3.4km);

2. Pegsdon to Hitchin (PDHN) – The next section of the B655 and therefore similar road quality. The road featured a number of less severe (but longer in duration) up gradients followed by a period of relatively level road (section length: 4.6km);

3. Hitchin to Pirton (HNPT) – A section of unclassified road linking the B655 to Pirton. The surface quality and the width of the road were inferior to the previous sections, therefore restricting forward speeds due to operator comfort (despite the test tractor having cab suspension). The route featured less severe, long up and downhill gradients (section length: 2.5km);

4. Pirton to Shillington (PTSH) – The final section was undertaken through the village of Pirton, characterised by narrow roads and parked vehicles, requiring a low forward speed and extreme care to be taken. The village featured three road junctions to be navigated, before a final steep section out of the village with a further road junction at the end to be navigated before completion of the section (section length: 1.1km).

### 5.7.4 Parameters

The acquired tractor parameters for this part of the investigation were outlined in Table 5.1. Flywheel power was also calculated at each time point during data processing (see Appendix A3.23).

### 5.7.5 Results

Data arising from the three replications for each section of the test route have been combined to produce individual flywheel torque-speed frequency distributions (see Figures 5-33, 5-34, 5-35 and 5-36). In addition, an overall combined torque-speed distribution for the entire route is presented in Figure 5-37. Analysis of each route section has been undertaken to determine the operational characteristics of the vehicle in terms of engine torque-speed demand, forward speed and gearshift frequency (see Table 5.12).

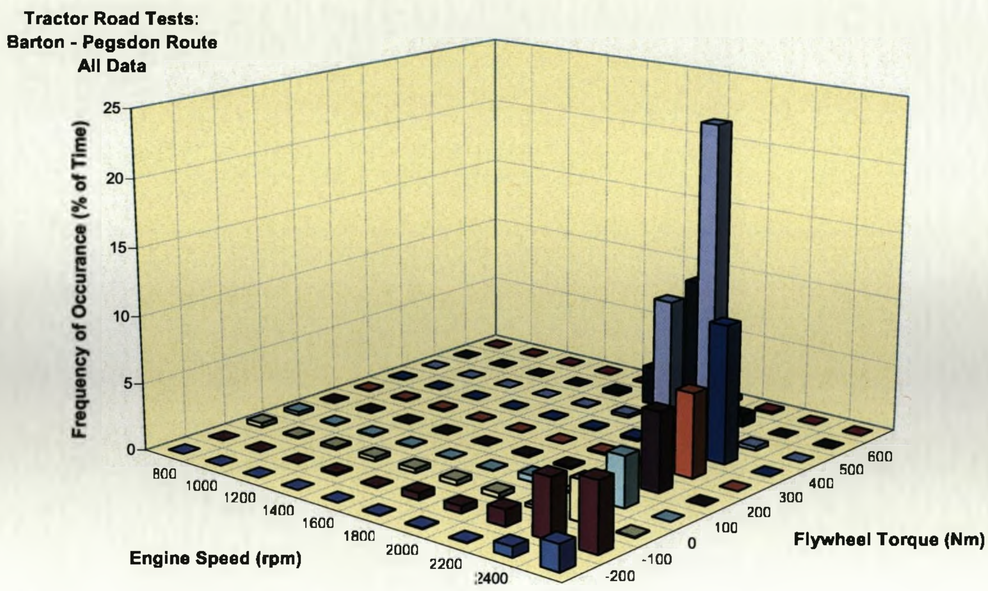



*Figure 5-33 – Flywheel torque – engine speed frequency distribution: Barton - Pegsdon section*

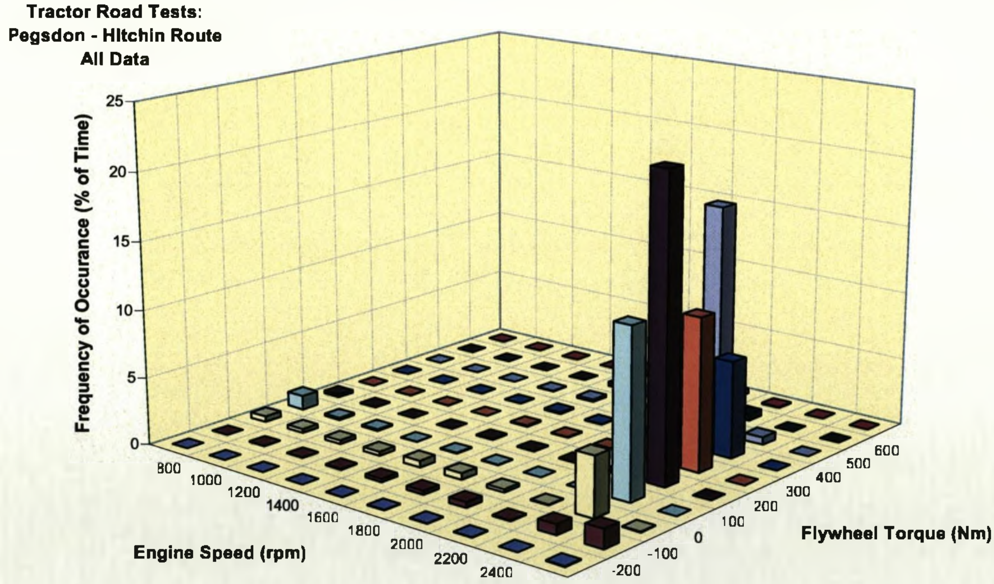



*Figure 5-34 – Flywheel torque – engine speed frequency distribution: Pegsdon - Hitchin section*

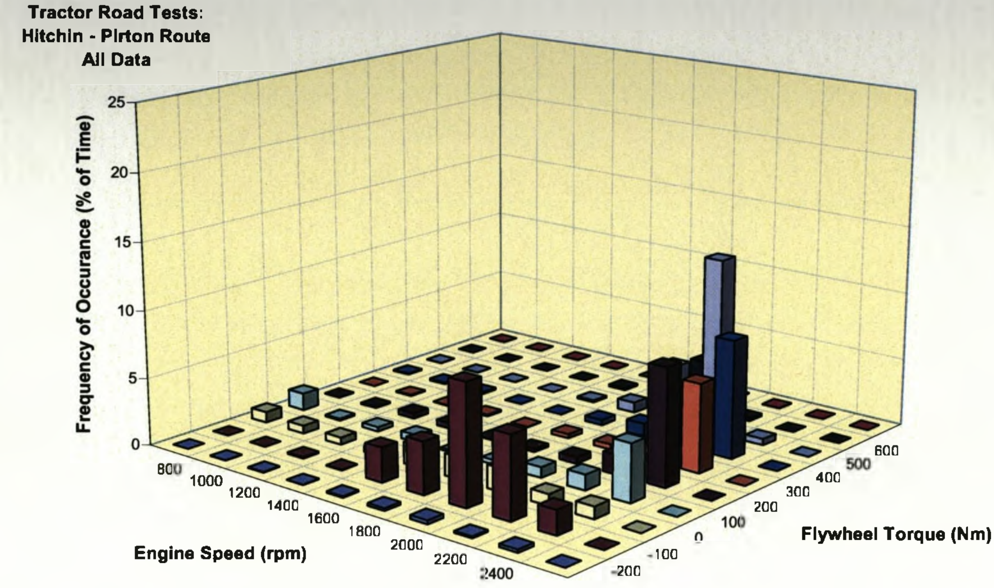



*Figure 5-35 – Flywheel torque – engine speed frequency distribution: Hitchin - Pirton section*

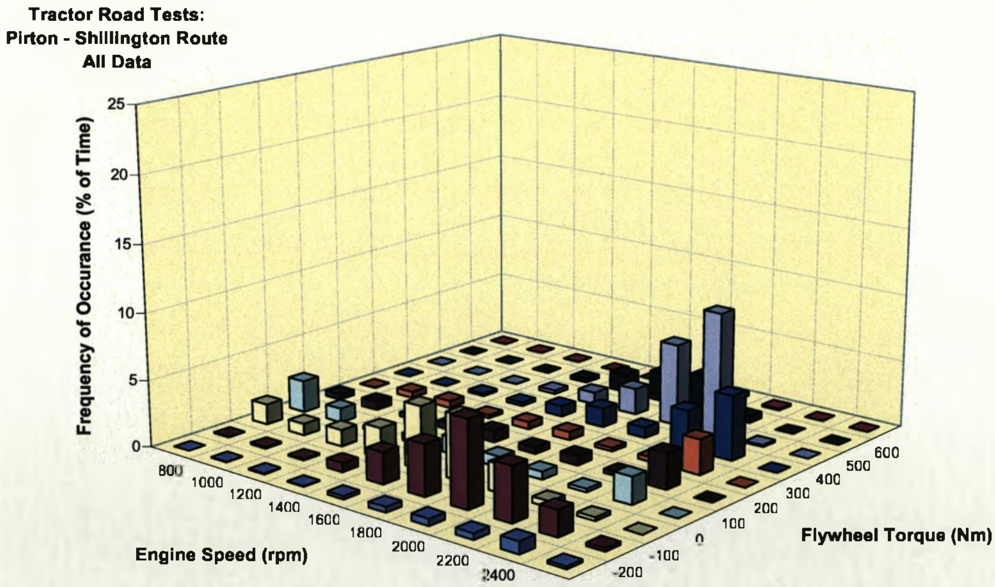



*Figure 5-36 – Flywheel torque – engine speed frequency distribution: Pirton - Shillington section*

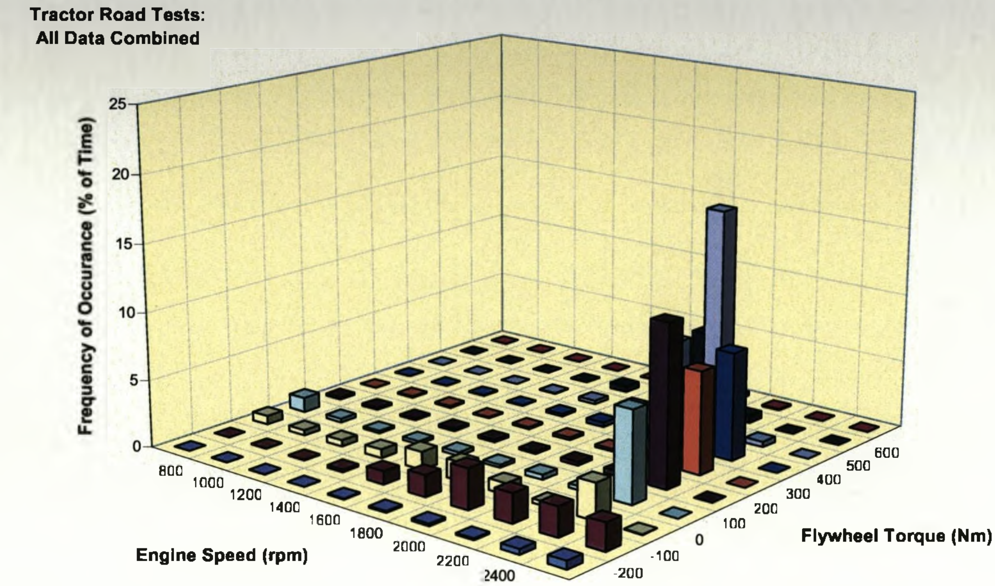



*Figure 5-37 – Flywheel torque - speed frequency distribution: combined data all routes*

| Characteristic | Section 1 (BTPD) | Section 2 (PDHN) | Section 3 (HNPT) | Section 4 (PTSH) | Overall |
|---|---|---|---|---|---|
| Engine speed below 1000rpm (% of time) | 1.1 | 2.3 | 3.4 | 7.6 | 3.0 |
| Engine speed above 2370rpm (% of time) | 11.0 | 4.1 | 0.3 | 1.1 | 4.9 |
| Engine speed above rated speed (% of time) | 41.4 | 60.2 | 31.5 | 15.7 | 42.3 |
| Engine speed between rated and max torque (% of time) | 56.4 | 36.4 | 61.9 | 68.8 | 52.1 |
| Flywheel torque negative (% of time) | 13.0 | 10.7 | 33.3 | 40.7 | 20.4 |
| Flywheel torque above 400Nm (% of time) | 48.6 | 32.2 | 24.0 | 26.5 | 34.4 |
| Flywheel torque above 500Nm (% of time) | 15.0 | 9.4 | 5.1 | 7.7 | 9.9 |
| Flywheel torque above 600Nm (% of time) | 0.7 | 0.6 | 0.1 | 0.6 | 0.5 |
| No throttle (% of time) | 9.7 | 3.0 | 9.8 | 21.0 | 9.1 |
| Maximum throttle (% of time) | 85.5 | 93.1 | 47.5 | 31.8 | 72.6 |
| Forward speed above 12km/h (% of time) | 94.3 | 95.5 | 92.5 | 86.5 | 93.1 |
| Forward speed above 30km/h (% of time) | 62.8 | 70.2 | 67.9 | 19.3 | 59.3 |
| Gear 13 or above (% of time) | 90.3 | 92.3 | 91.4 | 82 | 89.9 |
| Gear 16 (% of time) | 56.8 | 67.2 | 72.1 | 16.3 | 56.8 |
| Number of powershift gear changes | 32 | 28 | 18 | 30 | 108 |
| Number of other gearshifts | 4 | 4 | 2 | 4 | 16 |
| Distance (km) | 3.4 | 4.6 | 2.5 | 1.1 | 11.6 |
| total run time (min:sec) | 6:20 | 8:12 | 4:02 | 3:40 | 22:14 |
| Average speed (km/h) | 32.2 | 33.7 | 37.2 | 18.0 | 31.3 |

*Table 5.12 – Statistical Analysis of road transportation*

### 5.7.6 Discussion

The characteristics of each route are apparent from the summary data (see Table 5.12) and the frequency distributions. The good road quality but severe up and down gradients in the first section (BTPD) resulted in the test tractor operating along a narrow band at high engine speeds for the majority of the section, with the throttle fully open for 85% of the time (see Figure 5-33). The uphill gradients resulted in high flywheel torque loadings, above 400Nm for almost 50% of the time. The downhill parts resulted in negative flywheel torque for 13% of the time and the engine over-speeding for 11% of the time. The number of gradients also resulted in 36 gearshifts being made during the section.

The second section (PDHN) of the route, resulted in similar vehicle characteristics to the previous one, partly as it was a continuation of the same road. The long flat part of this section, where road surface was good, resulted in maximum vehicle forward speed being possible and the majority of the data occurring in the 2200rpm to 2400rpm engine speed range (see Figure 5-34). This relatively easy powertrain loading resulted in lower flywheel torque demand levels than the previous section. The test tractor was only loaded above 400Nm for 32% of the time, despite the early uphill gradients. These gradients resulted in a total of 32 gearshifts being made during the section.

The poor road surface during the third section (HNPT) resulted in a wider spread of engine torque-speed data (see Figure 5-35) as a consequence of not being able to operate the test tractor at maximum forward speed. During this section the throttle was fully open less than 50% of the time (see Table 5.12) and only 20 changes to transmission gear ratio were made. As engine braking was used to slow the vehicle during the downhill parts where road quality was poor, the tractor exhibited negative flywheel torque values for over 30% of the section.

During the final part of the route (PTSH) the test tractor displayed more torque-speed variation than previously (see Figure 5-36), as a result of the nature of the section. Four road junctions (one of which was hidden), together with a narrow village road, blind corners and parked cars resulted in extreme caution being required of the driver during this part of the route. In addition to the large torque-speed spread, lower forward speeds were typical (only above 30km/h for 12% of the time), and more gearshifts were made - 34 during a section lasting less than four minutes. The presence of a severe downhill gradient, but with a junction at the bottom, resulted in negative flywheel torques but without severe engine overspeed (1.1% of the time), because vehicle motion down the hill had to be controlled with the service brakes at all times.

The summary chart combining all the data (see Figure 5-37) shows the likely regions of operation if the test tractor was undertaking this route continuously - the high loading from some longer sections clearly influencing the overall engine torque-speed pattern. The total number of gearshifts shown (see Table 5.12) is merely a summation of the individual sections and as a result, if the section was driven continuously, it has been calculated that 78 powershift and 12 non-powered synchromesh transmission gear ratio changes would be necessary during the 20 minute journey. Admittedly using the 'autoshift' feature would reduce the operator burden, but may require driving style changes to be made. The severe loading, coupled with reduced engine speeds for some sections of the route, are a good indication as to the suitability of additional use for engine power boost to maintain progress for this application. Since this investigation was undertaken the manufacturers have indeed introduced power boost for the gears 13 to 16, i.e. road transport applications.

## 5.8 Transport – Wrest Park Drive

### 5.8.1 Objective

The objective of this work was to determine the nature of powertrain loading experienced during acceleration and deceleration of the test tractor and tractor-trailer combinations.

### 5.8.2 Experimental Design & Procedure

This part of the investigation was undertaken on the private drive to Wrest Park, avoiding the need to perform this type of experiment on a public highway. Acceleration and deceleration, with and without the laden trailer (ballasted as for Section 5.7) were considered. The drive has a slight down slope of approximately 1 in 100 from West to East (Ordnance Survey, 1988). As a result acceleration tests were conducted in both directions. The scenarios considered were all replicated three times. They were:

1. **East-West acceleration: tractor and trailer** – accelerating as quickly as possible from stationary, using maximum throttle position together with quick gear changes starting from gear 9 until maximum vehicle speed was achieved;
2. **East-West acceleration: tractor only** – as (1), but just the tractor (6,750kg) without the trailer;
3. **East-West acceleration: from gear 13** – as (2), but starting from gear 13 rather than gear 9;
4. **West-East acceleration: tractor and trailer** – as (1), but from the opposite direction;
5. **West-East acceleration: tractor only** – as (4), but without the trailer;
6. **West-East deceleration: tractor and trailer** – deceleration commenced by changing throttle demand to 0% instantaneously, followed by downshifting gears from 16 to 9 as quickly as possible without over-speeding the engine; and
7. **West-East deceleration: tractor only** – as (6), but without the trailer.

During the acceleration and deceleration trials, the data acquisition rate was increased to 50Hz in order to ensure an accurate representation of rapidly changing parameter values.

### 5.8.3 Parameters

The same parameters (both acquired and derived) were considered during the drive work, as were used for the road transport investigation (see Section 5.7.4). In addition, the engine droop mode ($\delta_I$) was also recorded.

### 5.8.4 Results

Example data time histories arising from Tests 1, 2, 3, 6 and 7, depicting the variation of engine speed, flywheel torque, transmission gear ratio and forward speed, are presented in Figures 5-38, 5-39 and 5-40.

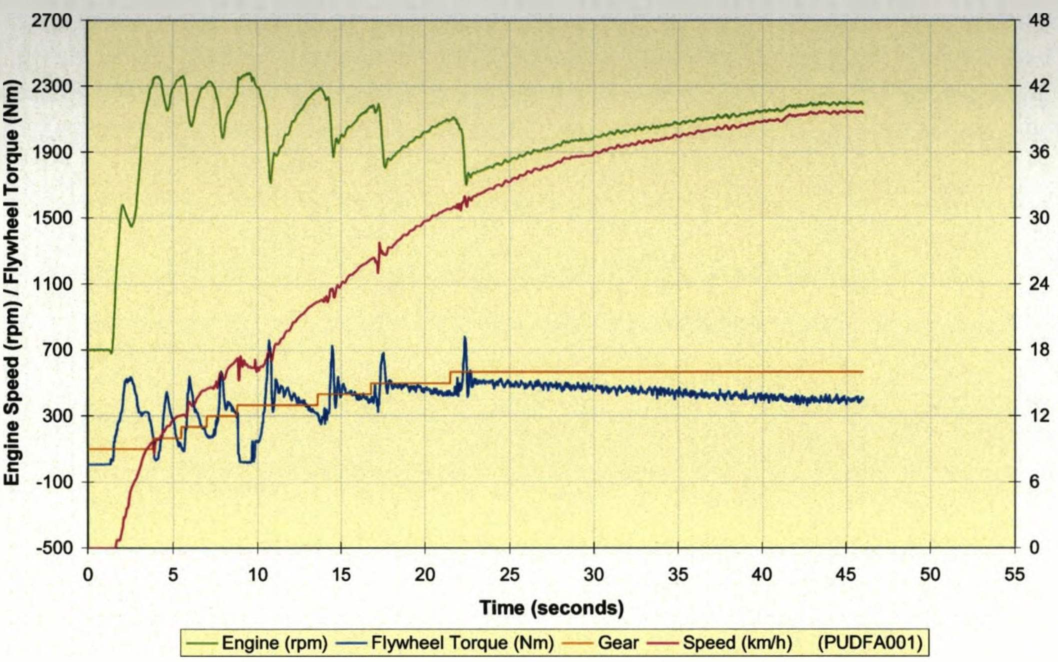



*Figure 5-38 – East-West acceleration: tractor and trailer (1)*

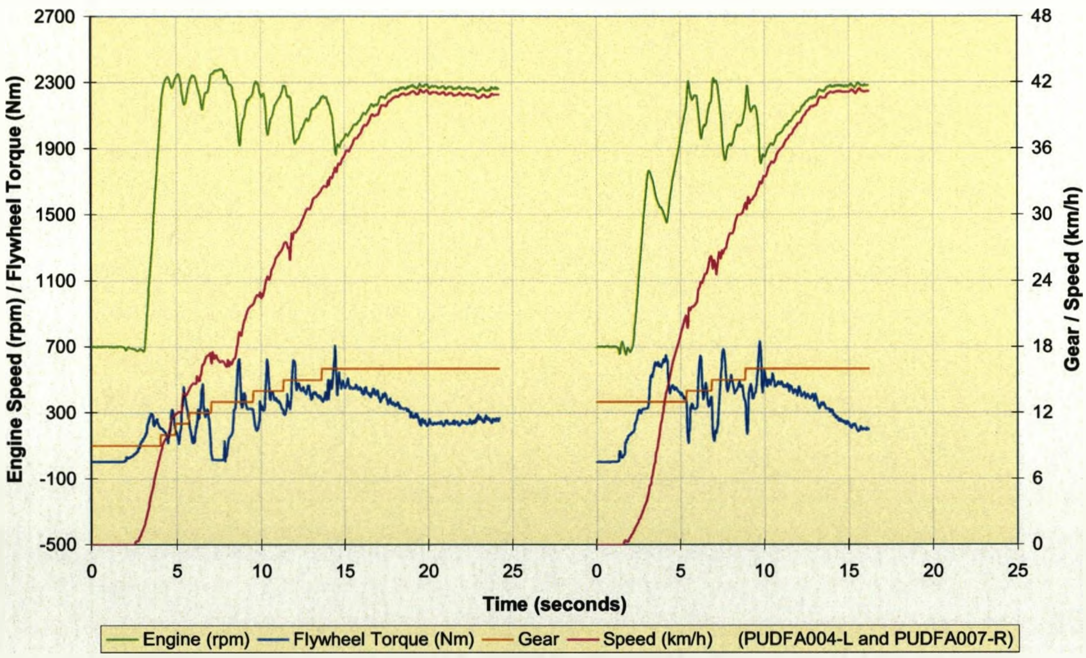



*Figure 5-39 – East-West acceleration (Tractor only) - Left: from G9 (2); Right: from G13 (3)*

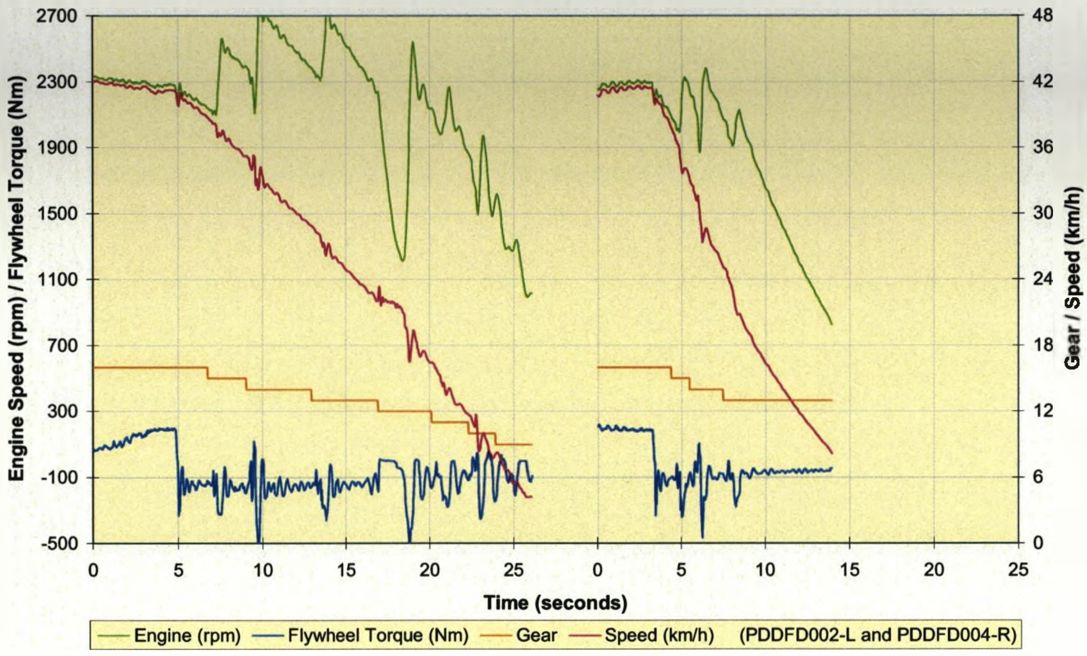



*Figure 5-40 – West-East deceleration - Left: tractor and trailer (6); Right: tractor only (7)*

### 5.8.5 Discussion

It was found there was little noticeable difference in vehicle performance between undertaking the trials in each direction along the drive. Therefore only an example of the most difficult operational situation has been presented, i.e. East-West whilst accelerating, and West-East whilst decelerating.

During the trailer acceleration test, the vehicle transmission upper range was selected and then gearshifts were made as quickly as possible, with full throttle setting maintained, in order to accelerate the tractor and trailer as quickly as possible (see Figure 5-38). It can be seen that, despite a heavily-laden trailer, as soon as the clutch was released the engine quickly accelerated the vehicle and a gearshift was quickly required. This suggests that despite the tractor defaulting to gear 9 when the upper range is selected, in reality a higher gear is probably more appropriate for pulling away on a level surface despite the high train weight. Following the shift into gear 10, a succession of quick shifts to gears 11 then 12 were possible. Again, perhaps not all were necessary. It was then necessary to make the non-powershift change to gear 13. This slow gear change temporarily reduced engine speed and forward speed and resulted in flywheel torque output temporarily becoming negative, which was felt by the operator. As the tractor forward speed increased, the laden trailer slowed the rate of acceleration. The maximum possible forward speed was reached around 45 seconds. The smoothness of the different gearshifts can be seen (see Figure 5-38), particularly the difference between a single-swap shift and a double-swap shift. The 13-14 and 15-16 shifts are both single-swap shifts, their smoothness being evident by the fact that the vehicle forward speed trajectory is only slightly disturbed (at approximately 14 and 22 seconds). In contrast the 14-15 double-swap shift causes the vehicle forward speed to oscillate more forcefully (at approximately 17 seconds). This 'roughness' of gearshift was noticeable to the operator and is an undesirable characteristic, although it is obviously better than the non-powershift gear change.

Figure 5-39 shows the acceleration profile for the tractor only. On the left the tractor was accelerated from gear 9 onwards, on the right the tractor was accelerated from gear 13. When started from gear 13, the tractor was able to reach 40km/h after 11.2 seconds, 4 seconds less than when starting from gear 9. Starting in gear 13 rather than the default gear 9 reduced the number of gearshifts required and excluded the need for the non-powershift gear change. However, it would be unlikely that the tractor could accelerate from a standstill in gear 13 under laden conditions, or if the vehicle was on a slope.

Figure 5-40 shows the deceleration profile for the tractor, both with (left side) and without (right side) a laden trailer. The combined mass of the tractor and trailer, and therefore its higher momentum, together with just the use of transmission gears and engine braking, required longer to reduce forward speed than the tractor only (an additional 10 seconds to slow the tractor and trailer to 8km/h). Tractor and trailer momentum also resulted in predominantly negative flywheel torque during deceleration, together with engine overspeeds and more pronounced flywheel torque fluctuations during downshifts. The poor shift characteristics of the double-swap shifts (15-14 at 10 seconds and 11-10 at 23 seconds) is again demonstrated by the higher flywheel torque fluctuations during these shifts, as the various powershift clutches engage and disengage to effect the gear change.

Perhaps more concerning are the characteristics during the 13-12 non-powershift change, which resulted in a harsh engine speed change and a large negative torque of 500Nm. This is due to the positive drive between the engine and the wheels being broken and therefore allowing the engine to decelerate (as there is no throttle setting). As the clutches are pressurised and drive restored, the tractor wheel speed then accelerates the engine again, causing the large engine speed fluctuation. This could be avoided with the application of a positive engine throttle setting during the shift, or perhaps the incorporation of a more appropriate aid to assist in deceleration, such as an engine exhaust braking system. This shift also took approximately three seconds from when the shift was requested until vehicle deceleration resumed. If this shift had been effected on substantial gradient, such as those encountered during the road investigation, the torque-speed fluctuations could have been more severe, increasing the potential to damage engine and/or driveline components.

## 5.9 Overall Summary

This part of the investigation has provided an understanding of the characteristics of the test tractor powertrain, the effects of different tractor-implement parameters and the effects of soil variation. Extensive data has been generated for use in the development and validation of the powertrain model and for further work with the axle dynamometer. The investigation has identified a number of powertrain characteristics, some worthy of further investigation and improvement, as well as confirming the limited impact others have upon tractor operation.

During mouldboard ploughing of the clay soil, draught and power requirements were higher than for sandy soil with identical tractor-implement configurations. This limited the operational range possible. Soil strength varied more in the clay soil, causing increased engine torque-speed fluctuations, often masking the effects of the load transients investigated. Powertrain loads were directly influenced by the working depth of the plough, but the furrow width influence was not as clear.

The sandy soil provided higher powertrain loading for identical tractor-implement configurations during power harrowing. Working depth directly influenced the power requirement, both from the P.T.O. and the traction drivelines. The power distribution between the two driveline paths was influenced by forward speed and working depth.

Road transportation characteristics were directly influenced by the road conditions and terrain. The high engine loads and reduced forward speed from some routes demonstrated a potential further application for the engine power boost feature.

The gearshift characteristics of this powertrain were influenced by the number of clutches being engaged and disengaged to make a given gear change. Where two pairs of clutches were changed (swapped), the shift was more pronounced and load fluctuations were higher than during the more common single-swap shifts. The single-swap shift characteristics were found to be similar regardless of the actual gear change, although the magnitude of torque and engine speed profiles were increased in the higher gears.

The type of operation being undertaken influenced the shift profile, primarily as a result of the different power demands on the powertrain. The high draught forces during ploughing resulted in severe torque 'spikes' during shifts. Power harrow shifts were relatively smooth as a result of the lower draught forces. During transportation the high tractor and trailer momentum impacted on the gearshifts, particularly during downshifts, where engine torque-speed fluctuations were severe. The non-powershift gear changes, encountered during transportation, were most noticeable.

# 6 Axle Dynamometer Experiments

## 6.1 Introduction

The field and road investigation (Section 5) generated a large volume of data pertaining to the steady state loads experienced, together with the effects of typical load transients. A number of interesting trends warranted further investigation, namely:

- is it possible to replicate steady state field loads in a controlled manner?
- how do vehicle driveline losses vary under different loading regimes?
- is driveline inertia sufficient to influence vehicle acceleration?

This part of the investigation was undertaken using the Silsoe Research Institute (SRI) axle dynamometer, which required overhauling and calibrating to undertake the desired investigations.

## 6.2 Dynamometer Design and Operation

The SRI axle dynamometer (see Figure 6-1) was installed to enable the torque variations, which occur at the axle ends of a vehicle during typical operations, to be replicated under repeatable laboratory conditions. In addition, the facility provided a means to determine tractor axle power to ISO standard 789/7 (International Standards Organisation, 1991), as a potential substitute for drawbar performance testing on standardised test tracks. Designed with agricultural applications in mind, the rig is best suited to low speed, high torque operation (Tinker *et al*, 1991). The rig consists of four computer-controlled low-inertia dynamometer units each based upon water-cooled, air-operated friction disc brakes: these units are coupled at each axle end of the vehicle under test in place of the wheels, to allow the weight of the vehicle to be supported by the units and the axle bearings to be loaded in the normal way. Individual load cells are used to measure the braking torque at each dynamometer via a moment arm (see Figure 6-2).

*Figure 6-1 – Test tractor mounted on the axle dynamometer*

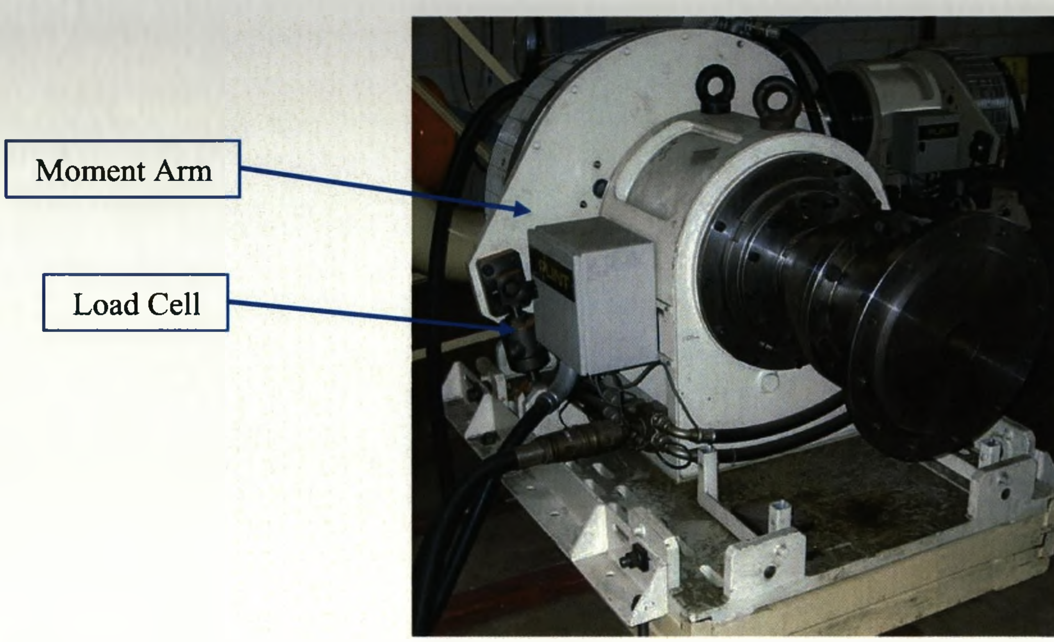



*Figure 6-2 – Dynamometer unit torque measurement via a moment arm and load cell*

In normal closed loop mode (see Figure 6-3) the operator pre-programmes the individual dynamometer torque demand at each time step in a test sequence file with the variations defined as steps or ramps. The maximum permissible frequency of torque demands is 1Hz. The computer control software then uses this torque demand and an inbuilt digital proportional-plus-integral-plus-derivative (PID) feedback controller, together with pressure prediction software, to control a pneumatic valve on the brake unit. The valve, featuring proportional-plus-integral (PI) control, then applies an air pressure to the friction brakes to provide the desired braking torque.

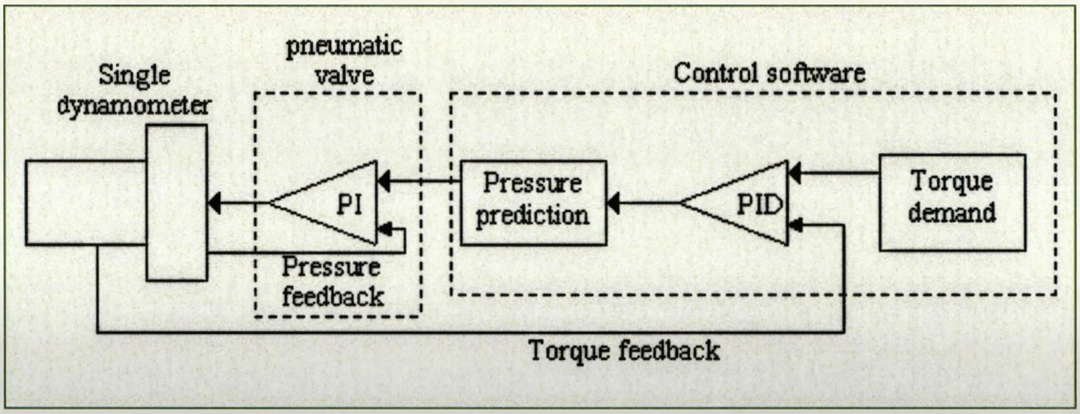



*Figure 6-3 – Axle dynamometer closed loop control diagram*

The pressure prediction algorithm allows a simple form of open loop control to operate on the brakes to provide the majority of the controlling effort required to give a desired brake torque. This permits the torque error, which needs to be dealt with by the PID algorithm, to be small and therefore reduces the required proportional gain and the integral time thereby improving the system response without introducing the instability characteristics normally associated with responsive PID control systems.

These values are used in the PID (series) feedback loop, displayed on the operator control screen (see Figure 6-4) and are recorded to an output file together with individual wheel speeds generated from tacho-generators. All calibration settings for the dynamometer are contained within a test definition file. The test definition, test sequence and output files are utilised within the bespoke dynamometer software to control, display and record data from the dynamometer.

Further detailed descriptions of the dynamometer design and operation are discussed by Tinker *et al* (1991). To reduce the workload in recommissioning and calibrating the dynamometer and to simplify the experimental programme, this part of the investigation was conducted with the rear axle brake units only.

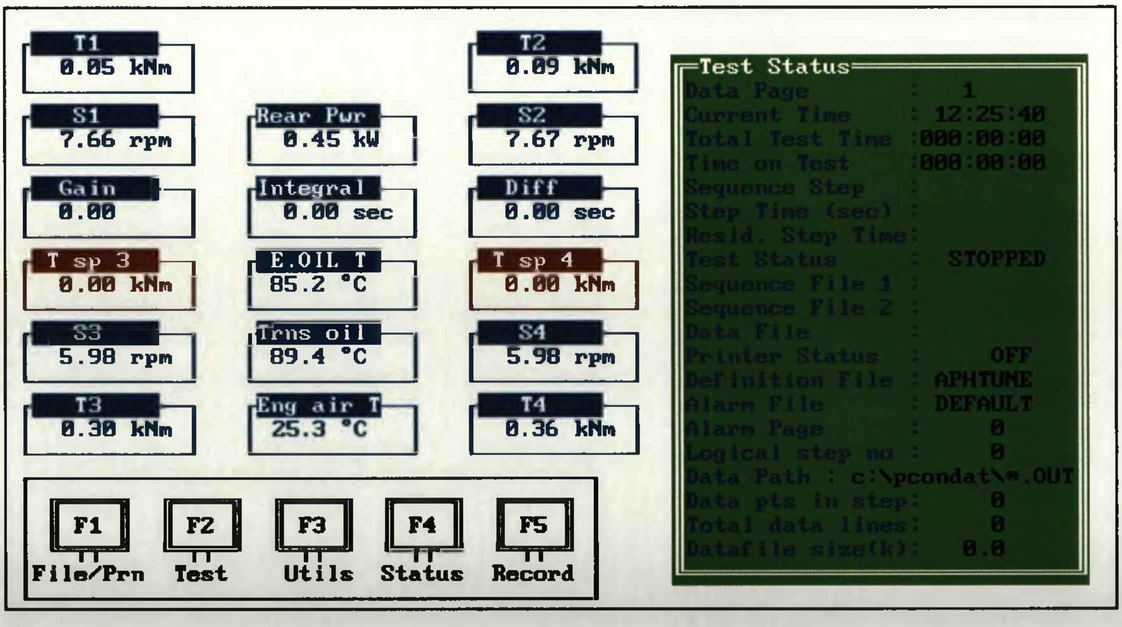



*Figure 6-4 – Axle dynamometer control screen*

## 6.3 Dynamometer Recommissioning and Calibration

### 6.3.1 Recommissioning

Prior to commencing this part of the investigation, and as a result of a period of inactivity, a thorough inspection and service of the brakes, pneumatic valves and some computer hardware was carried out prior to in-situ calibration. This ensured satisfactory operation of the dynamometer.

### 6.3.2 Load Cell Calibration

The individual dynamometer unit load cells were calibrated statically using a bespoke test rig (see Figure 6-5) prior to attaching the tractor to the dynamometer. Each dynamometer unit was bolted to the floor and a calibration arm attached, giving an overall horizontal distance between the brake centre and the hydraulic ram of one metre. This allowed the direct translation between the applied force and torque. The calibration frame was also bolted to the ground and a hydraulic ram and proving ring connected between the frame and the arm. A vertical, upwards force was applied by retracting the hydraulic ram, thereby loading the dynamometer unit and load cell in the normal operational manner. The actual applied force was measured with a standard 100kN proving ring and this was used to calibrate the load cell output gain in the controller software. The final offset was adjusted with the moment arm removed from the brake unit.

The final calibration was checked three times for brake torques between 0kNm and 40kNm. R.M.S. error was 0.118kNm and 0.136kNm for the left and the right rear dynamometer units respectively. This was reduced to 0.040kNm and 0.079kNm when a maximum torque of 30kNm was considered. This was still above the maximum braking torque achieved during practical experiments and is an error of less than 0.3% of full scale (see Appendix A5.1).

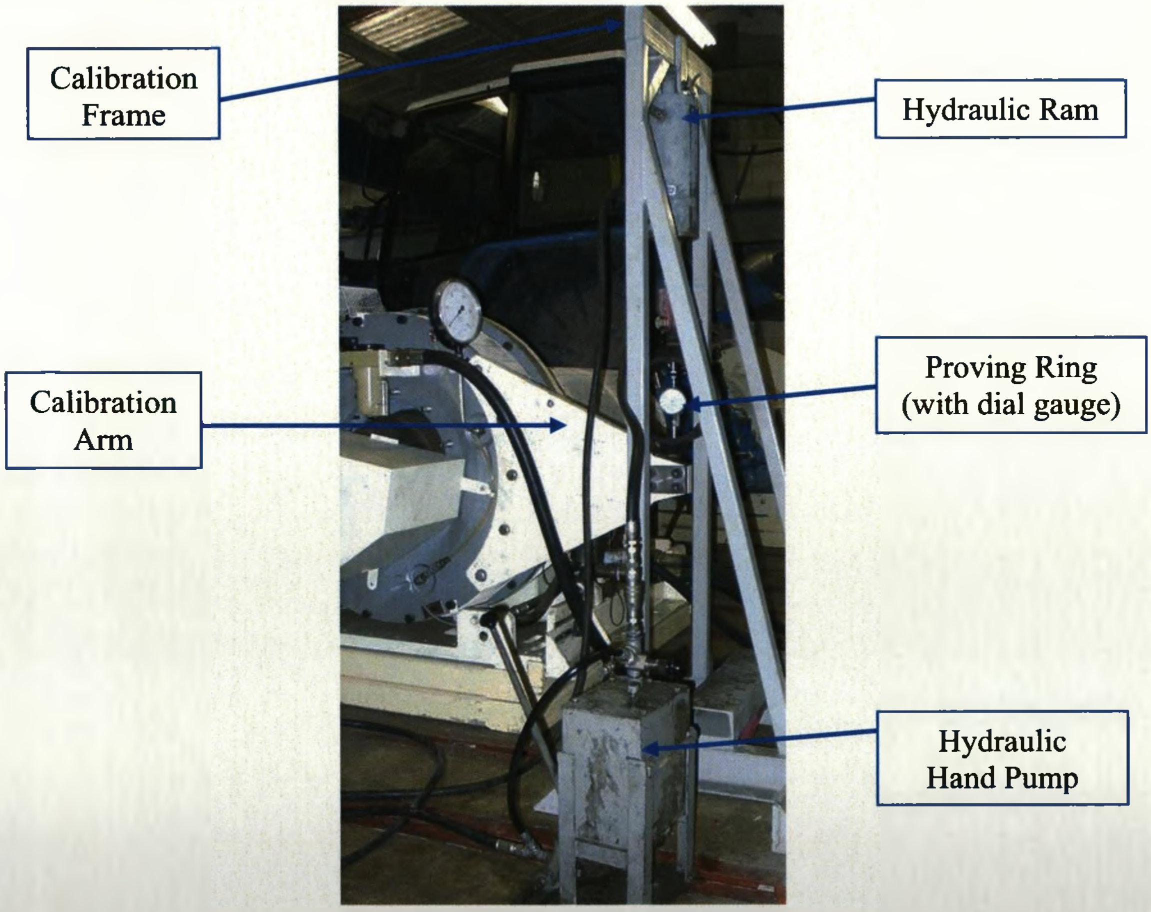



*Figure 6-5 – Load cell calibration equipment and setup*

### 6.3.3 Pressure Predictor Calibration

The pressure predictor algorithm required a characteristic slope and offset for each dynamometer unit to be quantified in the test definition file to optimise performance prior to setting the elements of the PID algorithm. The calibration procedure was performed with the tractor fitted to the dynamometer, to allow normal brake operation to occur. The settings and process were carried out as recommended in the guidance notes supplied by the manufacturers of the dynamometer system (Plint & Partners Ltd, 1997). The predictor slope was set to unity and the offset to zero. The feedback controllers were set to give a negligible proportional control with no integral or derivative action, thus forcing the system to operate in a closed loop control mode whilst actually providing little corrective action: i.e. essentially an open loop response.

With the tractor wheels rotating at around 20rpm at full throttle, the torque setpoints for each brake were ramped in 1kNm increments every twenty seconds from zero to 15kNm and back to zero. The mean wheel torque was recorded, following a delay to allow each torque increase to take effect. The resultant data showed both a non-linear response and the presence of hysteresis (see Figure 6-6).

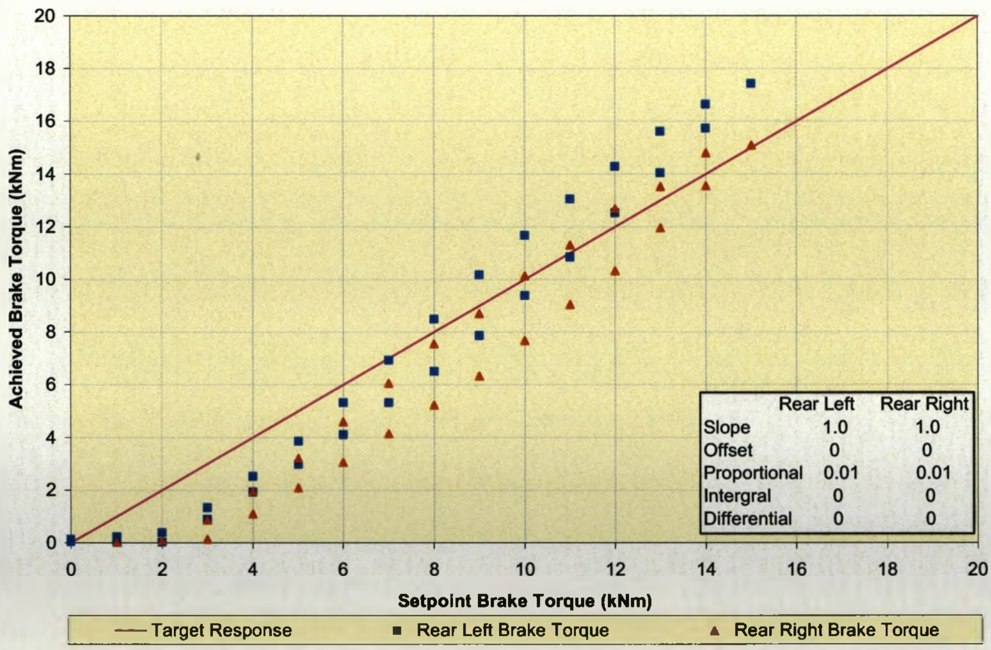



*Figure 6-6 – Dynamometer steady state response prior to pressure predictor calibration*

Using the mean of the stepping-up and down values for each non-zero set point, the slope was calculated for each brake. This was then adjusted in the test definition file and an iterative process used to further refine it until the characteristic slope was obtained to give the optimum response, albeit with some offset. The mean offsets were then calculated and adjusted in the test definition file to move each predictor response as close as possible to the ideal response. The optimum steady state response produced a far better response from both dynamometer units, as shown in Figure 6-7. Through optimising the pressure predictor, the R.M.S. error for the rear left dynamometer unit reduced from 1.56kNm to 0.3kNm and the rear right dynamometer unit R.M.S. error reduced from 1.76kNm to 0.53kNm.

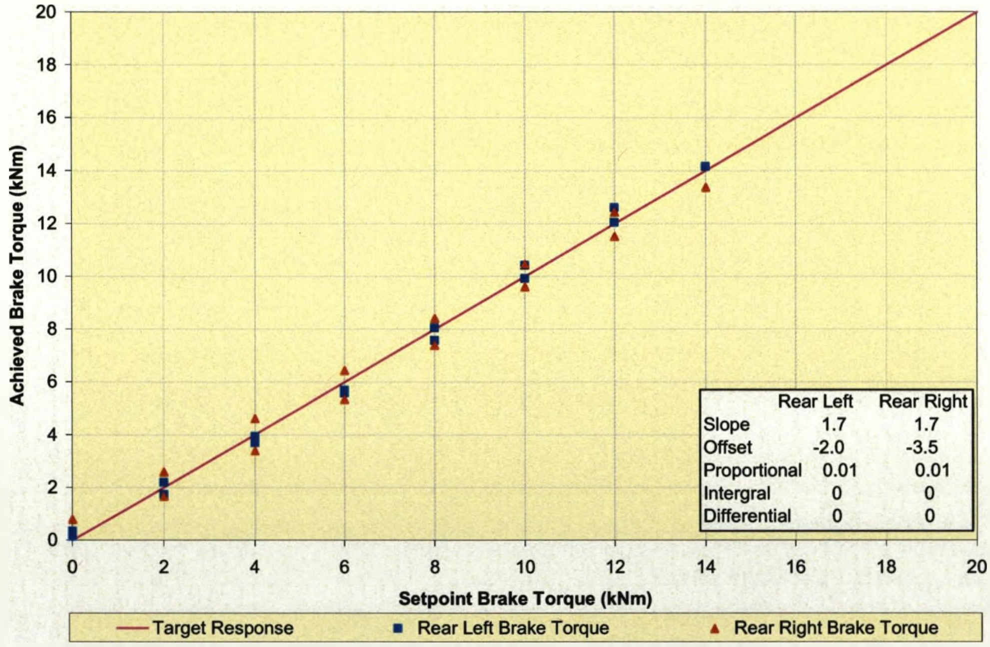



*Figure 6-7 – Dynamometer steady state response following pressure predictor optimisation*

## 6.3.4 PID Tuning

### *6.3.4.1 Open Loop Response*

For PID tuning the test sequence file was adjusted to give each dynamometer unit a step-change in torque from 8kNm to 10kNm and back to 8kNm, before repeating the sequence. A 2kNm step was the upper limit of magnitude of any instantaneous step change in equivalent axle torque seen during field experiments. The dynamometer was run in open loop control mode, i.e. the PID controller was disabled to determine the dynamometer response. The resultant performance for each dynamometer unit is shown in Figure 6-8. Whilst the action of the pressure predictors do indeed provide a basic control mechanism, it can be seen the performance of the dynamometer is not satisfactory, in either accurately maintaining the setpoint torque, or the dynamometer's response to a step change in the required torque. From this result it was deemed necessary to optimise the PID controller to give some form of corrective action to improve the overall dynamometer operation.

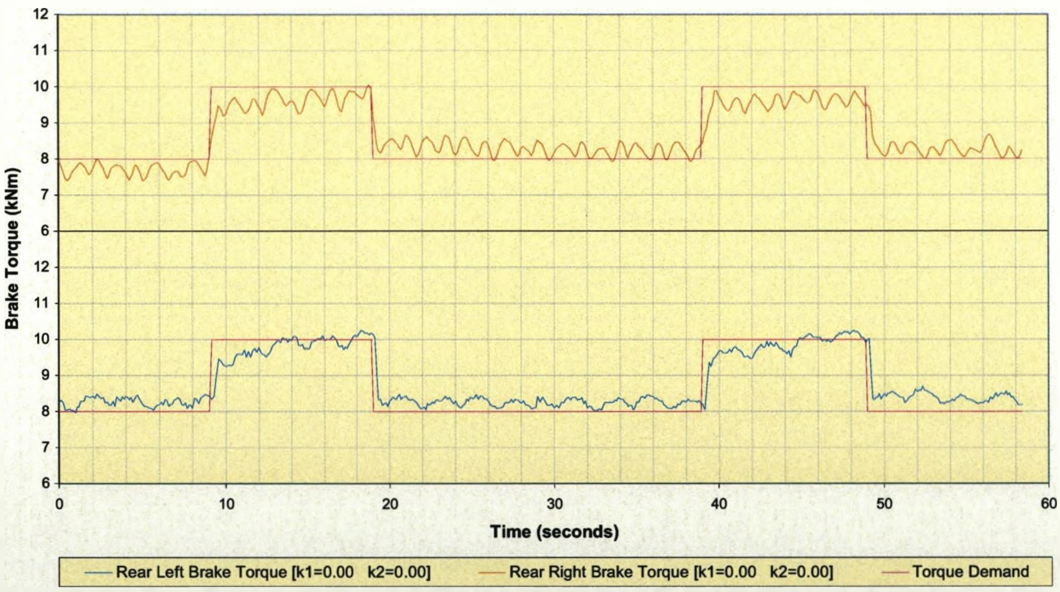



*Figure 6-8 – Open loop dynamic dynamometer response to step inputs, prior to optimisation*

### 6.3.4.2 PID Tuning Theory

Within the axle dynamometer control software, the PID control algorithm forms part of a series compensation control loop, as previously shown in Figure 6-3. For this type of control loop the standard controller output, m(t), is defined in the time domain according to Equation 6-1.

$$m(t) = k_1.e(t) + k_2 \int e(t).dt + k_3 \frac{d.e(t)}{dt} \qquad \textit{Equation 6-1}$$

e(t) is the error signal from the feedback loop and $k_1$, $k_2$ and $k_3$ are the numerical constants associated with the proportional, integral and derivative components respectively.

The proportional component returns a signal proportional (by a factor of $k_1$) to the error and is the basic controlling mechanism. The requirement of most control systems, including that of the axle dynamometer, is to minimise the steady state offset error. Increasing the proportional gain will help attain this aim, although often at the expense of unstable transient response, an undesirable characteristic in any system

(Schwarzenback and Gill, 1992). Adding a signal proportional (by a factor of $k_2$) to the time integral of the error can minimise steady state offset whilst maintaining satisfactory transient performance. The derivative term ($k_3$) contributes an anticipatory control action, as the term is most effective at modifying controller output when the error is rapidly changing.

The PID controller defined in Equation 6-1 can be expressed as a transfer function in the Laplace domain by Equation 6-2:

$$G_C(s) = k_1 + k_2 / s + k_3 s \qquad \textit{Equation 6-2}$$

more commonly presented in the form of Equation 6-3:

$$G_C(s) = k_c\left(1 + \frac{1}{T_i s} + T_d s\right) \qquad \textit{Equation 6-3}$$

Where $k_1 = k_c$, $k_2 = k_c/T_i$ and $k_3 = k_c T_d$. The format of Equation 6-3 was used by Ziegler and Nichols (1942) who took their results of empirical tests on a wide variety of control equipment to provide a simple rule of thumb procedure for estimating the values of $k_c$, $T_i$ and $T_d$, therefore allowing the determination of $k_1$, $k_2$ and $k_3$. This method is still the most widely used and reported method of obtaining an initial calibration of a PID controller (Badreddine *et al*, 2001). The aim is to determine experimentally the limiting condition of stability for a closed loop system under pure proportional control. The plant is subject to a step change and the value of $k_c$ is increased at each iteration until the step change results in a continual oscillation of m(t). This final proportional setting ($k_{crit}$) is then used, along with the time period of oscillation ($P_{crit}$) to determine the controller settings as outlined in Table 6.1.

*Table 6.1 – Ziegler-Nichols suggested controller settings for a closed loop calibration*

| | $k_c$ | $T_I$ | $T_d$ |
|---|---|---|---|
| P control | $0.5k_{crit}$ | - | - |
| P+I control | $0.45k_{crit}$ | $0.83P_{crit}$ | - |
| P+I+D control | $0.6k_{crit}$ | $0.5P_{crit}$ | $0.125P_{crit}$ |

### *6.3.4.3 Ziegler Nichols Closed Loop Tuning*

In the dynamometer calibration, a single step change in brake torque between 8kNm and 10kNm was made whilst increasing the value of $k_c$ in the test definition file. The response at the limit of stability for each rear dynamometer unit is shown in Figure 6-9. $k_{crit}$ for the rear left unit was determined as 1.25 and 1.4 for the rear right unit. The time period for each oscillation ($P_{crit}$) over a 5 second period was measured. The mean period of oscillation was 0.29 seconds for the rear left and 0.25 seconds for the rear right dynamometer unit.

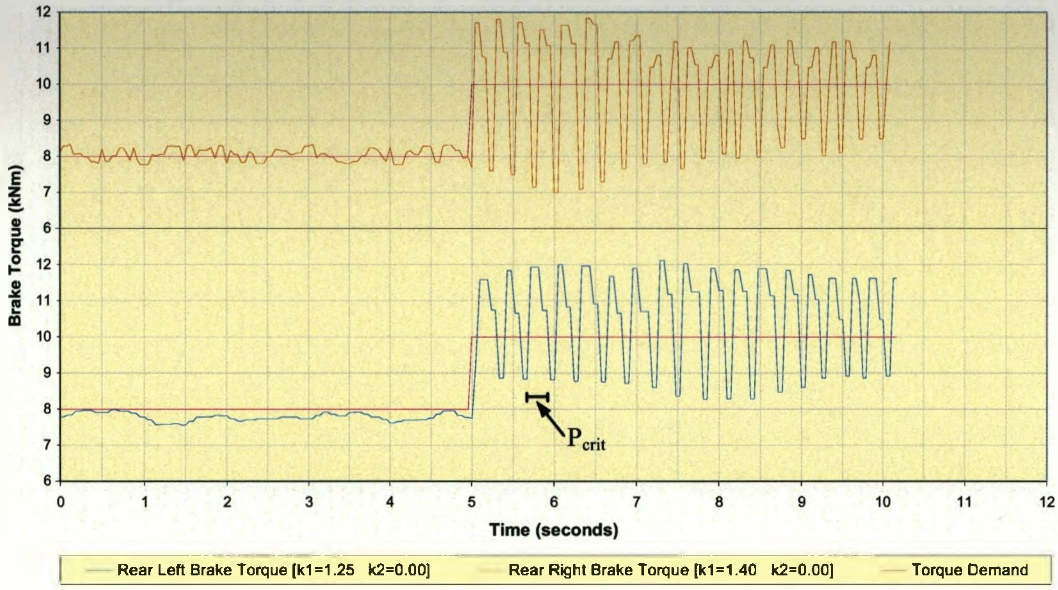



*Figure 6-9 – Oscillatory dynamometer response to a step input under proportional control at the limit of stability for Ziegler-Nichols closed loop control optimisation*

Using the values obtained for $k_{crit}$ and $P_{crit}$ it was then possible to calculate the desired settings for PI control and for PID control for each of the rear dynamometer units. The calculated parameters are shown in Table 6.2.

*Table 6.2 – PID parameters calculated according to the Ziegler-Nichols closed loop method*

| | $k_{crit}$ | $P_{crit}$ | $k_c$ ($k_1$) | $T_i$ | $T_d$ | $k_2$ | $k_3$ |
|---|---|---|---|---|---|---|---|
| PI (Left) | 1.25 | 0.29 | 0.56 | 0.24 | - | 2.33 | - |
| PID (Left) | 1.25 | 0.29 | 0.75 | 0.15 | 0.04 | 5.00 | 0.03 |
| PI (Right) | 1.4 | 0.25 | 0.63 | 0.21 | - | 3.00 | - |
| PID (Right) | 1.4 | 0.25 | 0.84 | 0.13 | 0.03 | 6.46 | 0.03 |

The axle dynamometer software required both proportional and differential settings in the format of $k_1$ (dimensionless) and $k_3$ (seconds) respectively. However, the integral term must be specified in seconds, i.e. $1/k_2$. The controller then inverts the value.

Following the calculation of the optimised parameters, PID control adhering to the recommended values in Table 6.2 was implemented. Even with this small derivative action, or a subsequently reduced term, the dynamometer unit output was highly unstable, necessitating the test to be aborted when the step change was encountered. Following consultation with the dynamometer manufacturer, all future work was undertaken with PI control, i.e. $k_3$=0.

With the dynamometer subject to a similar test regime as used for the open loop test (see Section 6.3.4.1), the response of each of the rear dynamometer units with the PI settings prescribed in Table 6.2 was determined (see Figure 6-10). As can be seen, the dynamometer response was now much better than the open loop test, not only in tracking the steady state torque request but also in the response to a torque step change.

Figure 6-10 also shows controller overshoot occurred at each torque step change, most likely as a result of the integral time being too short ($k_2$ being too large). The Ziegler-Nichols method, although used as the basis for many controller tuning exercises, is only ever intended to give a base setting which can then be fine tuned by examining the controller output (Badreddine *et al*, 2001 and Yang, 2004).

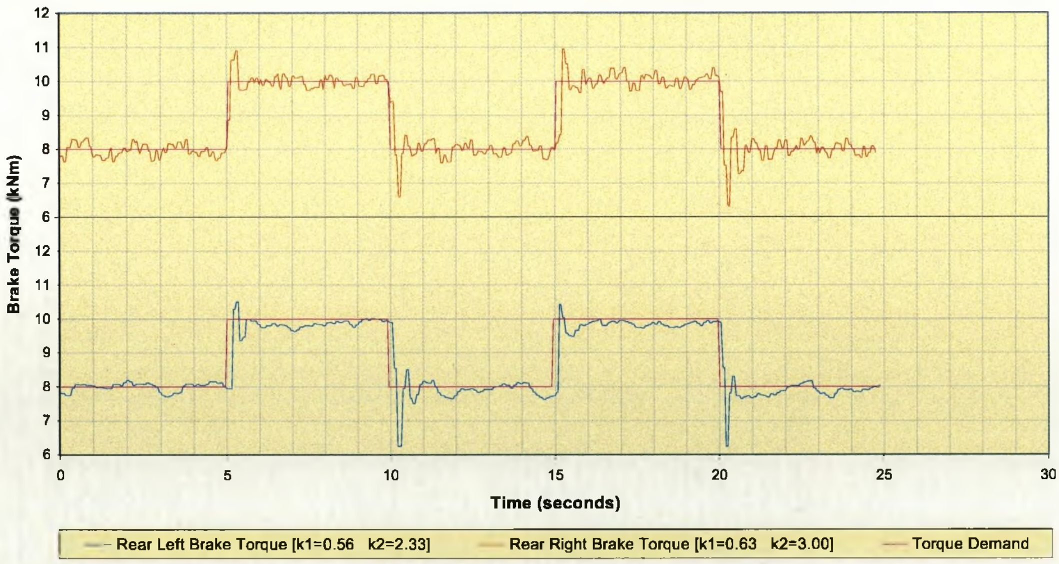



*Figure 6-10 – Dynamometer response to step inputs with initial Ziegler-Nichols proportional & integral control algorithm settings*

### *6.3.4.4 Final Refined Controller Settings*

Further refinement was carried out adjusting both the proportional and integral terms to optimise both the steady state performance of the dynamometer and the response to a torque step change. Many settings were tried around those identified in Section 6.3.4.3 before the optimal solutions were found. Table 6.3 shows the finalised values and the improvement in R.M.S. torque error. These finalised settings were tested with the dynamometer operating over a larger torque span. The resultant performances (see Figures 6-11 and 6-12) were deemed to be acceptable for the requirements of this investigation.

*Table 6.3 – Final PID algorithm values for the dynamometer controller*

| | $k_1$ | $k_2$ | $1/k_2$ | $k_3$ | rms improvement* |
|---|---|---|---|---|---|
| Rear Left | 0.60 | 0.667 | 1.5 | 0 | 71% (to 0.09kN) |
| Rear Right | 0.50 | 0.710 | 1.4 | 0 | 65% (to 0.13kN) |
| * from loop open settings, excluding the first 0.5 seconds after a step change | | | | | |

In comparing the improvement in R.M.S. torque error, the initial 0.5 seconds after a step change was ignored because the final test has three times as many torque setting changes as the open loop test, thereby making it an unfair comparison.

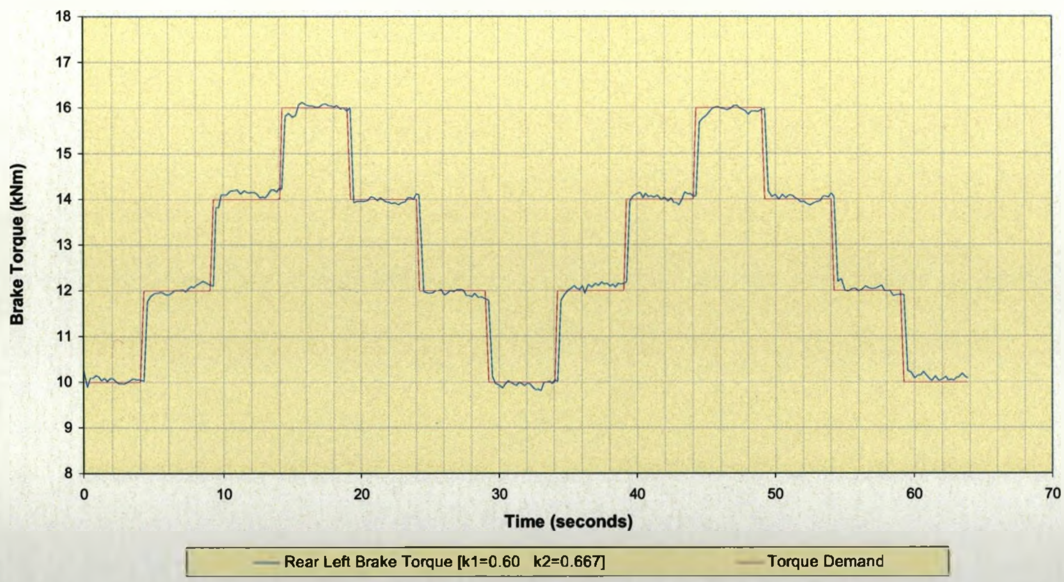



*Figure 6-11 – Dynamometer rear left unit response to step inputs with the final proportional and integral control algorithm settings*

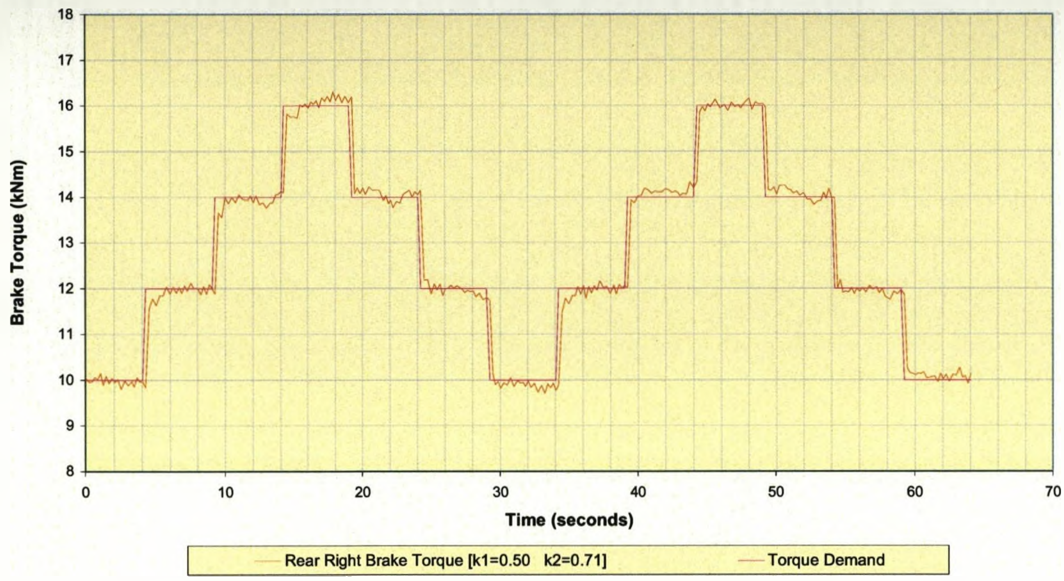



*Figure 6-12 – Dynamometer rear right unit response to step inputs with the final proportional and integral control algorithm settings*

### 6.3.5 Additional Dynamometer Instrumentation

Axle speed was measured by inbuilt tacho-generators within each brake unit, designed to work accurately to below 5rpm. A calibration check on these sensors with a handheld tachometer (*Superb Instrumentation 8300 s/n 8837*) across a range up to 80rpm returned a R.M.S. error of 0.38rpm and 0.48rpm for the rear left and right dynamometer units respectively.

A number of platinum resistance temperature probes were used to monitor tractor oil, air and coolant temperatures from the dynamometer control room. The probes were calibrated against a platinum resistance probe (*Datron S511/3/R100 s/n 189*). All returned a R.M.S. error less than 1°C in a working range from 10°C to 120°C.

## 6.4 Replication of Field Loading using the Axle Dynamometer

### 6.4.1 Background

Whilst one of the original objectives when the axle dynamometer was installed was to replicate field loads, organisational changes resulted in limited operation after commissioning. Work was limited to axle power determination (Brighton, 1997), comparisons with test track data (Mellor, 1999) and oil durability studies (Layton, 1999). In these cases the control software was bypassed and open loop, potentiometer-based dynamometer unit torque control unit was utilised. The were no examples in the literature to suggest that attempts have been made to replicate real-time field data prior to this investigation, although simulated axle torque loads were applied with a dynamometer as a function of time, for a tracked military vehicle (Schmid *et al*, 1988). Whelpley (1973) utilised measured field data and used the resulting frequency histogram to develop powertrain durability tests on a dynamometer.

### 6.4.2 Objectives

The objectives of this part of the experimental programme were:

1. to determine the ability of the axle dynamometer to accurately replicate field loading patterns; and
2. to devise a suitable methodology to transform measured flywheel torque from the field to an axle torque test sequence file for use with the dynamometer.

### 6.4.3 Experimental Design and Procedure

A ploughing dataset from the sandy soil experiments was chosen with the tractor operating in gear 5 with a furrow width of 356mm. The flywheel torque data was averaged to 1Hz, the maximum permissible rate for the dynamometer control system, (see Figure 6-13) and used to calculate equal torques ($T_L$ and $T_R$) for each of the rear dynamometer units, the combination of this giving the 'equivalent' axle torque ($T_{EA}$) to the measured flywheel torque. These equivalent torques purely took account of the total driveline ratio ($r_d$) and the conversion from the field measured torque (Nm) to the test sequence file requirement (kNm).

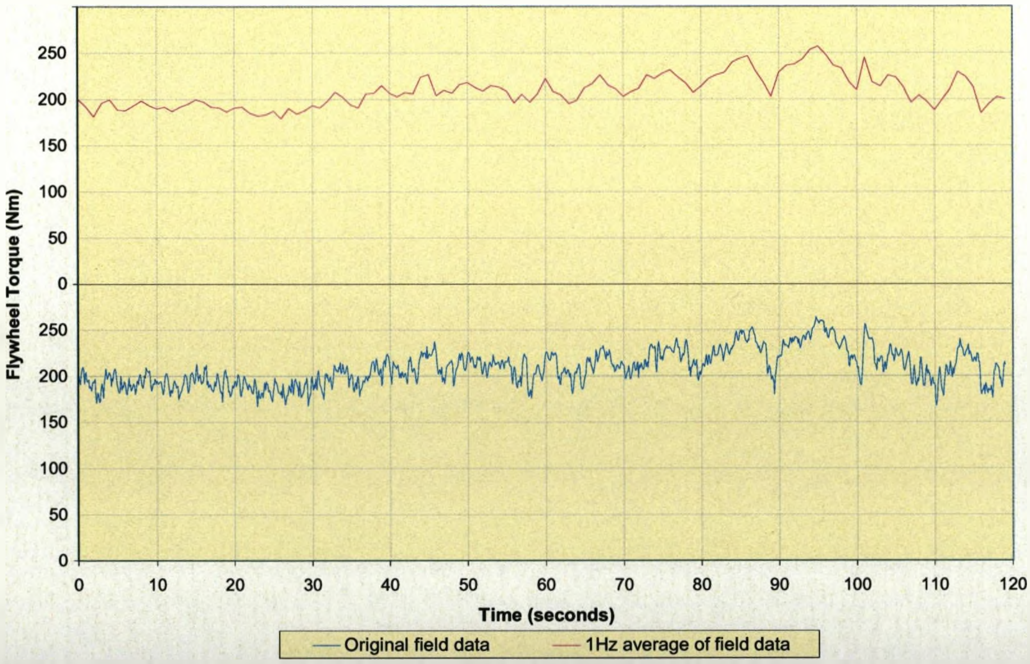



*Figure 6-13 – Field data from ploughing sandy soil in gear 5; original data (bottom) and 1Hz average (top)*

The calculations are presented in equations 6-4 and 6-5.

$$T_{EA} = \frac{T_F \times r_d}{1000} \qquad \textit{Equation 6-4}$$

$$T_L = T_R = \frac{T_{EA}}{2} \qquad \textit{Equation 6-5}$$

Where $r_d$ is the product of the transmission and rear axle ratios, as in Equation 6-6:

$$r_d = r_t \times r_a \qquad \textit{Equation 6-6}$$

The calculated values of torque for the left and right dynamometer units were then used to create a 120 second test sequence file with the requested torque at each rear dynamometer unit brake updating every second.

### 6.4.4 Initial Test – 100% of Field Load Applied

The tractor gear was set to match that used in the field and the axle dynamometer programmed to give the same level of flywheel torque as the first step of the TSF. Engine speed was then set to match that recorded in the field. Once the tractor operation was stabilised the test sequence file was executed.

Flywheel torque was recorded using the same data collection equipment previously used in the field experiments (see Section 5.2.3). Data was sampled at 100Hz and averaged to 10Hz prior to recording. The resulting time history of flywheel torque against the target torque (the averaged field data) is shown in Figure 6-14.

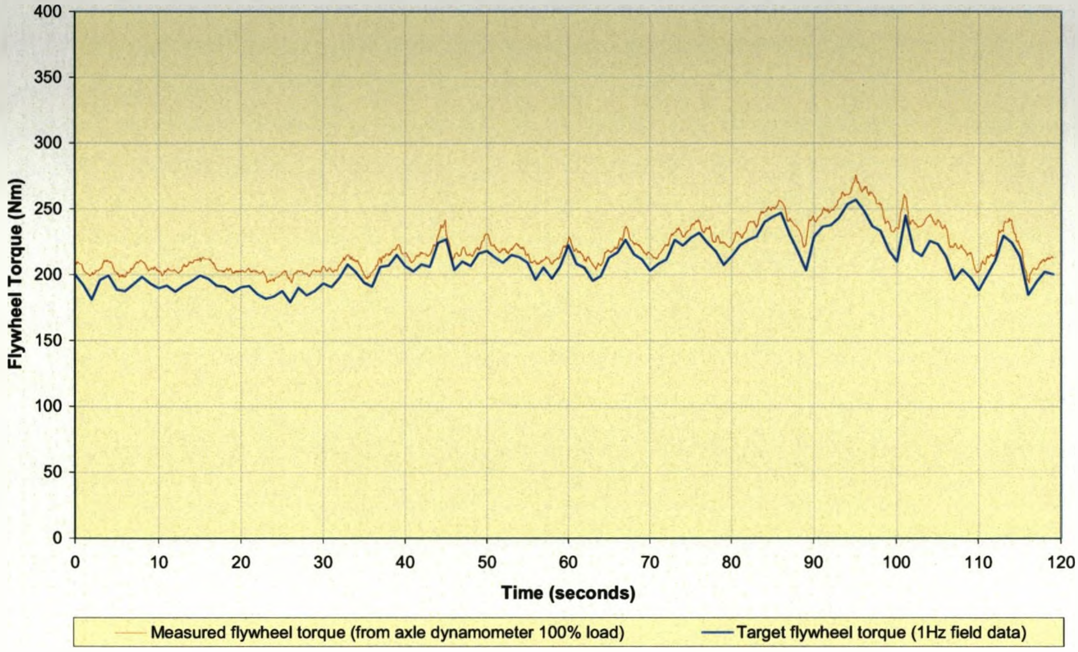



*Figure 6-14 – Initial field data replication (100% of field load applied at the rear axle)*

From Figure 6-14 it is evident that, although the pattern of loading produced by the dynamometer is reasonably representative of the flywheel torque measured in the field, it is of a greater magnitude to the field data in all instances. The primary reason for this is that during the calculation of equivalent wheel torques ($T_L$ and $T_R$), no account was taken for the additional torque required to overcome the losses in the driveline: primarily friction and viscous drag. These loss requirements, when considered at the flywheel, are added to any input torque at the rear axle. Hence there was higher flywheel torque from the axle dynamometer.

### 6.4.5 Improved Test – Reduced Axle Torque

In order to improve the accuracy of the axle dynamometer field data replication, the torque values in the test sequence file were progressively reduced until an optimum response was achieved closest to the field data recorded. The best relationship was found to be when the brake torque values in the test sequence file were set to 90% of the original calculated values. This matches the findings of Culshaw (1988), who found the maximum power available at the axles was 9% less than that from the P.T.O. when analysing dynamometer results from the CEMAGREF test station. The results of the 90% TSF file are shown in Figure 6-15.

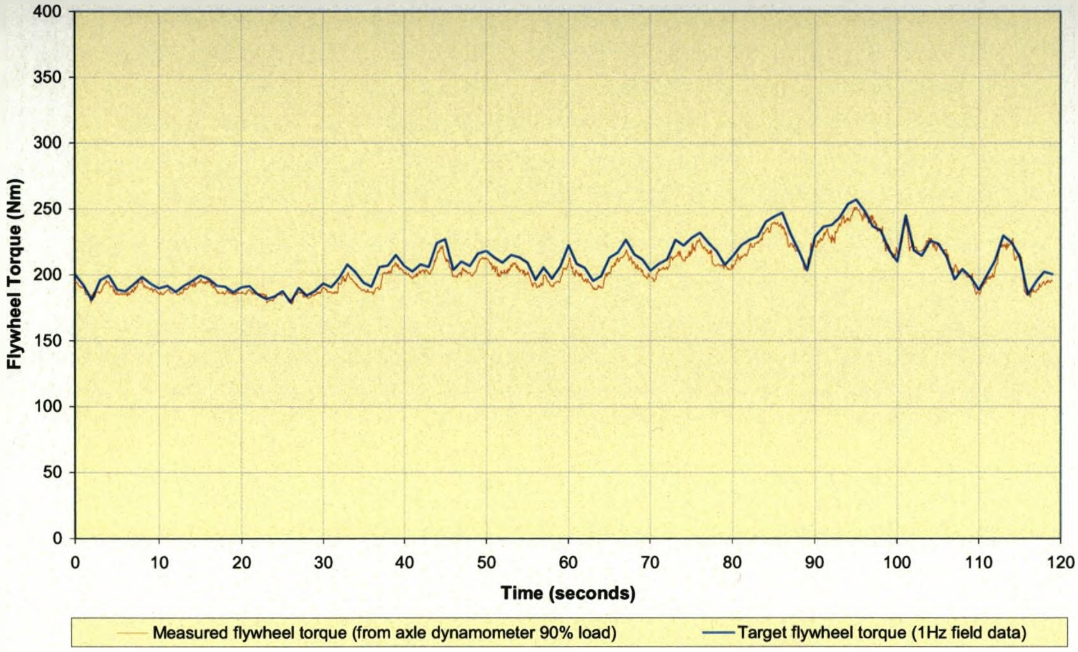



*Figure 6-15 – Improved field data replication (90% of field load applied at the rear axle)*

As can be seen, the response is much closer to the original field data; the R.M.S. torque error was reduced from 12.6Nm in the original test to 6.5Nm in this example. It is reasonable to assume, that if the TSF was further refined, it would be possible to further reduce discrepancies. It is evident from these results that it is possible to replicate field loads on the axle dynamometer to an acceptable level of accuracy allowing the facility to be used in the place of field experiments, thus allowing tests to be replicated in a controlled manner.

However, the differences between axle and flywheel torque seen during this test highlighted the need to make proper consideration of driveline torque losses. This was compounded by further field data replication practice tests in different gears and torque loadings where the difference between axle and flywheel torque varied. It is also envisaged that engine speed would have a significant effect on the torque losses experienced. In addition to improving the accuracy of field data replication, a thorough understanding of the magnitude of torque loss in the driveline was a valuable addition to the mathematical powertrain model (Section 4). The next part of this investigation therefore attempts to quantify these driveline losses.

## 6.5 Driveline Loss and Efficiency Measurements

### 6.5.1 Background

As was established in Section 2.7, the design, complexity and operating principles of a transmission can result in marked differences in the effective transfer of torque and therefore its operating efficiency. These differences can occur both for different configurations within one driveline (different gears, loads or speeds), and between different driveline designs. For this reason, despite a number of previous investigations into tractor driveline losses (McCarthy and Kolozsi, 1974; Schultz *et al*, 1987; Reiter, 1990; Ryu *et al*, 2003), it was necessary to undertake a practical investigation in this specific case.

Previous work did, however, lead to the development of a suitable methodology and highlighted the need to establish both loaded and no-load (speed) related losses (McCarthy and Kolozsi, 1974; Reiter, 1990). No-load losses are a result of the energy required to rotate the driveline elements, including disengaged multi-plate clutches, bearing losses and losses due to lubrication (including oil churning). Load dependent losses are a result of friction between gear teeth. Reiter (1990) concluded that load-related losses were most significant under 5km/h vehicle forward speed, and speed-related losses were most dominant above 20km/h. Overall efficiencies for the transmission tested were found to be in the range of 80% to 88% for forward speeds below 15km/h. This dropped to 65% for higher forward speeds (30km/h) in low-load situations. However, the exact configurations of the (stepped-ratio) transmission investigated are unknown. Schultz *et al* (1987) established that the most important factors (in order of influence) for the transmission under test were: the load, the engine speed and then the gear ratio. Other factors such as angle of operation, oil temperature and gear properties can also influence efficiency. Although the exact specification of the transmission under test was again unknown, it is reported to be a non-synchromesh stepped-ratio transmission as fitted to an Ursus C-330 tractor. Schuster (2000) presented an average efficiency value for each gear in a truck transmission. This showed the marked differences between different gear ratios across a range of operating loads and vehicle forward speeds. A valid point from this work, which could be significant to this investigation, was that the transmission gear ratios with the highest efficiency tended to be those with the simplest power transmission path through the driveline.

During the field experiments (see Section 5) it was apparent that the test tractor seemed to operate more effectively in some gears compared to others. However, with the interaction of all the other tractor-implement parameters investigated, the effects were difficult to quantify. The replication of field loads with the axle dynamometer (Section 6.4) further highlighted the need to quantify transmission losses.

### 6.5.2 Objectives

The objectives of this part of the investigation were:

1. to define the power and torque losses, and therefore efficiency characteristics of the test tractor driveline in different operational situations; and
2. to develop a generic model of driveline torque loss suitable for use in the vehicle simulation model.

### 6.5.3 Experimental Equipment

The axle dynamometer was used to provide a means of loading the driveline of the test tractor in a repeatable manner. The rear axle dynamometer units were used to apply a known steady state torque at the axle ends ($T_L$ & $T_R$) which was then compared to the torque at the tractor flywheel ($T_F$), shown diagrammatically in Figure 6-16. The investigation was restricted to the rear axle driveline to limit the complexity of this part of the investigation.

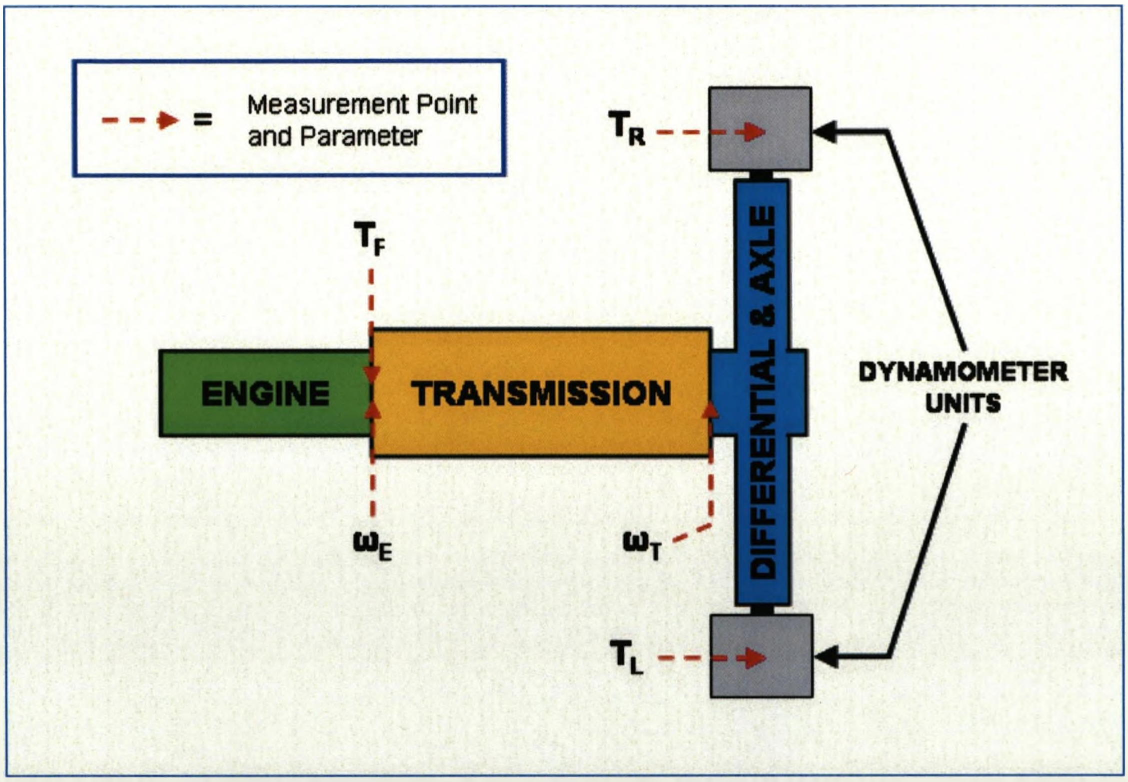



*Figure 6-16 – Measurement locations to determine driveline power and torque losses*

### 6.5.4 Experimental Design and Procedure

Four levels of flywheel torque and three transmission input speeds were investigated for each of the 16 transmission gear ratios. Flywheel torques of 100Nm, 250Nm and 400Nm were considered, together no application of the dynamometer unit brakes. Each torque level was tested in conjunction with three engine (transmission input) speeds ($\omega_E$), of 1400rpm, 1800rpm and 2200rpm. Where possible, all speed and torque combinations were investigated for each gear (G), although some of the higher levels of flywheel torque had to be excluded in the lower gears, as the very high overall driveline ratio meant the required torque applied at the dynamometer units was above the maximum safe loading levels of the test tractor rear axle, as specified by the manufacturer (see Table 6.4).

*Table 6.4 – Theoretical axle torque loadings and excluded tests*

| Gear (G) | Transmission gear ratio ($r_t$) | Driveline gear ratio ($r_d$) | Theoretical wheel torque (kNm) required for flywheel torque ($T_F$) of: 100Nm | 250Nm | 400Nm |
|---|---|---|---|---|---|
| 1 | 8.51 | 300.1 | 15.0 | 37.5 | 60.0 |
| 2 | 6.94 | 244.7 | 12.2 | 30.6 | 48.9 |
| 3 | 5.70 | 200.9 | 10.0 | 25.1 | 40.2 |
| 4 | 4.65 | 163.8 | 8.2 | 20.5 | 32.8 |
| 5 | 3.62 | 127.8 | 6.4 | 16.0 | 25.6 |
| 6 | 2.95 | 104.1 | 5.2 | 13.0 | 20.8 |
| 7 | 2.43 | 85.5 | 4.3 | 10.7 | 17.1 |
| 8 | 1.98 | 69.7 | 3.5 | 8.7 | 13.9 |
| 9 | 2.18 | 76.8 | 3.8 | 9.6 | 15.4 |
| 10 | 1.77 | 62.6 | 3.1 | 7.8 | 12.5 |
| 11 | 1.46 | 51.4 | 2.6 | 6.4 | 10.3 |
| 12 | 1.19 | 41.9 | 2.1 | 5.2 | 8.4 |
| 13 | 0.93 | 32.7 | 1.6 | 4.1 | 6.5 |
| 14 | 0.76 | 26.7 | 1.3 | 3.3 | 5.3 |
| 15 | 0.62 | 21.9 | 1.1 | 2.7 | 4.4 |
| 16 | 0.51 | 17.8 | 0.9 | 2.2 | 3.6 |
| | | = Excluded tests (exceeds safe axle torque) | | | |

The rear dynamometer units were used to apply a braking torque at each axle end, the total axle torque ($T_A$) being their sum:

$$T_A = T_L + T_R \qquad \text{Equation 6-7}$$

The total axle torque can be translated into an "equivalent" flywheel torque ($T_{EF}$) by dividing by the total driveline ratio ($r_d$) and converting into the same units (Nm). The difference between $T_{EF}$ and the measured flywheel torque is accounted for by the driveline torque loss:

$$T_F = \frac{T_A \times 1000}{r_d} + T_{LOSS} = T_{EF} + T_{LOSS} \qquad \text{Equation 6-8}$$

Flywheel power loss ($P_{LOSS}$) is calculated by:

$$P_{LOSS} = T_{LOSS} \times \omega_E \qquad \text{Equation 6-9}$$

Whilst driveline efficiency (η) is then defined by as the ratio between $T_{EF}$ and $T_F$:

$$\eta = \frac{T_{EF}}{T_F} \times 100 \qquad \textit{Equation 6-10}$$

During each test, the applied axle torque was increased until the required flywheel torque was achieved, at which point engine speed was adjusted to the correct level. The parameters were then sampled at 10Hz for 20 seconds, at the end of which the mean and standard deviation of each were recorded. The tests were carried out at random and each replicated three times. Accounting for those excluded, this resulted in 504 individual tests being undertaken. In addition, the no-load tests were repeated with the tractor rear wheels fitted (in place of the dynamometer units) but raised off the ground in order to determine whether any dynamometer brake drag torque present differed from the steady state wheel torque.

### 6.5.5 Parameters

The parameters which were recorded are presented in Table 6.5; the only parameter recorded from the dynamometer was total axle torque, all other parameters were recorded from the tractor via the CAN-bus using the same data acquisition equipment employed during the field trails. During data processing, flywheel power loss, torque loss ($T_{LOSS}$) and transmission efficiency (η) were calculated.

*Table 6.5 – Transmission efficiency recorded parameter details*

| No | Parameter | Notation | Data Source | | Output |
|---|---|---|---|---|---|
| | | | Transducer Type | Transducer Location | |
| T1 | Transmission Input Speed (Engine Speed) | $\omega_E$ | variable reluctance | crankshaft tone wheel | rpm |
| T2 | Flywheel Torque | $T_F$ | Hall effect | flywheel damper | Nm |
| T4 | Gear Number | G | calculated by (1) and (5) | | integer |
| T5 | Transmission Output Speed | $\omega_T$ | Hall effect | transmission output shaft | rpm |
| T6 | Theoretical Forward Speed | $v_t$ | Calculated from (5) | | km/h |
| D1 | Total Axle Torque | $T_A$ | 2 x Load Cells | Dynamometer Units | kNm |

### 6.5.6 Results

The mean torque loss data is presented in the form of histograms, with the error bars showing the standard error of the mean (S.E.M.) of the three replicates. Additional numerical data for each configuration is presented in Appendix A5.2. Figure 6-17 presents the no-load torque loss for each gear at the three different transmission input speeds. Theoretical vehicle forward speed is also shown on this chart. The same no-load tests are presented for power loss in Figure 6-18. The power losses at the three loaded flywheel torque levels investigated are presented in Figures 6-19, 6-20 and 6-21. For additional torque loss charts for these datasets see Appendix A5.3. Driveline efficiency charts for the loaded tests are shown in Figures Figure 6-22, 6-23 and 6-24. Due to the high driveline loads experienced in the 400Nm test, the actual transmission input speeds, when different from the stated value, are also shown on Figure 6-21. The comparison between no-load tests on the axle dynamometer and tests with the rear wheels fitted to the tractor is shown in Figure 6-25, with statistical data presented in Appendix A5.4.

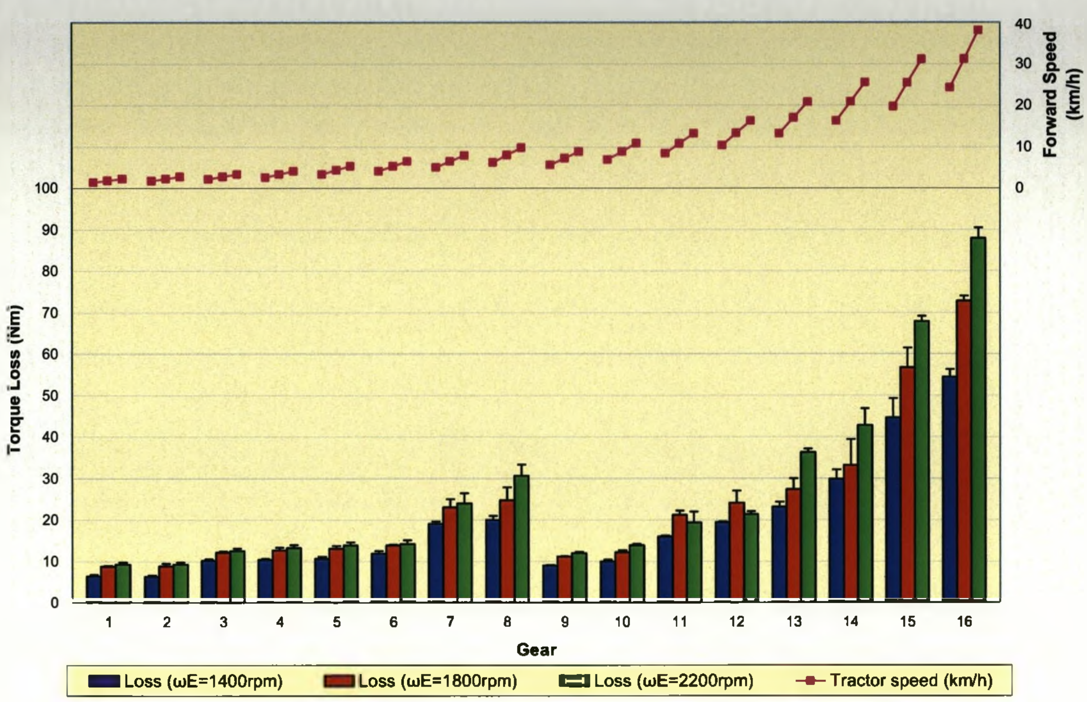



*Figure 6-17 – Mean no-load torque losses (with theoretical vehicle forward speeds) at varying transmission input speeds in each gear*

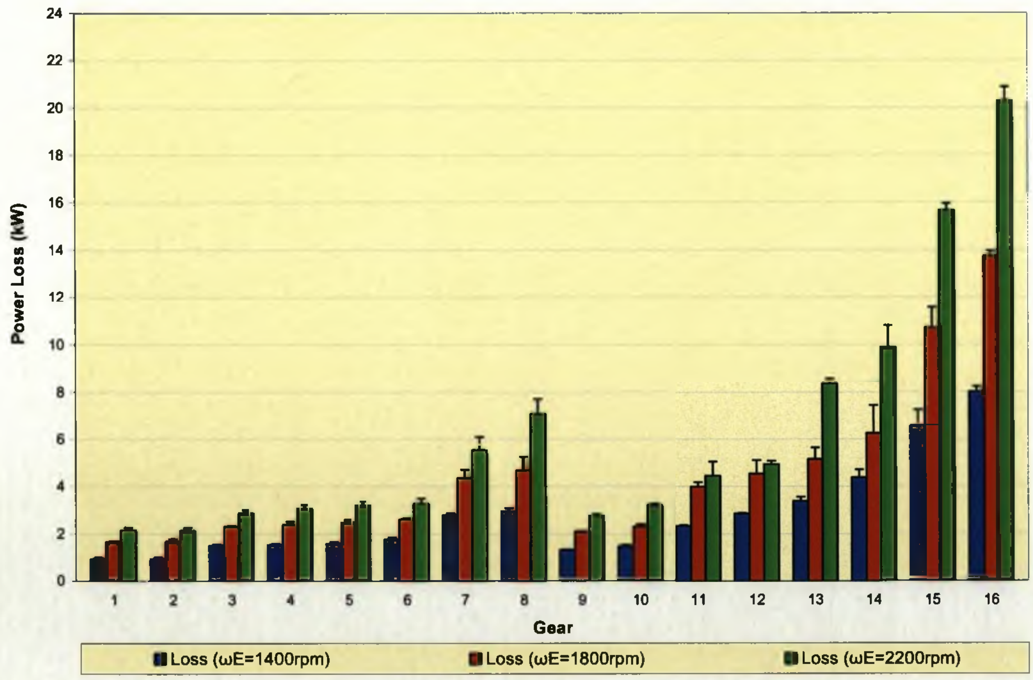



*Figure 6-18 – Mean no-load power losses at varying transmission input speeds in each gear*

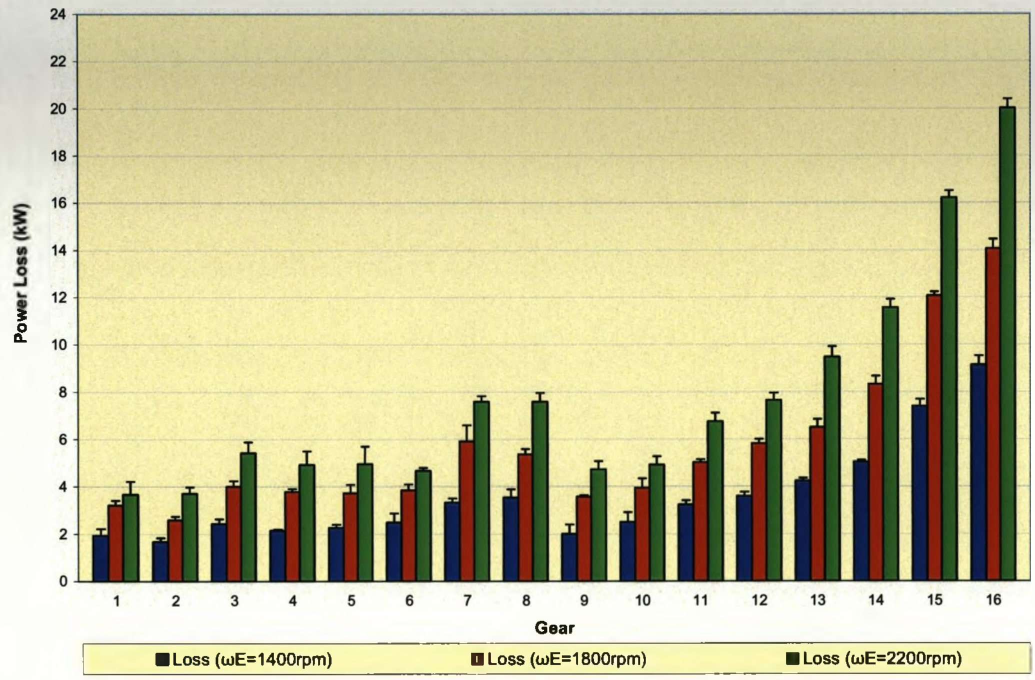



*Figure 6-19 – Mean power loss at varying transmission input speeds in each gear when flywheel torque load = 100Nm*

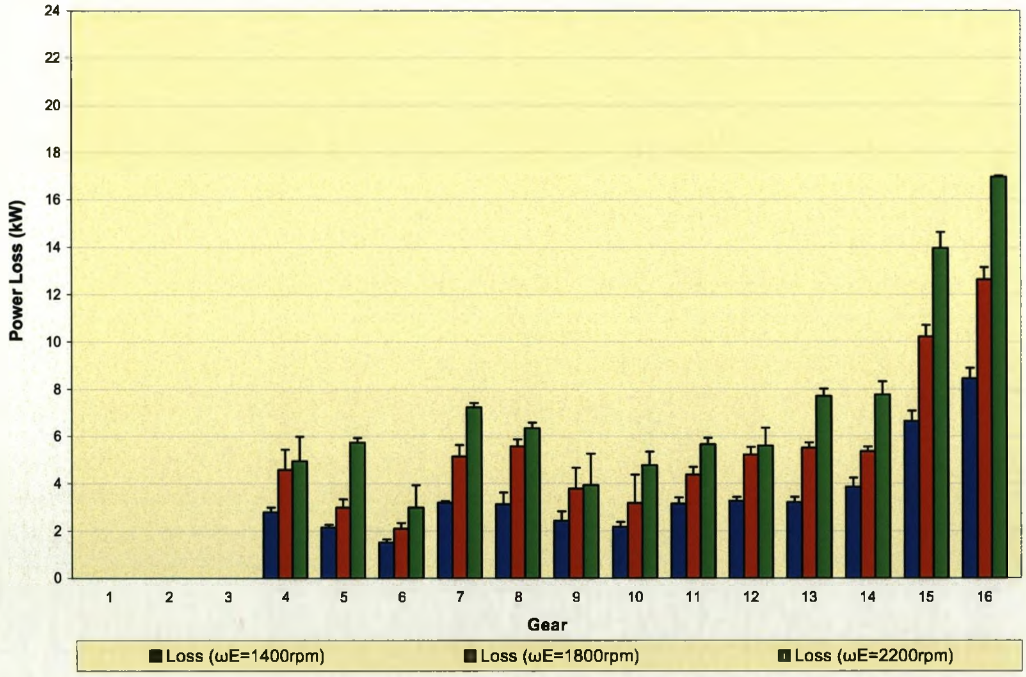



*Figure 6-20 – Mean power loss at varying transmission input speeds in each gear when flywheel torque load = 250Nm*

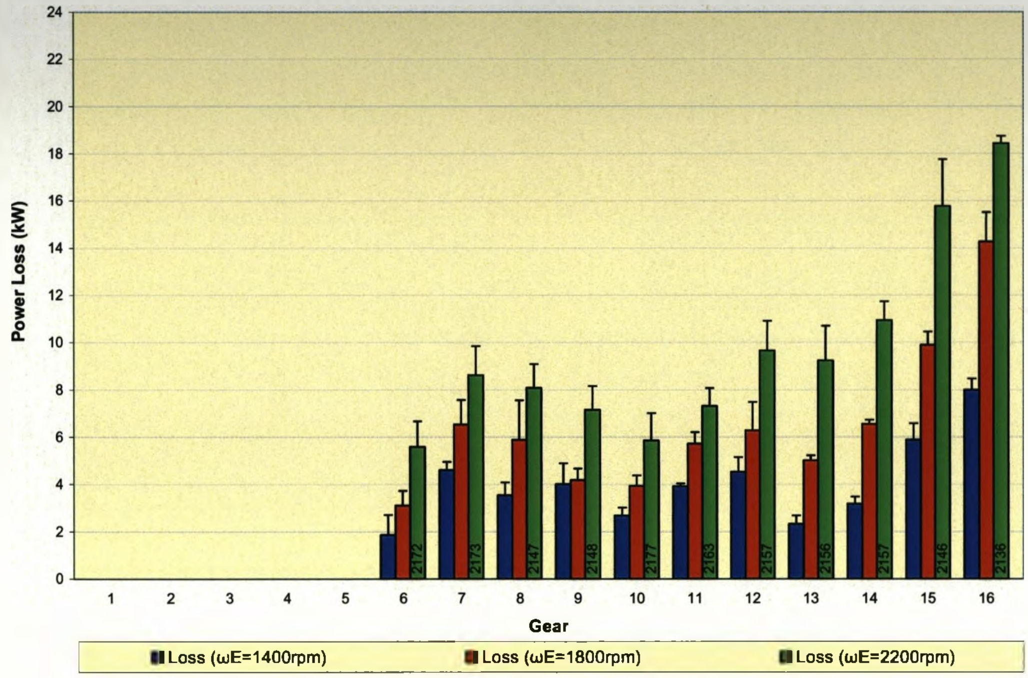



*Figure 6-21 – Mean power loss at varying transmission input speeds in each gear when flywheel torque load = 400Nm*

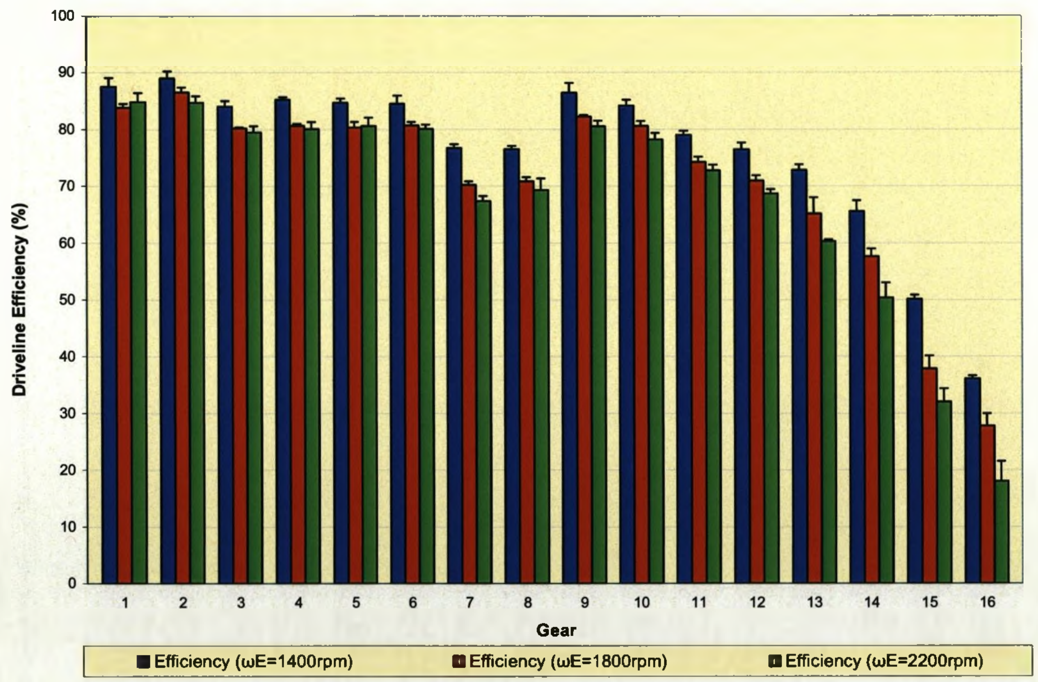



*Figure 6-22 – Mean driveline efficiency at varying transmission input speeds in each gear when flywheel torque load = 100Nm*

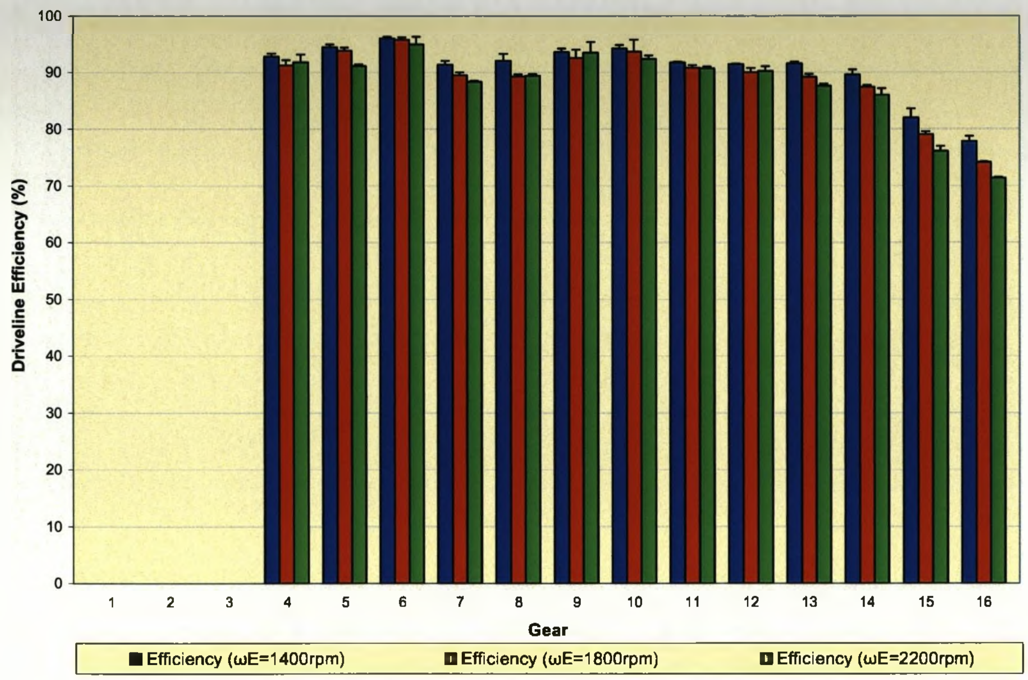



*Figure 6-23 – Mean driveline efficiency at varying transmission input speeds in each gear when flywheel torque load = 250Nm*

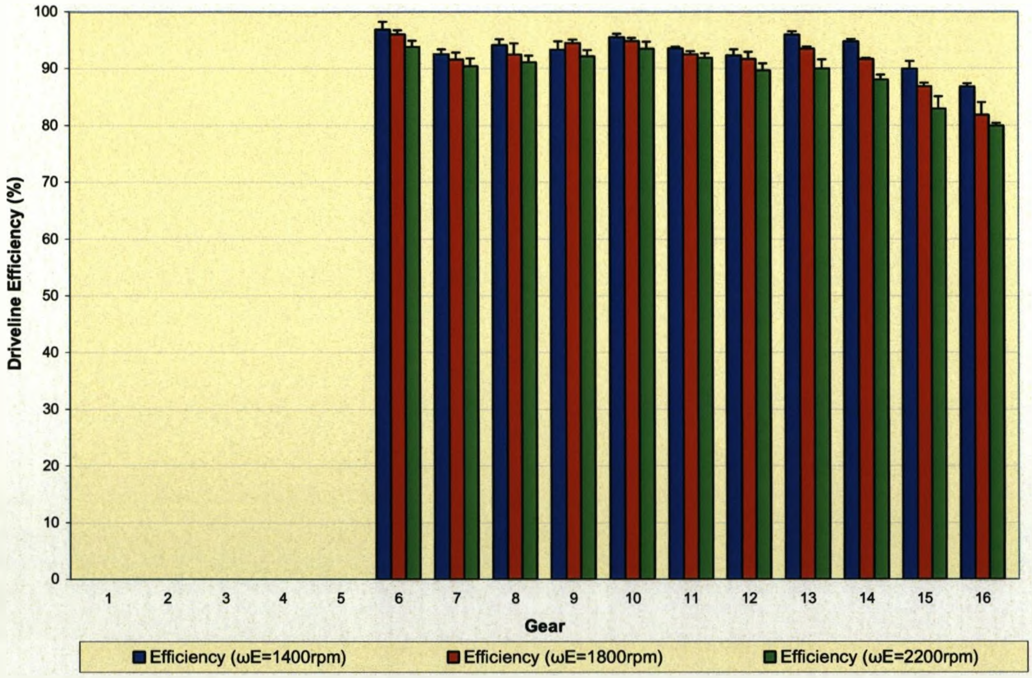



*Figure 6-24 – Mean driveline efficiency at varying transmission input speeds in each gear when flywheel torque load = 400Nm*

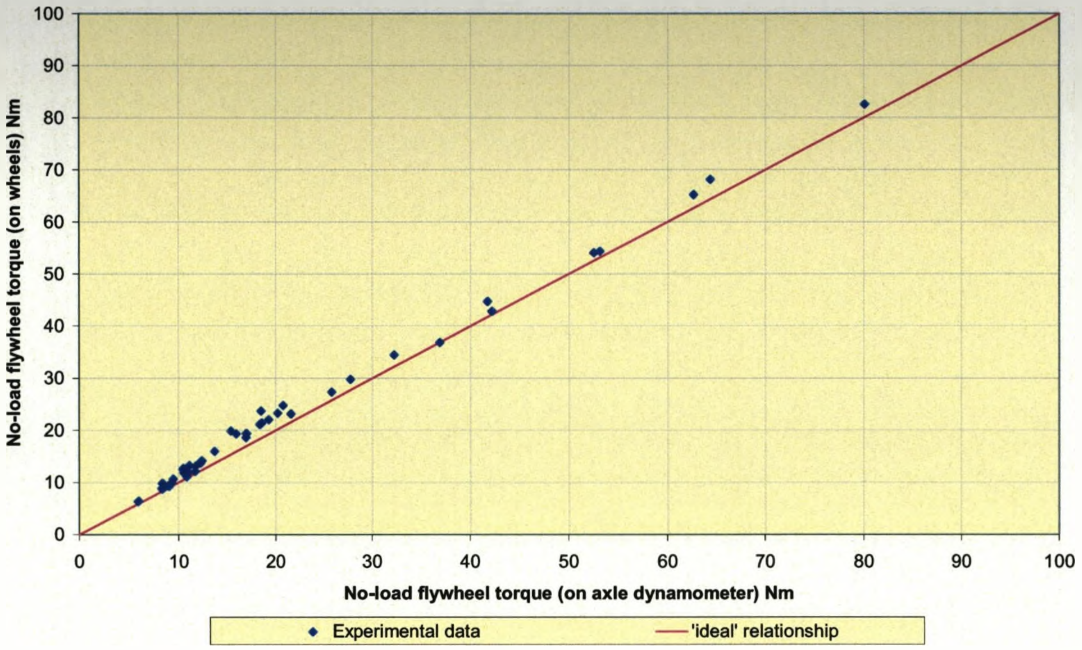



*Figure 6-25 – No-load torque losses on axle dynamometer and on wheels*

### 6.5.7 Discussion

This part of the investigation has produced a comprehensive dataset, not only on the magnitude of transmission losses for different operating scenarios, but also the nature in which the losses are determined. This information has not been available previously due to the need for specialist test equipment such as the axle dynamometer - which has been proven to be ideal for this task. With this information available, it is possible to develop control strategies to optimise vehicle performance and efficiency.

The torque and power loss graphs for the no-load tests (see Figures 6-17 and 6-18) show that there are significant losses in this transmission under no-load, i.e. the majority of the losses from this transmission are speed rather than load related, reflecting the transmission design. The test tractor transmission, described in Section 3.4, has been designed to run at high speeds: the top four gears are all overdrive ratios. Whilst a high-speed / low-torque design allows smaller components to be used, it does lead to increased losses at higher transmission speed. It is also significant that all transmission components, regardless of the gear ratio engagement, rotate at all times, also adding to the no-load losses.

All the tests, regardless of the flywheel torque setting or transmission input speed showed there to be marked differences in the losses experienced in each gear. The general trend is an increase in losses as gear number increases. The graphs (particularly Figure 6-19) show the losses for adjacent pairs of gears are very similar. This is due to transmission design, previously described, whereby changing between adjacent gears (1-2 or 3-4) involves only one deviation on the path through the transmission (see Figure 3-16). In the higher gears this pairing is less pronounced, due to effects of higher rotational speed. The overall trend in rising losses with higher gear does have some discontinuities, particularly when changing ranges from the lower eight to upper eight gears. Also gear 6 has minimal losses at most speed and torque settings. This gear has the most direct path through the transmission, so it was expected to be efficient (see Figure 3-17).

Figure 6-17 shows the no-load torque loss increased rapidly for the higher gears as $\omega_T$ and therefore $v_t$ increased. The correlation coefficient between torque loss and these two factors was 0.95, indicating their likely high influence on torque loss. The no-load power loss graph (see Figure 6-18) follows a similar pattern. Within each gear a higher $\omega_E$ led to increased losses. This was more definite for the higher gears, especially gears 13 to 16 where the speed increase of transmission components, for the same rise in input shaft speed, was greater. In gear 8, increasing $\omega_E$ from 1800rpm to 2200rpm changes $\omega_T$ from 910rpm to 1112rpm, whereas in gear 16 the same input speed rise results in $\omega_T$ increasing from 3557rpm to 4348rpm. These very fast transmission output speeds resulted in high losses. Over 20kW of engine power was consumed purely rotating the drivetrain in gear 16 when $\omega_E$ was 2200rpm.

Figure 6-19 shows the effects on power losses of applying a constant load, equivalent to 100Nm at the flywheel, from the axle dynamometer. In most cases for the first 12 transmission gears this resulted in a statistically significant increase in the power loss when compared with the no-load losses. This is due to the higher torque which needs to be transmitted in the lower gears to achieve 100Nm flywheel torque, thereby increasing the load-dependant losses. In the highest four gears the power loss increases were not statistically significant. The four highest gears were the overdrive gear ratios, where no-load losses were so high that only a small dynamometer torque application was required to obtain 100Nm at the flywheel, hence no significant difference in losses.

Increasing flywheel torque to 250Nm (see Figure 6-20) broadly had no significant impact on the magnitude of driveline losses compared to either no-load or 100Nm tests. In some examples, in the highest four gears, the power loss actually reduced slightly compared to the no-load situation. Whilst there is no obvious explanation for this trend, it could potentially be attributed to optimum transmission geometry or bearing loading under this particular torque setting.

At the highest flywheel torque loads (see Figure 6-21) there was again an increase in power losses in the lower gears when compared with the no-load losses. As with

250Nm, there were some situations in the highest four gears where the losses were lower than the no-load situation, but none were statistically significant. In addition, the power losses at the highest $\omega_E$ setting would have been higher had it been possible to achieve 2200rpm.

The driveline efficiency for the 100Nm (see Figure 6-22), 250Nm (see Figure 6-23) and 400Nm (see Figure 6-24) flywheel torque settings show that as more torque is transmitted the driveline efficiency increases. In Figure 6-22 driveline efficiency reduces below 20% in gear 16, mainly as a result of the high no-load losses meaning little torque is actually transferred through the driveline. Figure 6-24 shows the highest efficiency was obtained in gear 6: at 1400rpm and 400Nm peak efficiency was 96.7%. The efficiency charts also show that, in the working range, a gear-up, throttle down approach, i.e. using a higher gear and reducing engine speed to provide the same engine power (Zoerb *et al*, 1983), not only improves engine operational efficiency, but also the lower speed operation generally would improve transmission efficiency.

It was determined statistically that there was no significant difference between the axle dynamometer and the tractor wheels in terms of no-load torque for all but two of the 48 configurations considered (see Figure 6-25 and Appendix A5.4). Therefore the axle dynamometer units were deemed to be a suitable substitute for the losses which would be experienced by the wheels during the tests.

The data generated during this experiment was used to determine a transmission loss model, to form part of the driveline sub-model (see Section 4.4.3).

### 6.5.8 Driveline Torque Loss Model

To improve the transmission sub-model (see Section 4) the driveline loss data was used to develop a regression model to predict the flywheel torque from a known input torque (i.e. determine the additional torque required to be produced to overcome the driveline losses). The objective was to produce the most accurate possible generic model, i.e. one avoiding transmission-specific factors. This would allow the model to be applied to similar transmission configurations with only the coefficients needing to be adjusted. The trends identified in Section 6.5 were taken into consideration when choosing model parameters.

The factors considered for inclusion were:

- equivalent flywheel torque ($T_{EF}$);
- transmission input speed ($\omega_E$);
- transmission output speed ($\omega_T$);
- driveline gear ratio ($r_d$);
- gear number (G);
- number of torque transferring gear meshes ($G_m$).

During initial investigations a number of additional parameters, including $\omega_E^2$, $\omega_T^2$, $G^2$ and $G_m^2$, were considered, but were found to be insignificant or were found to add no significant improvement, whilst they reduced the degrees of freedom. A factor for overdrive gear ratios, to account for the differences experienced from the experimental results, was tested but not found to be statistically significant. $G_m$ was added to account for the differences in transmission path in each gear, expressed as the number of torque transferring gear meshes (see Table 6.6):

*Table 6.6 - Number of torque transferring meshed transmission gear pairs*

| G | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| $G_m$ | 5 | 3 | 7 | 5 | 3 | 1 | 5 | 3 | 3 | 3 | 5 | 5 | 1 | 1 | 3 | 3 |

$T_{EF}$ was included in all models, as the torque input is always transferred through the transmission. The other factors were then determined on a trial and error basis. Table 6.7 shows ten of the models in the final development process leading the optimal model. Also included are four models which would be considered obvious developments of others and the factors which were not statistically significant, thereby prohibiting their use. The complete regression parameters and statistics are presented in Appendix A5.5.

*Table 6.7 – Driveline torque loss models tried (chosen model highlighted)*

| Model | Fitted parameters | | | | | | Not significant (5%) level | Residual sum of squares | % variance accounted | Std error of observations |
|---|---|---|---|---|---|---|---|---|---|---|
| | $T_{EF}$ | $\omega_E$ | $\omega_T$ | $r_d$ | G | $G_m$ | | | | |
| 1 | X | X | | X | | | - | 101,684 | 99.0 | 14.20 |
| 2 | X | X | | X | X | | $r_d$ | - | - | - |
| 3 | X | X | X | | | | - | 27,380 | 99.7 | 7.39 |
| 4 | X | X | X | | | X | ωE | - | - | - |
| 5 | X | | X | | | | - | 32,587 | 99.7 | 8.06 |
| 6 | X | | X | X | | | - | 25,471 | 99.7 | 7.13 |
| 7 | X | | X | | X | | G | - | - | - |
| **8** | **X** | | **X** | | | **X** | **-** | **22,832** | **99.8** | **6.75** |
| 9 | X | | X | X | | X | $r_d$ | - | - | - |
| 10 | X | | X | | X | X | - | 20,593 | 99.8 | 6.42 |
| X = Parameter included in regression model | | | | | | | | | | |

Although model 10 was the most accurate, gear number is not a physical entity, so this model was rejected, the slight accuracy improvement not justifying a move away from a fundamental relationship. The chosen regression model (number 8) is shown in Equation 6-11:

$$T_F = (1.00860 \times T_{EF}) + (0.016359 \times \omega_T) + (1.66 \times G_m) \qquad \textit{Equation 6-11}$$

A sensitivity analysis was undertaken to determine the required resolution of the coefficient values for $T_{EF}$, $\omega_T$ and $G_m$: the final model is shown in Equation 6-12:

$$T_F = (1.01 \times T_{EF}) + (0.016 \times \omega_T) + (1.7 \times G_m) \qquad \textit{Equation 6-12}$$

This increased the residual sum of squares to 22930, thereby slightly increasing the standard error to 6.77. The percentage variance accounted for remained at 99.8% due to the small increase in residual sum of squares compared to the total sum of squares (25,784,117).

The final model (Equation 6-12) was used as part of the overall vehicle model. If it was desirable to determine the torque loss ($T_{LOSS}$) through the transmission, this could be calculated from (Equation 6-8).

## 6.6 Driveline Inertia

### 6.6.1 Background

Inertia, as established by Newton's 1st law (Meriam and Kraige, 1998), is the tendency of a body to maintain its state of uniform motion unless acted on by an external force. Powertrain dynamic behaviour under conditions of changing speed is dependant on the effects of both linear and rotational inertia (Schmid *et al*, 1988). The system requires a force (or torque) to overcome inertia and accelerate it, as defined by Newton's 2nd law (and its rotational derivation).

Linear inertia is determined easily from the vehicle mass. Rotational inertia is more difficult to quantify as the moment of inertia of a rotating body is a result not only of its mass, but also the distribution of mass from the centre of rotation. For dynamic analysis it is useful to sum the inertias of a number of components together at one point to aid calculations. The inertia of any one component is constant regardless of its rotational velocity, but when transferring to a reference point its 'equivalent' inertia will vary according to the speed ratio between the component and the reference point (see Equation 6-13). This also means that even in this driveline where all components rotate at all times, the driveline inertia will be different in each transmission gear.

$$equivalent\ \ inertia = component\ \ inertia \times \left( \frac{component\ \ speed}{reference\ \ speed} \right)^2 \qquad \textit{Equation 6-13}$$

In the test tractor, the total driveline rotational inertia includes the individual inertias of all shafts, gears, synchronisers, bearings, clutches, brakes, wheels and tyres. The result of Equation 6-13 is that the high ratio reduction in the lowest gears reduces the equivalent inertia of the major components such as the wheels, when considered at the flywheel. However, given the size and weight of the wheels, the effects in the higher gears could still be significant. It was therefore necessary to determine the driveline inertia, at least for the highest transmission gear ratios.

Previous work by Drouin *et al* (1991) determined the moment of inertia for a tractor driveline theoretically, as a high accuracy was required for the calculation of torsional vibrations. Each component was analysed individually with calculations made from dimensions and material properties. For determining driveline performance, a less rigorous approach is usually sufficient. Vaughan and Banisoleiman (1986) made inertia estimates for a twin-layshaft ten speed truck transmission based on the approximate physical dimensions of key components, quoting an overall inertia for each gear. A practical approach using a moment of inertia rig was used by Phillips (1991) to determine the inertia of some major car transmission components. With limited time and resources available a novel approach to determine inertia was used.

### 6.6.2 Objective

The objective of this part of the investigation was to determine the magnitude of the test tractor driveline rotational inertia in the upper range of gears (9-16), referenced to the engine flywheel.

### 6.6.3 Experimental Approach

#### *6.6.3.1 Methodology*

The tractor rear axle was supported by axle stands and the wheels were fitted, allowing them to rotate in the air. By measuring the flywheel torque required to accelerate the driveline at a known (measured) rate, it was theoretically possible to determine the driveline and wheel inertia at the flywheel derived from the rearranged, rotational form of Newton's 2nd law and expressed by Equation 6-14:

$$I_{DF} = \frac{T_{FA}}{\dot{\omega}_T \times r_t}$$ *Equation 6-14*

This same principle was successfully used by Harari and Sher (1995) to determine engine friction, by cutting fuel to an engine connected to a known high inertia mass and measuring the deceleration rate.

In this experiment the engine speed was raised to maximum and then the clutches engaged and the driveline accelerated. Normal clutch engagement was bypassed to allow the C3/C4 clutch to engage prior to C1/C2 clutch, thus allowing the whole driveline to be accelerated from rest together. Clutch control was adjusted in the transmission software to completely fill the C3/C4 clutch at the start of the clutch pedal travel and the C1/C2 clutch at the end of pedal travel. Whilst this would not be recommended for normal vehicle operation, the limited cycles and the wheels being in mid-air reduced the possibility of driveline damage.

Tests were undertaken in gears 9 to 13. Transmission software prevented setting driveline engagement from rest in a higher gear than 13. Transmission output speed, gear and flywheel torque were all recorded at their maximum rate of CAN-bus transmission (100Hz) and each test was replicated at least three times.

From the recorded data, the acceleration torque at the flywheel ($T_{FA}$) was calculated, by subtracting the torque to overcome driveline losses ($T_{LOSS}$) (calculated from the driveline loss model) from the measured flywheel torque ($T_F$) at each time step (see Equation 6-8). The transmission output acceleration ($\dot{\omega}_T$) was calculated and transformed to the input shaft and Equation 6-14 was used to determine the driveline inertia at each time step during acceleration. Plots of transmission acceleration against inertia were then used to determine mean driveline inertia, for that gear only, during continuous acceleration.

#### *6.6.3.2 Results*

The calculated driveline inertia for each experimental run is shown in Table 6.8, along with the standard error of the mean (S.E.M.) and the overall mean of the three replications. The transmission acceleration and inertia at each time step are shown for gear 9 (see Figure 6-26) and gear 13 (see Figure 6-27), the vertical lines representing the portion of the test used to calculate driveline inertia.

*Table 6.8 – Experimental results for driveline inertia determination*

| Gear | Rep | $r_t$ | $I_{DF}$ | S.E.M. ($I_{DF}$) | File Ref | Mean $I_{DF}$ |
|---|---|---|---|---|---|---|
| 9 | 1 | 2.18 | 0.11 | 0.08 | INERT203 | 0.12 |
| 9 | 2 | 2.18 | 0.14 | 0.07 | INERT206 | |
| 9 | 3 | 2.18 | 0.12 | 0.07 | INERT207 | |
| 10 | 1 | 1.77 | 0.19 | 0.07 | INERT202 | 0.19 |
| 10 | 2 | 1.77 | 0.19 | 0.10 | INERT204 | |
| 10 | 3 | 1.77 | 0.18 | 0.12 | INERT208 | |
| 11 | 1 | 1.46 | 0.24 | 0.16 | INERT301 | 0.25 |
| 11 | 2 | 1.46 | 0.27 | 0.19 | INERT302 | |
| 11 | 3 | 1.46 | 0.25 | 0.15 | INERT304 | |
| 12 | 1 | 1.19 | 0.38 | 0.17 | INERT303 | 0.36 |
| 12 | 2 | 1.19 | 0.37 | 0.18 | INERT305 | |
| 12 | 3 | 1.19 | 0.34 | 0.19 | INERT306 | |
| 13 | 1 | 0.93 | 0.53 | 0.19 | INERT205 | 0.55 |
| 13 | 2 | 0.93 | 0.54 | 0.25 | INERT209 | |
| 13 | 3 | 0.93 | 0.58 | 0.25 | INERT201 | |

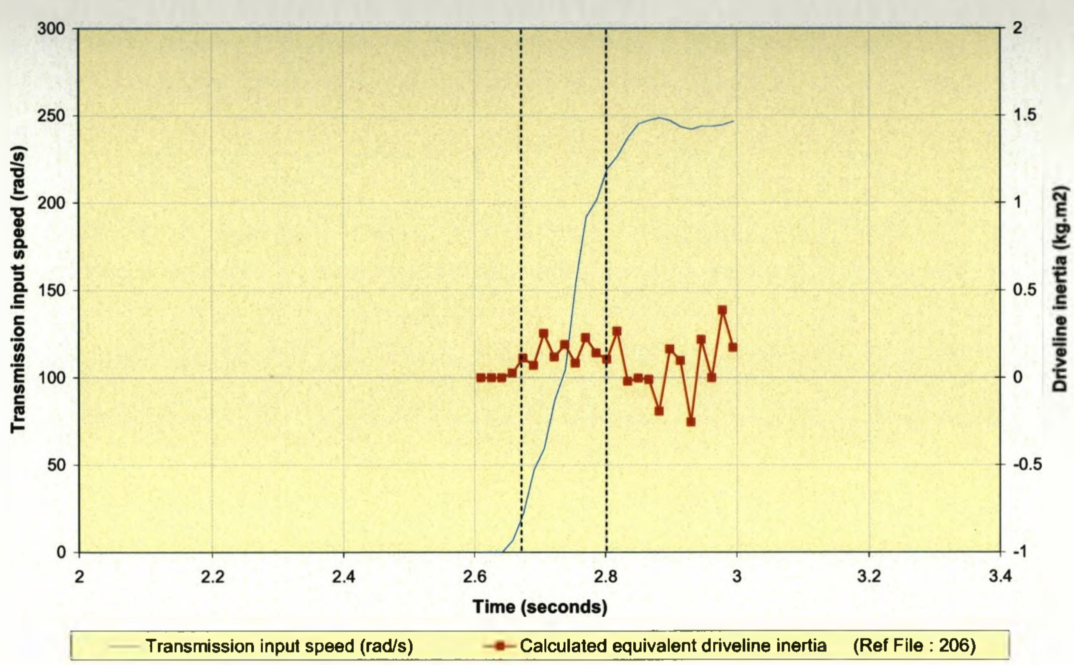



*Figure 6-26 – Transmission input speed and driveline inertia during acceleration in gear 9*

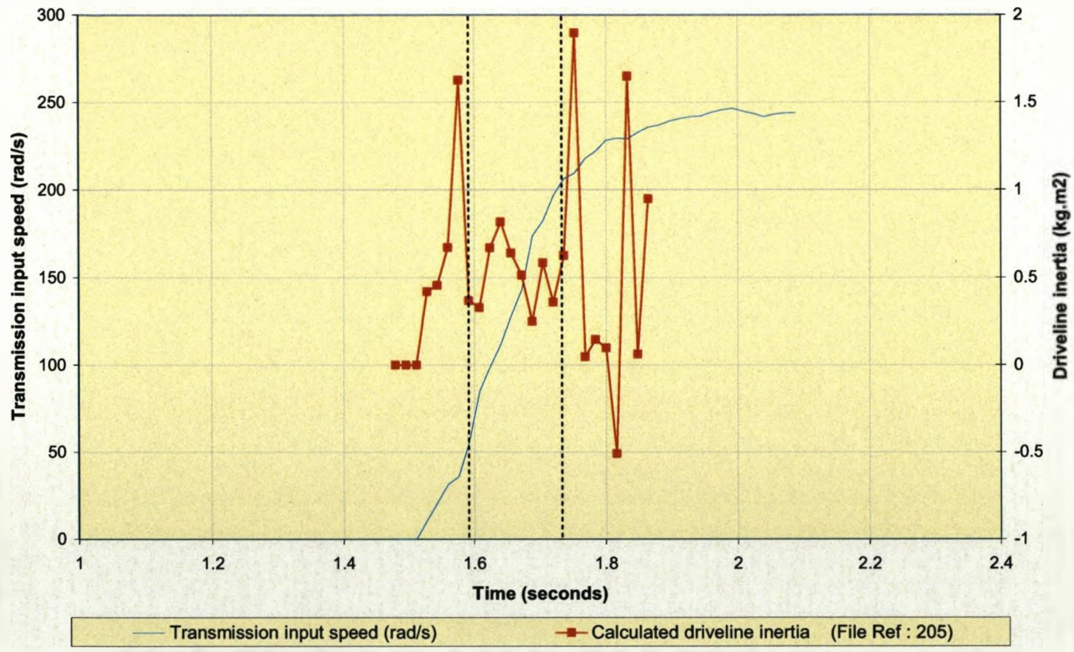



*Figure 6-27 – Transmission input speed and driveline inertia during acceleration in gear 13*

### 6.6.3.3 Discussion

This experimental method seemed to provide a relatively robust and repeatable means of determining driveline inertia. The standard error of the mean in each test increased with increasing transmission gear ratio: the higher torque requirements to overcome losses and accelerate the driveline in these gears resulted in a more significant engine speed reduction. In gear 9, the minimum engine speed was 2249rpm, whereas in gear 13 it was 2017rpm from a no-load speed of 2370rpm. Figure 6-27 shows the effects of this changing engine speed. The acceleration profile is more varied than in Figure 6-26, resulting in more inertia variation.

The upward trend in driveline inertia with increasing gear ratio matches theoretical expectations, i.e. the higher transmission speed ratios resulted in increased driveline inertia. The most significant inertia is likely to arise from the rear wheels and tyres, due to their high mass and large diameter. Their location in the driveline would result in a larger reduction in their equivalent inertia in the lower gears than in the upper gears.

### 6.6.4 Pendulum Method for Wheel Inertia Determination

#### *6.6.4.1 Methodology*

Further investigation of wheel inertia was undertaken to determine the influence on overall driveline inertia, and then to allow an approximation for the driveline inertia in gears 14 to 16. A variation on the pendulum method of determining wheel inertia described by Metz *et al* (1990) was used. A pendulum has the property that its period is constant even as its speed of oscillation changes. Setting up the wheel and tyre as a pendulum from a fixed pivot point (see Figure 6-28), and timing the period of oscillation ($\tau$) as a result of a small angular deflection, allowed the wheel inertia about the pivot ($I_{WO}$) to be determined (Equation 6-15). This was subsequently used to determine the inertia about its centre of rotation ($I_W$) using Equation 6-16. The time taken for 10 oscillations was recorded (to minimise the effects of timing errors) and repeated three times.

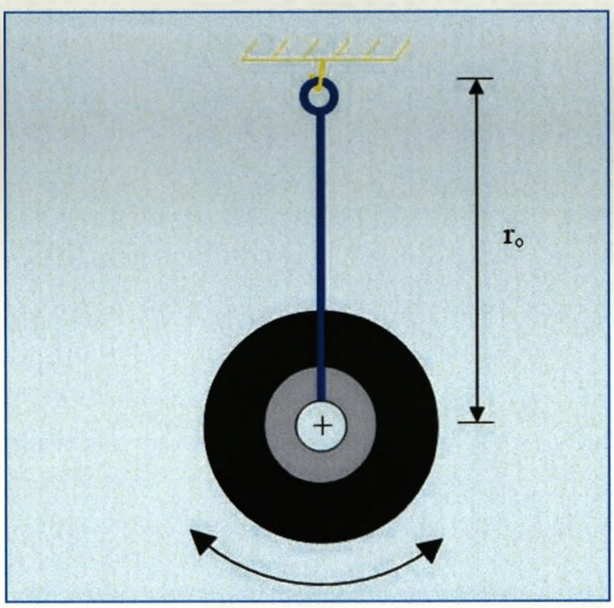



*Figure 6-28 – The pendulum method for determining wheel inertia*

$$\tau = 2\pi\sqrt{\frac{I_{WO}}{M_W \times g \times r_o}}$$ *Equation 6-15*

$$I_W = I_{WO} - \left(M_W \times r_o^{\,2}\right)$$ *Equation 6-16*

### 6.6.4.2 *Results & Discussion*

Table 6.9 shows the results from the wheel inertia experiment. Unfortunately, due to the large mass and pendulum length involved, this method was not successful, as the overall result was negative. Analysis on the right of the table shows that with the mass of the wheel and the pendulum length necessary, a small change in period of 0.1 seconds is sufficient to change the result by over 300kg.m$^2$. It is likely that friction between the hook and the pendulum arm (chain) would have been sufficient to slow down the period to produce this result.

*Table 6.9 – Wheel inertia determination using the pendulum method*

| For 10 Oscillations: | Time (sec) | Mean of 3 reps | |
|---|---|---|---|
| Replication 1 | 43.01 | 43.06 | |
| Replication 2 | 42.95 | | |
| Replication 3 | 43.22 | | |
| | Actual Data | Effects of timing | |
| Period per cycle ($\tau$) (sec) | 4.306 | 4.300 | 4.400 |
| Wheel Mass ($M_W$) (kg) | 363 | 363 | 363 |
| Pendulum Length ($r_o$) (m) | 4.74 | 4.74 | 4.74 |
| Inertia about Pivot ($I_{WO}$) (kg.m$^2$) | 7928 | 7905.53 | 8277.51 |
| Conversion to about Centre | 8156 | 8155.74 | 8155.74 |
| Wheel Inertia ($I_W$) (kg.m$^2$) | **-228.1** | -250.21 | 121.769 |

## 6.6.5 Theoretical Wheel & Tyre Inertia Determination

### 6.6.5.1 *Methodology*

A theoretical approach to determining wheel and tyre inertia was taken, whereby the wheel dish and rim were modelled as solids from engineering drawings, using AutoCAD mechanical desktop. The tyre was modelled using physically-measured dimensions. Difficulties in determining dimensions and the need to estimate the effects of tread, by adding a small additional layer to the overall tyre diameter at ¼ of the tread height, reduced the potential accuracy of the tyre model. Mild steel densities were used for the wheel rim and dish. The tyre density was unknown, so the material density was adjusted in the model until the correct tyre mass was achieved.

### *6.6.5.2 Results & Discussion*

Table 6.10 shows the mass properties for each component part and the overall rotational inertia for one tractor wheel. Appendix A5.5 contains the CAD model and full mass property data.

*Table 6.10 – Mass properties summary table for one rear wheel & tyre*

| Component Part | Material Type | Material Density ($kg/m^3$) | Mass (kg) | Radius of Gyration (m) | Moment of Inertia about spin axis ($kg.m^2$) |
|---|---|---|---|---|---|
| Wheel dish | Mild steel | 7850 | 40.8 | 0.265 | 2.87 |
| Wheel rim | Mild steel | 7850 | 133.5 | 0.442 | 26.1 |
| Tyre | Rubber & steel wire | 773 (calculated) | 189 | 0.801 | 121.3 |
| Assembly | - | - | 363.3 | - | **150.3** |

The resultant tyre density from the calculations appeared to be too low, natural rubber having a density of approximately $1000kg/m^2$ (Ashby, 1999) with the wire content in tyres increasing this. The likely error was probably overestimating the tyre wall thickness and in accounting for the tread, but in the absence of any further information it was decided to proceed using the generated model data.

The equivalent total driveline inertia at the flywheel in each gear, as a result of two rear wheels and tyres is shown in Table 6.11; these values being calculated using Equation 6-13. In the lower range of transmission gears, wheel inertia is negligible. Also shown is the proportion of measured inertia accounted for by the wheels and tyres. This was between 40% and 50% of the total driveline inertia. It was therefore decided to estimate the total driveline inertia for the highest three gears from the measured data.

*Table 6.11 – Flywheel equivalent total inertia of the rear wheels in each gear*

| **Gear** | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| **$I_{WF}$ ($kg.m^2$)** | 0 | 0.01 | 0.01 | 0.01 | 0.02 | 0.03 | 0.04 | 0.06 | 0.05 | 0.08 | 0.11 | 0.17 | 0.28 | 0.42 | 0.63 | 0.95 |
| **$I_{WF}$ as a % of measured $I_{DF}$** | | | | | | | | | 42 | 42 | 44 | 47 | 51 | - | - | - |

Subsequent to the completion of this work, tyre inertia was confirmed to be 107.5kg.m$^2$ and the wheel rim and dish inertia from the supplier model was confirmed as 42.55kg.m$^2$ (Moncelsi, 2005). This suggests the wheel material was of a higher density. As the overall inertia of their wheel and tyre was 150.05kg.m$^2$, almost perfectly matching this result, no recalculations were necessary.

### 6.6.6 Calculation of Driveline Inertia for all Upper Gear Ratios

The experimental data provided inertias for gears 9 to 13 and for a wheel. The transmission was considered as a series of lumped inertias (see Figure 6-29) which approximately represented the different speed ratios through the transmission. The transmission inertias ($I_{TA}$, $I_{TB}$ and $I_{TC}$) represented the front, middle and rear sections of the transmission (as previously defined in Section 3.4.1). In addition, an inertia represented the differential and epicyclic components of the rear axle ($I_D$); the previously determined wheel inertia ($I_W$) completed the representation.

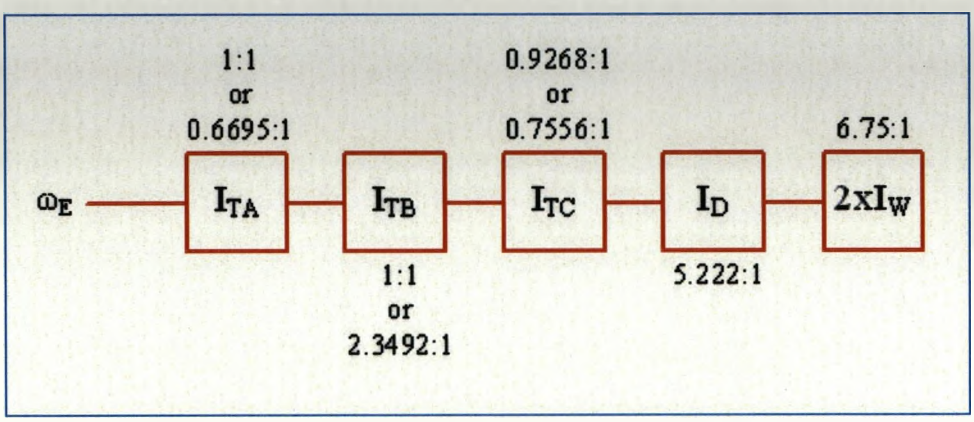



*Figure 6-29 – The lumped inertias in the upper gear range (ratios relative to previous lumped inertia in the chain)*

By summing these inertias using the equivalent inertia principle defined in Equation 6-13, it was possible to generate five equations, one for each of the gears 9 to 13 (see Appendix A5.7). As $I_W$ was already known, this was subtracted from the equations. The final equations (in matrix form) are shown in Figure 6-30:

$$\begin{bmatrix} 1 & 0.1812 & 0.2110 & 0.0077 \\ 1 & 0.1821 & 0.3174 & 0.0117 \\ 2.231 & 0.4043 & 0.4706 & 0.0173 \\ 2.231 & 0.4043 & 0.7081 & 0.0260 \\ 1 & 1 & 1.1642 & 0.0427 \end{bmatrix} \bullet \begin{bmatrix} I_{TA} \\ I_{TB} \\ I_{TC} \\ I_{D} \end{bmatrix} = \begin{bmatrix} 0.0690 \\ 0.1133 \\ 0.1362 \\ 0.1889 \\ 0.2686 \end{bmatrix}$$

*Figure 6-30 - Driveline inertia matrix*

This over-constrained matrix was solved using a least-squared, non-negative MATLAB function. The determined values of $I_{TA}$, $I_{TB}$, $I_{TC}$ $I_D$ and $I_W$ were then used to estimate the inertias of all eight upper gears. The results are presented in Table 6.12. The good correlation with the measured results can be seen in Figure 6-31.

*Table 6.12 – Final calculated driveline inertia (including rear wheels) in each upper transmission gear ratio*

| Gear | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 |
|---|---|---|---|---|---|---|---|---|
| Calculated $I_{DF}$ | 0.12 | 0.16 | 0.26 | 0.37 | 0.55 | 0.82 | 1.23 | 1.83 |

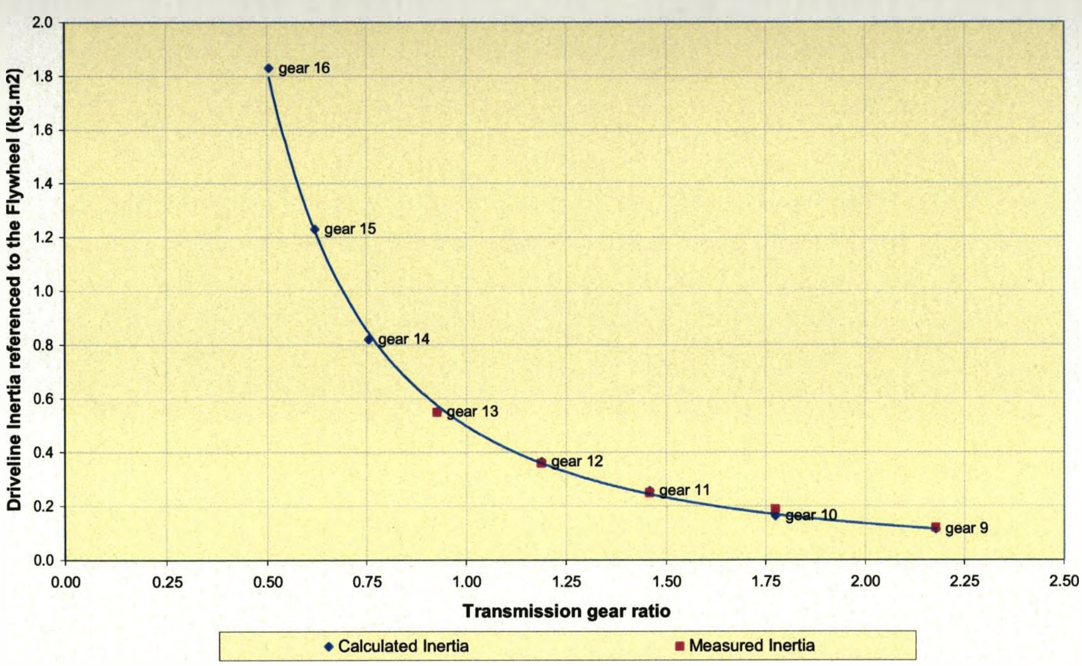



*Figure 6-31 – Calculated and measured driveline inertia for each upper gear ratio*

This method has provided a very good correlation between measured and calculated driveline inertia. For consistency calculated values have been used for subsequent work.

## 6.7 Engine Power Boost

During field trials, the power boost feature was found to operate differently to its intended design. The Axle dynamometer (in conjunction with a P.T.O. dynamometer) was used to further investigate the engine power boost feature. This is discussed in Section 9.

## 6.8 Overall Summary

It was found to be possible to replicate field loads on the axle dynamometer facility in a controlled manner. However, it was necessary to make some provision for driveline losses when taking flywheel torque data measured in the field and converting it to applied axle torque values.

The axle dynamometer was used to apply steady state axle torques to investigate the losses through the driveline and therefore determine its efficiency. It was found that, for this transmission configuration, speed related losses are the most significant. Torque load and number of meshed gears were also significant. The data was used to generate a model to calculate flywheel torque from axle torque, transmission output speed and number of meshing gear pairs transmitting torque in the transmission. This model was developed to be incorporated into the overall vehicle model (see Section 4.4.3.2).

Through a means of experimental and theoretical calculations it was possible to determine the test tractor rear wheel inertia and subsequently the driveline inertia in each of the upper transmission gear ratios. The tractor wheels and tyres form up to 50% of the total equivalent driveline inertia when referenced to the flywheel. The inertia data was then used as part of the overall vehicle model (see Section 4.4.3.1).

# 7 Model Validation

## 7.1 Engine Validation

### 7.1.1 Introduction

Due to the modular construction of the tractor-implement model, it was possible to determine the correct operation of most of the sub-systems during the model construction process. The complex nature of the engine model, together with the presence of an integration block, resulted in the need for a more rigorous validation process, through comparison of model and actual engine data.

A number of different torque-speed regimes were considered during the engine model validation process. The proportional and integral settings for the fuel controller sub-model were also determined during this process ($k_1$=0.04, $k_2$=0.06). This was performed using a trial and error method, to match the response of the engine model to the actual engine response on a P.T.O. dynamometer.

### 7.1.2 No-load, Throttle Adjustments

With no external sources of loading, the test tractor foot throttle position was adjusted by the operator over a period of approximately three minutes. The time, throttle position and engine speed were sampled at 100Hz and recorded at 10Hz. This data was used to generate a model test input file to give a throttle setting at each time step.

The engine sub-system was isolated from the remainder of the model and the test input file was connected in place of the foot throttle % block (see Figure 4-3). When the simulation commenced, the foot throttle position, and therefore the set point engine speed, was controlled by the data recorded from the test tractor. The engine speed was averaged to 10Hz during recording from the model and then compared with the actual engine speed recorded from the test tractor (see Figure 7-1).

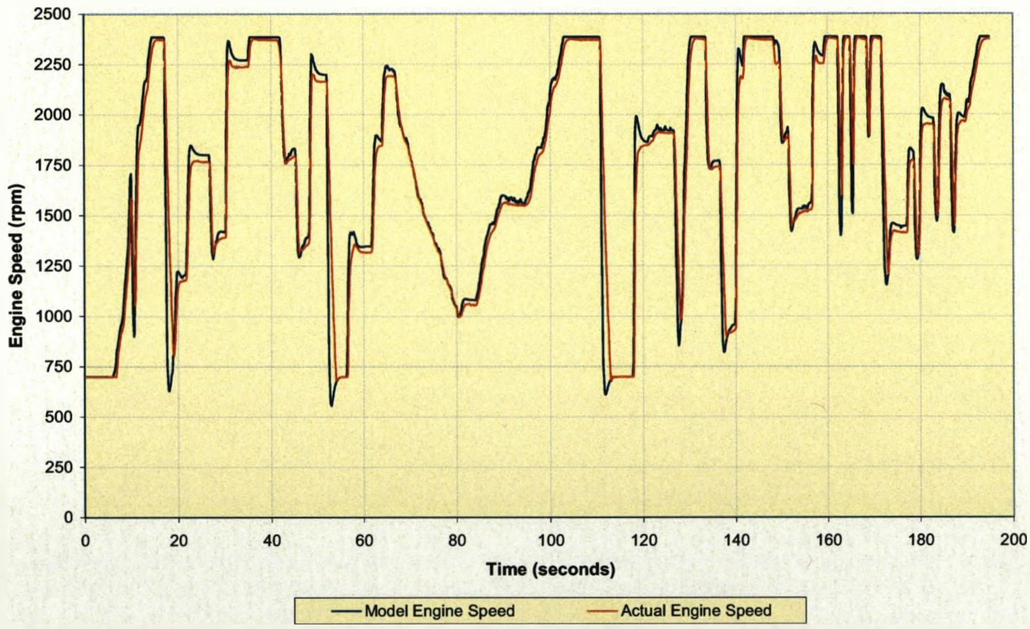



*Figure 7-1 – Test tractor and model engine speed response to changes in throttle setting with no applied torque*

There was a high correlation (0.97) between the two engine speed signals. Nonetheless, there was some error from the model, resulting in a R.M.S. engine speed error of 140rpm. The main sources for error occurred when engine speed was changed extremely rapidly, when the model tended to overshoot the true engine speed. This was partly as a result of the basic fuel controller used for this model. However, it is unlikely during normal tractor operation that the throttle setting will be adjusted so rapidly in a no-load situation, and so the error was considered to be acceptable.

### 7.1.3 Load and Throttle Adjustments

The test tractor engine was loaded by means of an eddy current P.T.O. dynamometer. A constant load was set and then engine speed adjusted by changing the foot throttle position. The change in throttle position caused the load from the dynamometer to alter as the unit's rotational speed changed. During the test throttle position, engine speed and flywheel torque were sampled at 100Hz and recorded at 10Hz.

The recorded data was used to generate a model test input file, in this instance giving a throttle setting and a torque load at each time step. The throttle setting and torque load from the tractor test were connected to the appropriate parts of the model and the simulation run. The resultant engine speed and flywheel torque from the model were recorded and compared with the actual engine speed and flywheel torque recorded from the test tractor (see Figures 7-2 and 7-3).

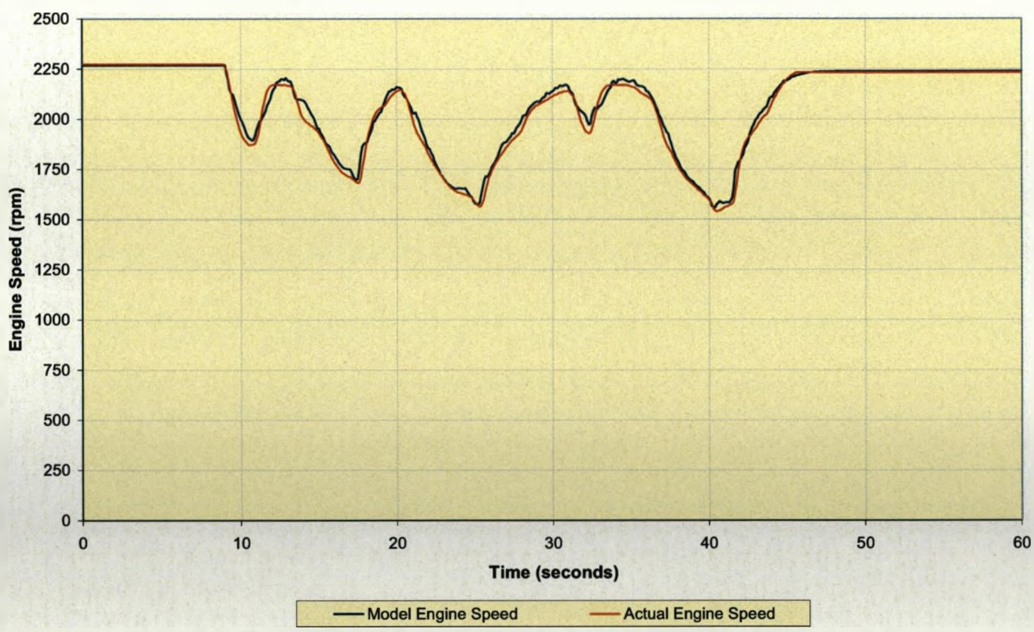



*Figure 7-2 – Test tractor and model engine speed in response to throttle and load variation*

The correlation between the two engine speeds was higher (0.99) than the no-load test, as a result of the engine being under moderate load and therefore not subject to the same high rates of engine speed change as were experienced in the no-load test. Consequently the R.M.S. engine speed error was reduced to 28rpm.

There was a small variation between measured and simulated flywheel torque, although the simulation generally followed the measured data. The correlation between the two was high (0.97), with a R.M.S. flywheel torque error of 13Nm.

It was possible to adjust the response of engine speed and flywheel torque in the model by adjusting the parameters in the PI controller. It was therefore possible to reduce the dynamic error by increasing the proportional term, but at the expense of increased torque and speed overshoots during rapid changes. The final values used were selected to give the optimum dynamic response to speed and load changes.

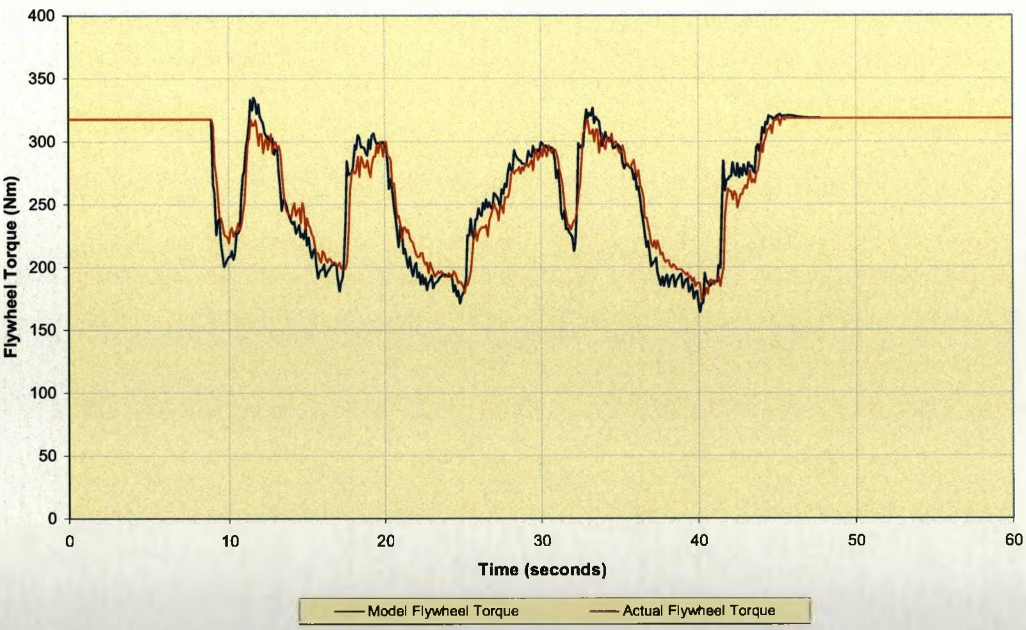



*Figure 7-3 – Test tractor and model flywheel torque response to load and throttle variation*

### 7.1.4 Full Throttle, Load Adjustments

The final part of the engine simulation was to consider performance of the engine when operating on the full-load curve. The test tractor throttle was set to maximum and a varying load applied using the P.T.O. dynamometer. During the test, engine speed and flywheel torque were sampled at 100Hz and recorded at 10Hz.

The recorded data was used to generate a model test input file, whereby the torque load at each time step formed the input to the model. The resultant engine speed and flywheel torque data from the model were recorded and compared with the actual engine speed and flywheel torque from the test tractor (see Figures 7-4 and 7-5).

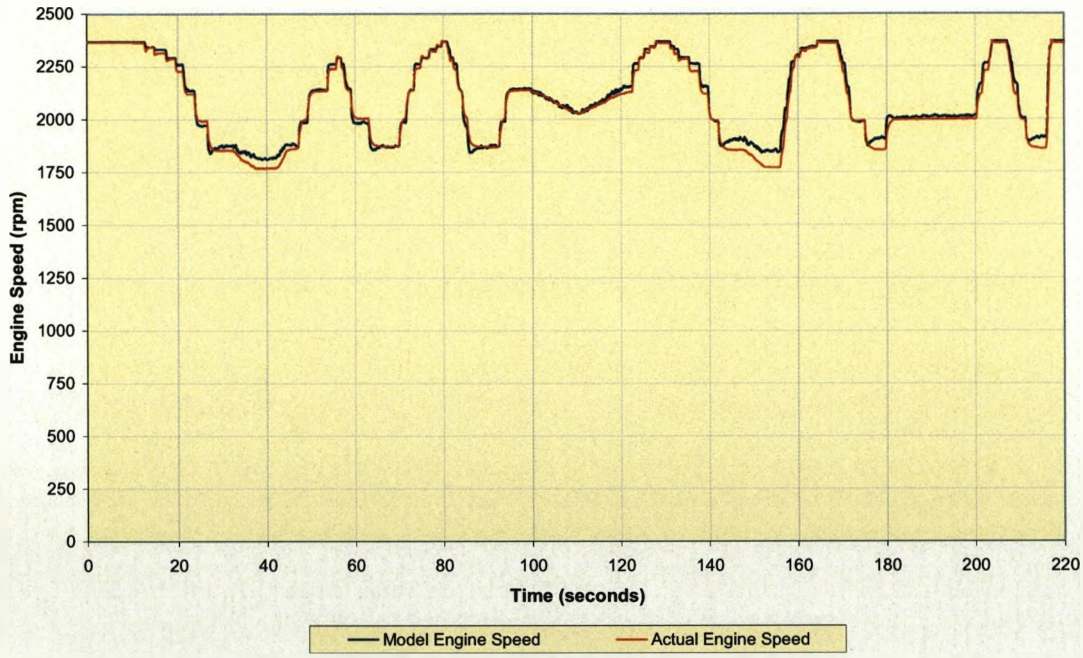



*Figure 7-4 – Test tractor and model engine speed response to changes in applied torque at maximum throttle setting*

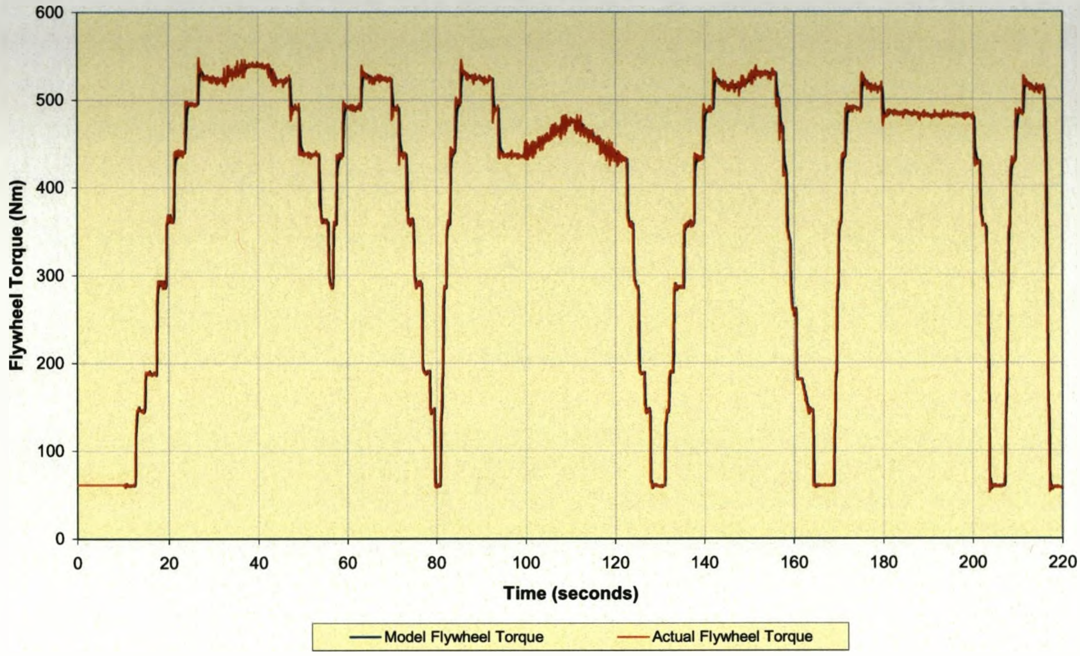



*Figure 7-5 – Test tractor and model engine torque output in response to changes in applied P.T.O. torque*

Again, the correlation between the two engine speeds was very high (0.99), with a R.M.S speed error of 23rpm. Similarly the correlation between measured and predicted flywheel torque was 0.99, with a R.M.S. torque error of 10.8Nm.

### 7.1.5 Governor Droop

The droop characteristics of the model were compared to the test tractor response measured on the P.T.O. dynamometer during flywheel torque calibration (see Figure 3-6). Comparisons were made across the working range of the engine under steady state loading. The results of the comparison are presented in Table 7.1: it can be seen the model and the tractor engine have very similar response characteristics.

*Table 7.1 – Test tractor and model speed droop effects (5% droop setting)*

| No-load engine speed (rpm) | Flywheel torque (Nm) | Tractor engine speed (rpm) | Model engine speed (rpm) |
|---|---|---|---|
| 2200 | 300 | 2155 | 2155 |
| 2200 | 400 | 2135 | 2138 |
| 2000 | 300 | 1956 | 1956 |
| 2000 | 450 | 1930 | 1930 |
| 1800 | 300 | 1758 | 1755 |
| 1800 | 500 | 1720 | 1721 |
| 1600 | 300 | 1556 | 1553 |
| 1600 | 500 | 1519 | 1519 |
| 1400 | 300 | 1360 | 1355 |
| 1400 | 500 | 1323 | 1322 |

### 7.1.6 Summary

The engine model validation undertaken has shown that the engine simulation is an accurate representation of the actual tractor's engine characteristics across the full torque-speed operating range. There are some small differences between the model and the test tractor, but given the model was constructed with theoretical engine data and the test tractor engine is a 'real' system, these differences are deemed to be within acceptable limits.

## 7.2 *Tractor Implement Model Validation*

### 7.2.1 Approach

Steady state field data from Section 5 was used to determine the overall performance of the tractor-implement model. A selection of datasets from both ploughing and power harrowing were included in the validation, covering the operating envelope considered during the field investigation.

Each field dataset was reduced to include only the parameters required for the model and for comparison with the model output. For ploughing this included time, draught force, forward speed, gear, throttle setting, engine speed and flywheel torque. For power harrowing this was time, draught force, forward speed, gear, throttle setting, engine speed, flywheel torque and P.T.O. torque. The dataset was then saved as a test input file in the MATLAB workspace, for subsequent use in the simulation.

### 7.2.2 Ploughing

For the ploughing validation, field-measured draught force was substituted into the model in place of the model-generated draught force. With the correct gear and throttle setting, the simulation was run for the period of available data (120 seconds) and the output, both from the model and the field dataset, for engine speed, forward speed and flywheel torque were recorded.

Three example datasets are presented:

- sandy soil, gear 5, 356mm furrow width (see Figures 7-6 and 7-7);
- sandy soil, gear 5, 406mm furrow width (see Figure 7-8);
- sandy soil, gear 7, 356mm furrow width (see Figures 7-9 and 7-10).

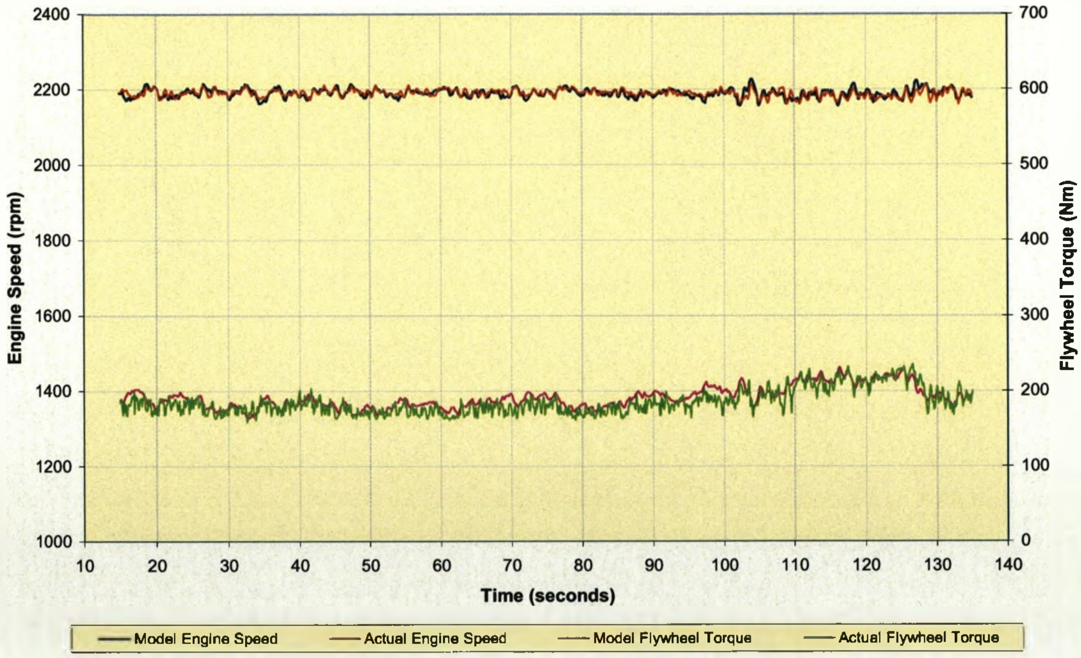



*Figure 7-6 – Model and field data comparisons of engine speed and flywheel torque whilst ploughing sandy soil (gear 5, 356mm furrow width)*

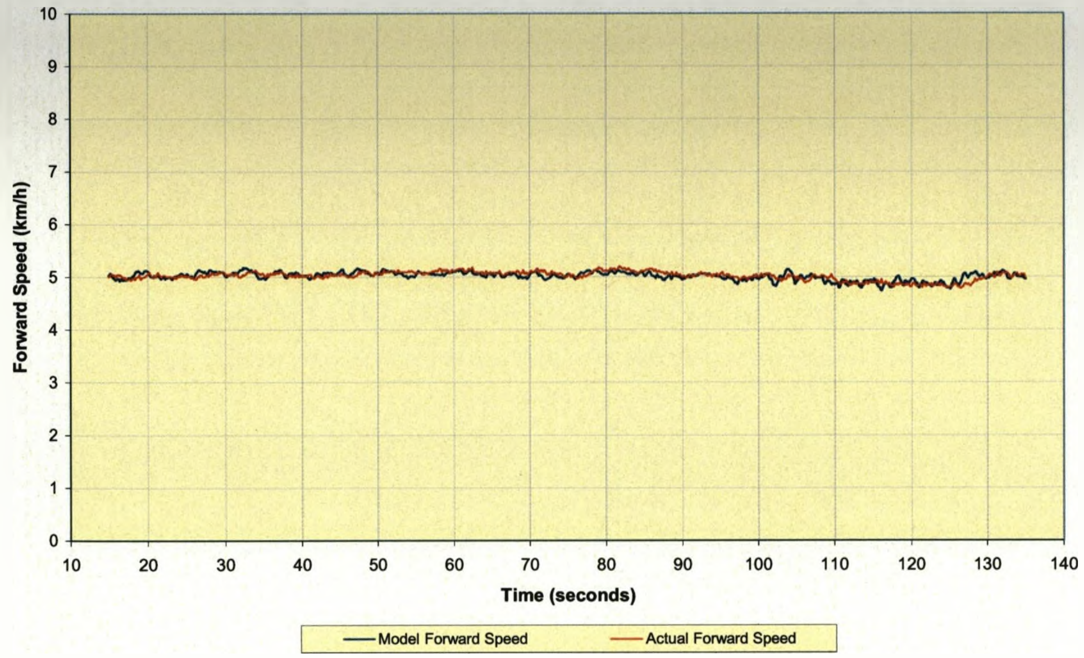



*Figure 7-7 – Model and field data comparisons of true forward speed whilst ploughing sandy soil (gear 5, 356mm furrow width)*

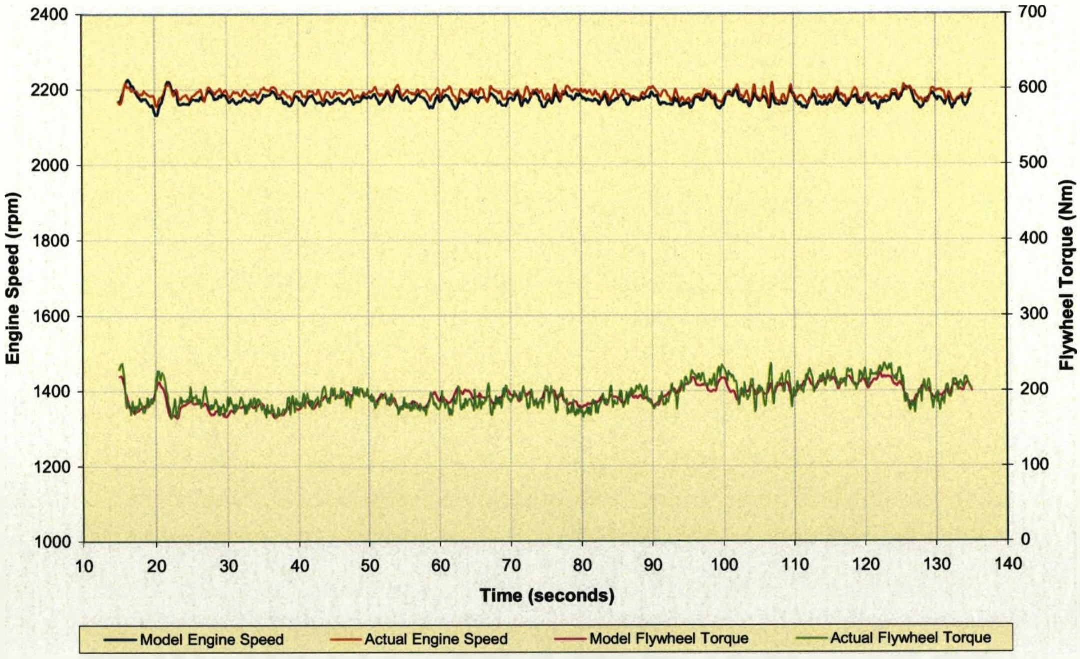



*Figure 7-8 – Model and Field data comparisons of engine speed and flywheel torque whilst ploughing sandy soil (gear 5, 406mm furrow width)*

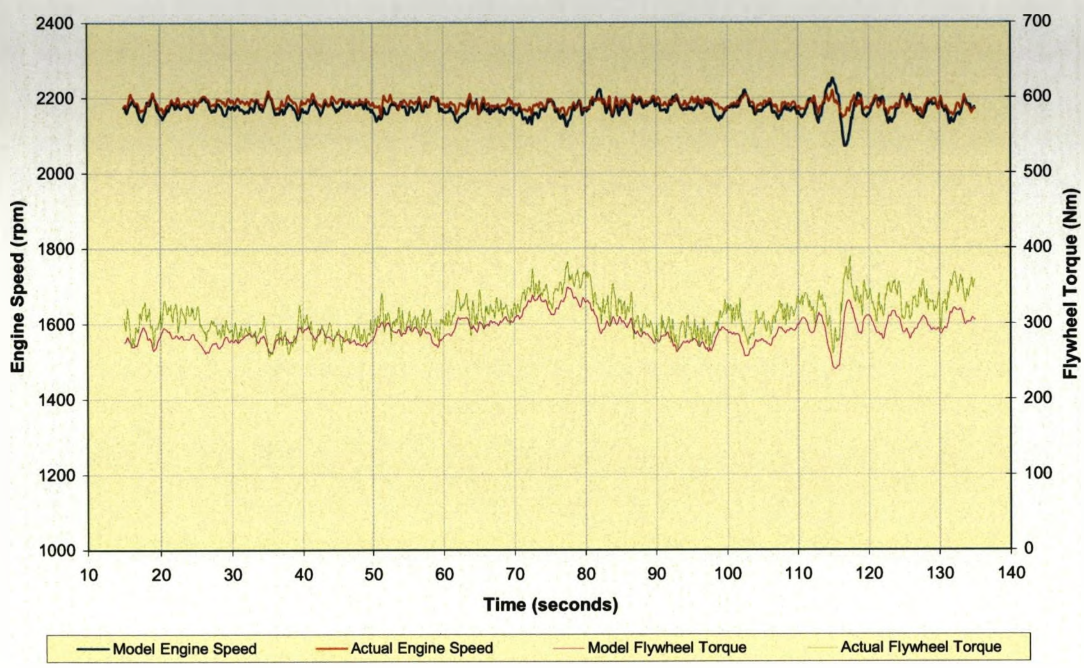



*Figure 7-9 – Model and field data comparisons of engine speed and flywheel torque whilst ploughing sandy soil (gear 7, 356mm furrow width)*

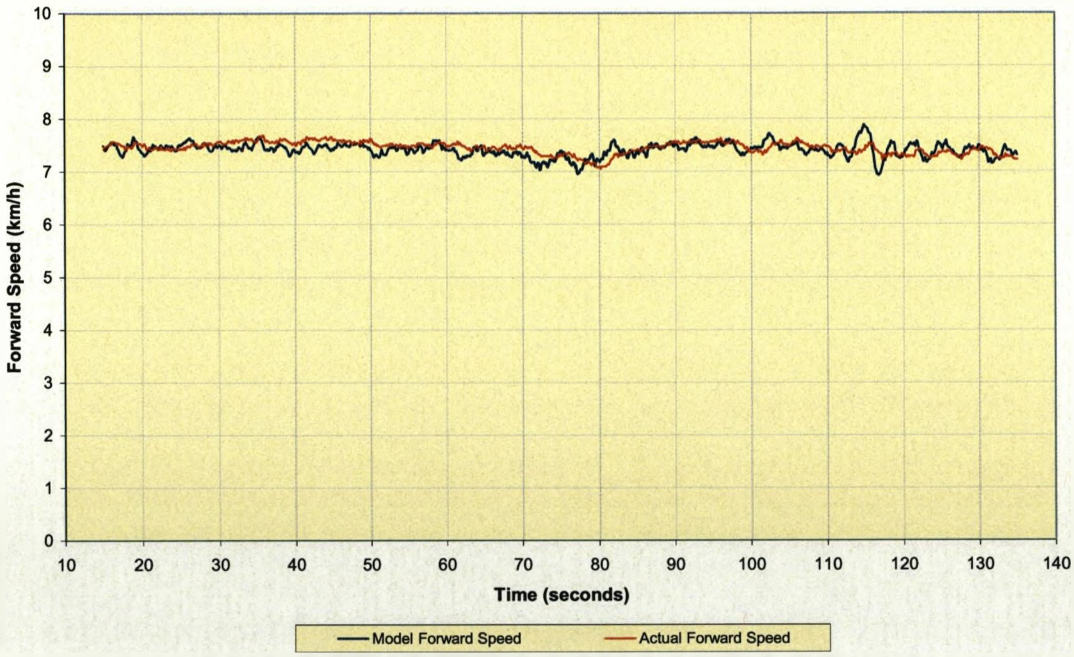



*Figure 7-10 – Model and field data comparisons of true forward speed whilst ploughing sandy soil (gear 7, 356mm furrow width)*

The plough validation data shows the model is a good representation of the tractor-implement system, given a known dynamic draught requirement from the implement. Engine speed closely followed (within 1%) that which was measured in the field, especially for the tests in the lower transmission gear ratio (see Figures 7-6 and 7-8). Flywheel torque followed the overall trend that the field data exhibited (within 5%); however the model showed less variation than the field data. For gear 7 (see Figure 7-9) flywheel torque magnitude was higher for the field measured data. Given draught force is the same for the measured and simulated implement, the additional torque load is probably as a result of in-field variation in slip and rolling resistance. The lower magnitude and variation experienced by the model is due to the use of a single coefficient of rolling resistance and the simplified slip-pull relationship employed. The additional torque deviation experienced in gear 7 also resulted in small differences in forward speed between the measured and model data (see Figure 7-10).

### 7.2.3 Power Harrowing

For the power harrowing model validation, field-measured draught force and P.T.O. torque were used in place of the model generated values. With the correct gear and throttle setting, the simulation was run and the output, both from the model and the field dataset, for engine speed, forward speed and flywheel torque were recorded.

Two example datasets are presented:

- sandy soil, gear 4, 100mm working depth (see Figures 7-11 and 7-12);
- sandy soil, gear 4, 125mm working depth (see Figures 7-13 and 7-14).

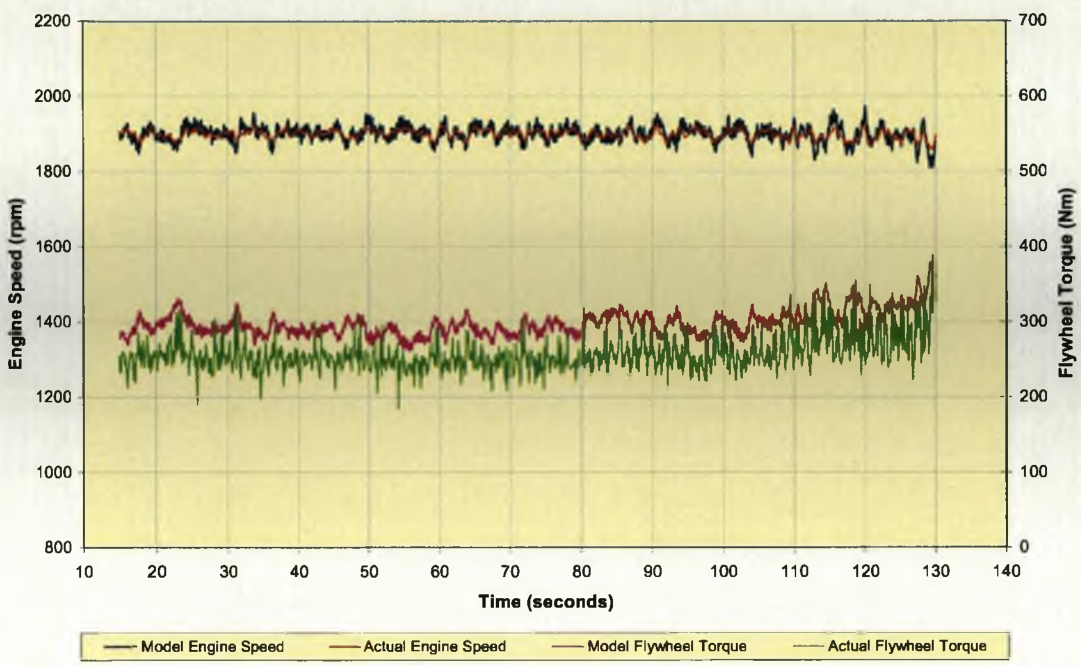



*Figure 7-11 – Model and field data comparisons of engine speed and flywheel torque whilst power harrowing sandy soil (gear 4, 100mm working depth)*

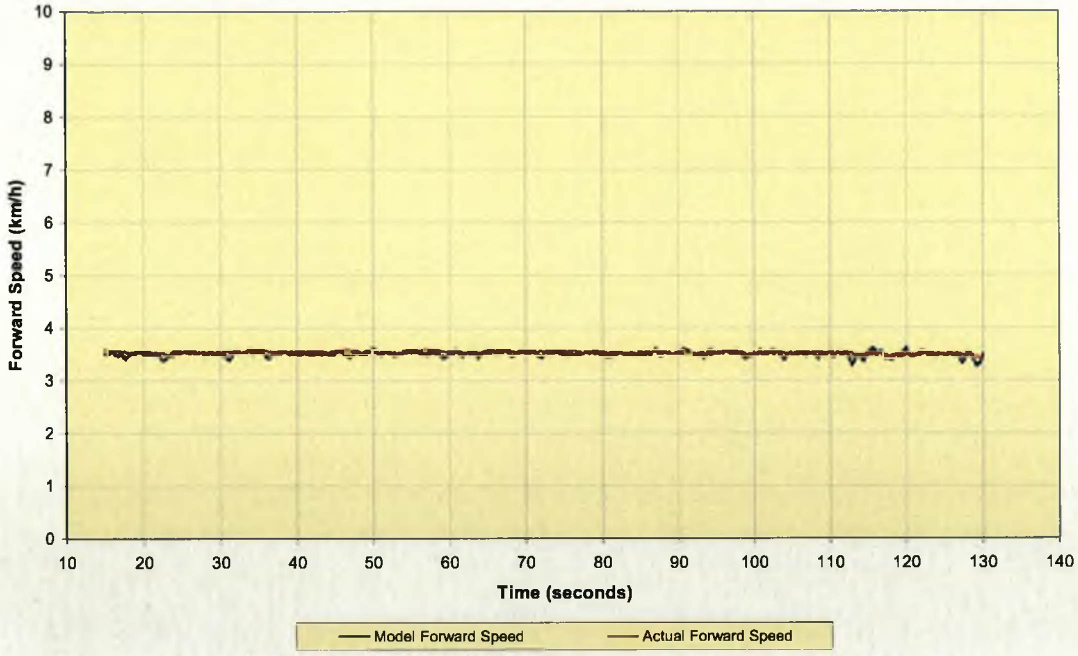



*Figure 7-12 – Model and field data comparisons of true forward speed whilst power harrowing sandy soil (gear 4, 100mm working depth)*

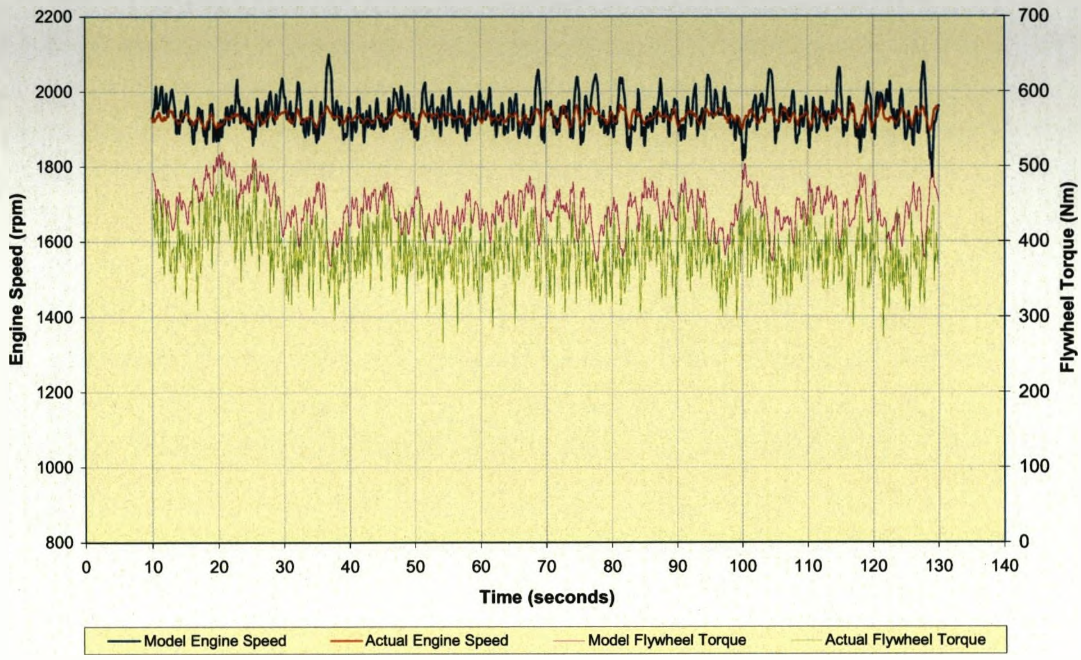



*Figure 7-13 - Model and field data comparisons of engine speed and flywheel torque whilst power harrowing sandy soil (gear 4, 125mm working depth)*

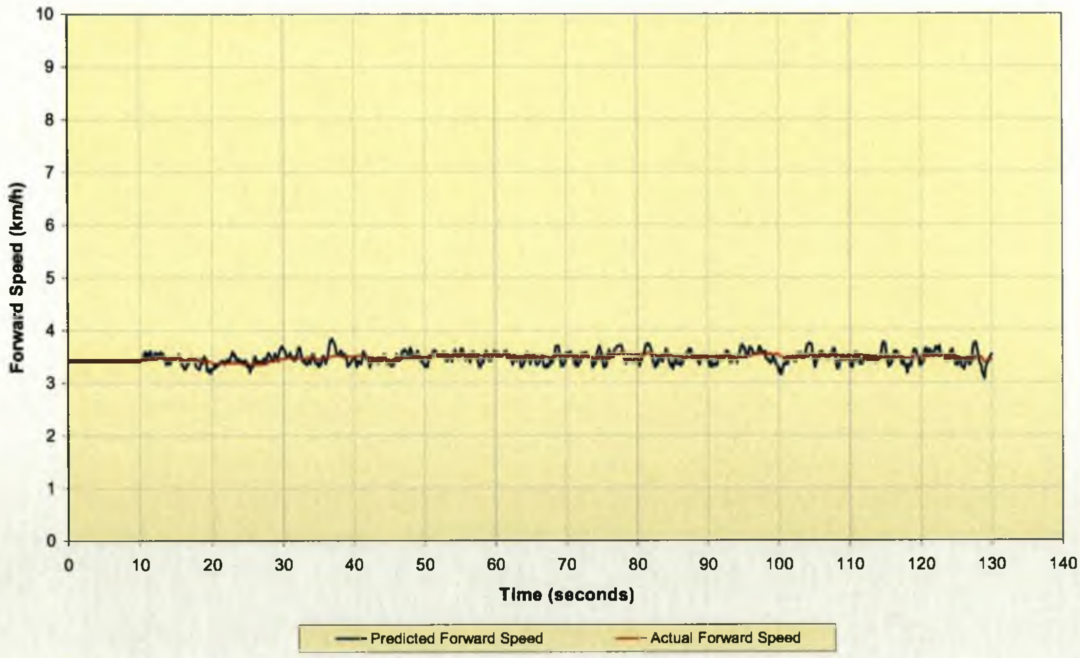



*Figure 7-14 – Model and field data comparisons of true forward speed whilst power harrowing sandy soil (gear 4, 125mm working depth)*

The model represents the response of the tractor to variations in draught force and P.T.O. requirement well, for both engine speed and forward speed in the two examples show. As can be seen from Figures 7-11 and 7-13, although the model flywheel torque follows the general trend of the field data, its magnitude is greater. After analysis of the model, it was suspected that the estimate of rolling resistance made in the initial model development was a major source of error. Rolling resistance data from a concurrent investigation with the same equipment and fields (Scarlett *et al*, 2003) was analysed. The rolling resistance force ($H_{RR}$) was calculated by:

$$H_{RR} = \frac{T_{EF} \times r_d}{r \times 1000} \qquad \textit{Equation 7-1}$$

Where $T_{EF}$ was determined from the flywheel torque recorded data, less the calculated driveline losses. The coefficient of rolling resistance ($C_{RR}$) was then calculated by:

$$C_{RR} = \frac{H_{RR}}{(M_T + M_I) \times g} \qquad \textit{Equation 7-2}$$

The mean rolling resistance coefficient on ploughed clay soil across the range of gears used was 0.093. For sandy soil the mean coefficient was 0.11. This data confirmed the over-estimation in the original model and therefore the dataset was used in the model again with the rolling resistance for power harrowing coefficient reduced to 0.1; the results being presented in Figure 7-15. The improvement in the fit of the model to the field data (flywheel torque within 10% of measured values) resulted in the coefficient being changed to 0.1 in the refined version of the model.

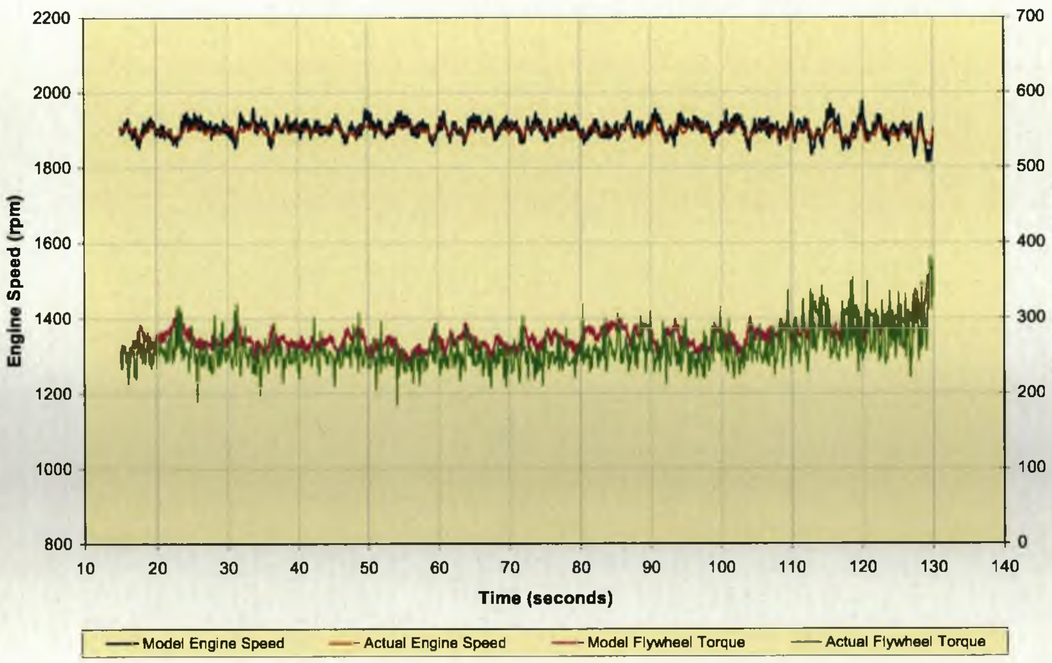



*Figure 7-15 - Model and field data comparisons of engine speed and flywheel torque whilst power harrowing sandy soil (gear 4, 100mm working depth) with the coefficient of rolling resistance = 0.1*

Both the ploughing and power harrowing validation showed greater variation in measured flywheel torque than the model. This is due to the simplification of a number of dynamic factors in the model such as the slip-pull traction relationship. Indeed, from analysis of the field data for power harrowing (see Section 5.5) there is dynamic variation in P.T.O. torque loading which is not accounted for in the power harrow model.

It is also noticeable that simulated engine speed showed slightly more variation than the test engine data. This is as a result of the relatively simplistic fuel controller in the model with a lack of filtering which is present in the actual fuel controller (see Section 4.4.2.3). Nonetheless, the model does provide a good representation of the true engine speed. Future development of the model could improve simulated engine speed control.

## 7.3 Summary

This part of the investigation has validated that the engine model and the overall model, provide a good correlation with test data. The model can therefore be confidently used for further investigation of powertrain characteristics, development of control strategies, or for comparisons between operational characteristics of powertrain components.

# 8 Commercial Considerations

## *8.1 Introduction*

The test tractor used for this investigation featured a high degree of sophistication, particularly through the use of microprocessor-based sub-system control, and a high performance modern diesel engine. In comparison the transmission design is relatively old technology which has been improved and its life extended with electronic control systems. The recent developments and performance of different transmission technologies, including power split CVTs, were outlined in Section 2. This part of the investigation focuses on the wider issues facing the project sponsor with respect to which type of transmission system is likely to dominate in the future. An analysis of the UK tractor market identified current and potential trends together with the relevant factors likely to shape future market direction, effort being focused on the UK market due to the lack of accurate data from other countries. The strategic focuses of the manufacturers themselves were considered, as well as the implications for manufacturing and dealers. Users of both powershift-type transmissions and CVTs were interviewed to identify the factors behind their purchasing decisions and the advantages or disadvantages they had found with CVTs over conventional powershift transmissions. Data from a recent Europe-wide survey of tractor buyers was used to validate the trends.

## *8.2 UK Tractor Market Analysis*

Machinery purchases in the UK account for between 33% and 50% of farmers' annual expenditure (Mintel Group, 2003) with tractors forming the major constituent. As a result the UK market for agricultural tractors (see Figure 8-1) has been directly influenced by the economic environment for farmers, itself influenced by a range of factors.

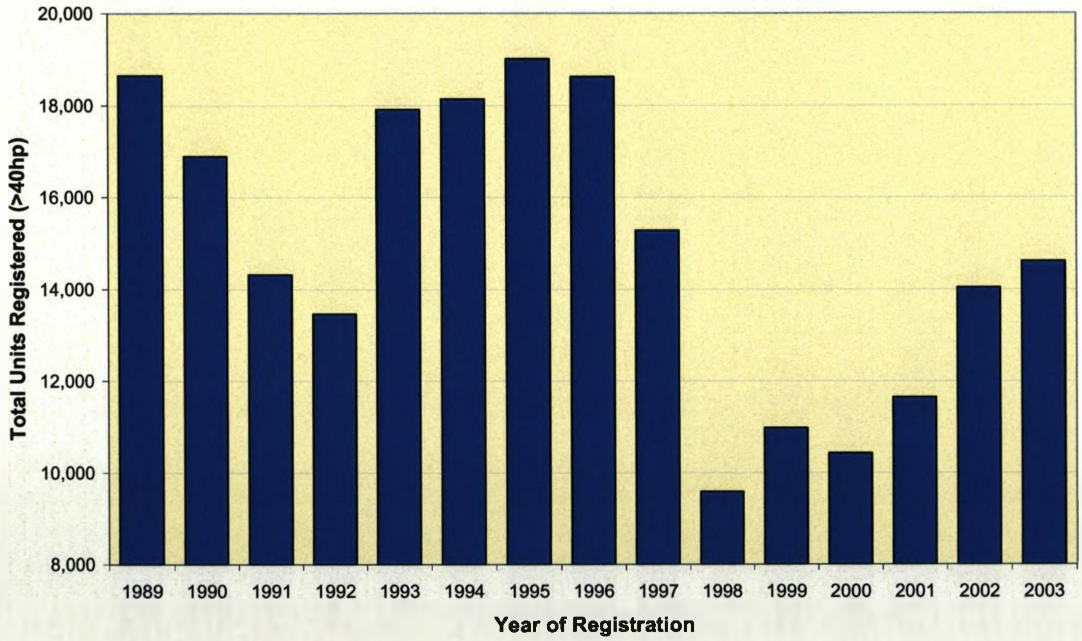



*Figure 8-1 – UK agricultural tractor sales 1989-2003 (Source: AEA statistics)*

European Union (EU) controls through the Common Agricultural Policy, specifically subsidy payments and price support for farm produce, have been a major influence on farmers' propensity to purchase tractors. Recent changes to decouple subsidies and production of agricultural products (Department for Environment Food and Rural Affairs, 2004) alongside the introduction of a single farm payment will force many farmers to review future machinery purchases as some crop growing areas become uneconomic. Lower farm incomes will reduce machinery investments, potentially reducing future demand for tractors, and the amount of money available for each purchase. The situation will mean that farmers will take every step possible to reduce their costs of production. However, the potential increase in workrates and reduction in fuel consumption provided through the use of CVT transmissions could increase the future demand.

Recent poor and volatile prices of agricultural produce reduced the demand for tractors, particularly during 1998 to 2000 when prices were particularly depressed. Grain prices have since been erratic with feed wheat less than £60/tonne in 2002 and at the beginning of 2005, whilst reaching over £100/tonne at the end of 2003. This price recovery, together with compensation payments for the 2001 foot and mouth crisis helped to stimulate tractor demand during the last three years.

Overall demand for agricultural tractors has been negatively affected by the decline in the number of farms in the UK and the increasing average size of the remaining farms. Whilst the quantity of tractors has reduced, larger farm sizes have resulted in a continual increase in the average tractor engine power (see Figure 8-2). Although this has recently stabilised, the proportion of tractors between 75kW and 120kW has continued to rise, accounting for only 14% of total sales during 1989 and over 50% in 2003. Larger farms tend to be focused on improving productivity and with a larger area to spread costs over, could potentially increase the future demand for CVT tractors provided the purchase cost is not prohibitive.

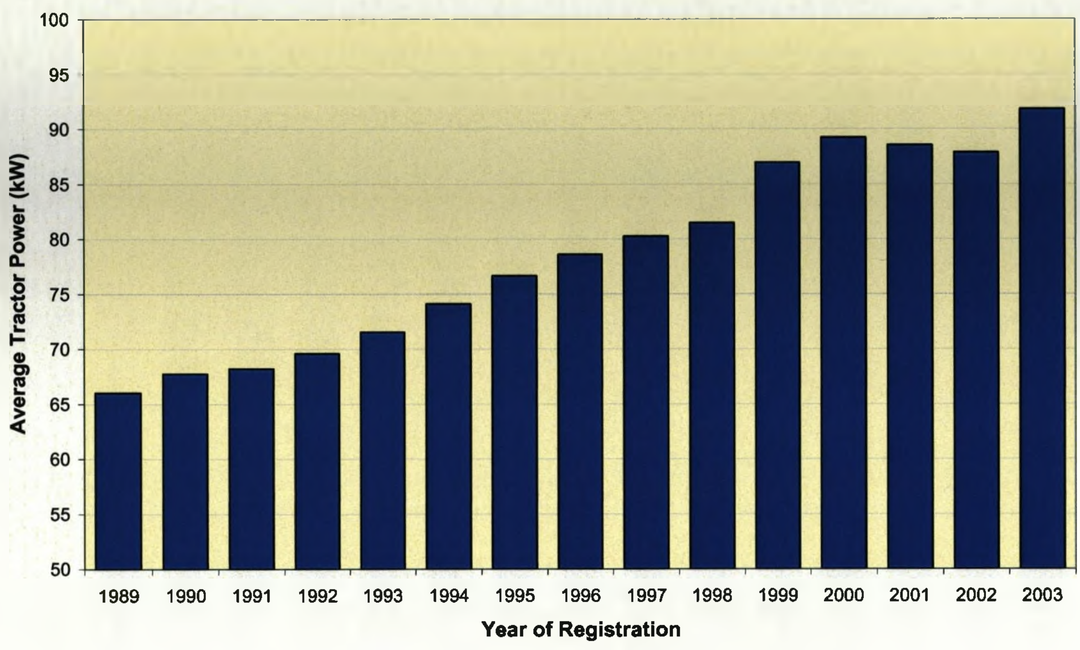



*Figure 8-2 – Average tractor engine power (1989-2003) (source: AEA statistics)*

The proportion of tractors featuring CVTs was difficult to determine as manufacturers only provide total sales and engine power information to the Agricultural Engineers Association (AEA). CVT tractor sales for 2001-2003 have been estimated using a variety of sources, including personal communication with marketing representatives (see Table 8.1). The proportion of CVTs was likely to be less than 5% of total sales in 2003. Total UK sales in 2004 rose to 14,955 units. Full year sales for the Massey Ferguson CVT and the introduction of the New Holland CVT, together with organic sales growth meant it was quite likely total CVT sales in the UK would have reached 5% of the UK total sales in 2004. However information on sales by manufacturers is retained for one year meaning a more accurate analysis was not possible.

*Table 8.1 – Estimated size of the historic UK market for CVT tractors*

| Manufacturer | Total Sales | | | Estimated CVT sales | | | % of Manufacturers Total | | |
|---|---|---|---|---|---|---|---|---|---|
| | 2001 | 2002 | 2003 | 2001 | 2002 | 2003 | 2001 | 2002 | 2003 |
| Fendt | 172 | 220 | 314 | 172 | 220 | 314 | 100 | 100 | 100 |
| CNH (Case/Steyr) | 4,228 | 4,171 | 4,680 | 80 | 85 | 70 | 1.9 | 2.0 | 1.5 |
| John Deere | 3,302 | 4,110 | 3,939 | - | 102 | 200 | - | 2.5 | 5.1 |
| Massey Ferguson | 2,012 | 2,615 | 2,299 | - | - | 58 | - | - | 2.5 |
| | | | | | | | % of Total UK tractor Sales | | |
| Combined sales (all manufacturers) | 12,449 | 14,761 | 15,043 | 252 | 407 | 642 | 2.0 | 2.8 | 4.3 |

The future developments of the UK tractor market are likely to be a continuation of recent trends. The continual effects of the single farm payment is likely to result in further consolidation of the industry, leaving larger farms focussed on producing crops at world prices. This focus will further stimulate demand for fewer, high-horsepower tractors which are efficient and maximise productivity. The requirement to reduce costs could rule out CVT tractors at their current prices. The 2004 EU enlargement is unlikely to have any short-term profound effects on UK farm incomes, but the huge potential area for crop production could, in time reduce the profitability and indeed viability of some parts of UK agriculture, leading to a reduction in demand for tractors in the UK. Admittedly, this increase in production in the accession countries would require considerable investment in tractors to increase production to levels to influence grain prices, but it is unlikely these will be high technology CVT tractors at the outset. Recent issues concerning the use of reduced duty ‘red’ diesel during road operation and the potential to reduce or even remove duty assistance will continue to focus tractor manufacturers on producing more efficient powertrains.

## 8.3 Tractor Pricing Strategies

The actual purchaser-paid price differences between a CVT and a powershift-type transmission is difficult to determine as discounts, specification differences and trade-in valuations all influence the actual price paid. Comparisons of book prices (see Table 8.2) shows that the CVT premium varies between £6,000 and £16,500 for similar models around 100kW rated engine power (10-15%). John Deere and Massey Ferguson (MF) models are the most comparable as they are the same basic tractor with a different transmission. The Case (and other CNH) CVT models are a separate tractor range making comparisons difficult. The Fendt CVT is priced above all other tractors, reflecting the higher tractor specification and the perceived value of the brand.

*Table 8.2 – Price and specification differences between tractors with CVT and powershift-type transmissions around 100kW engine power*

| Manufacturer | Model | Rated Engine Power (kW) | Boosted Rated Power (kW) | Transmission (Powershift Speeds) | Max Speed (km/h) | Book Price ex VAT (£) |
|---|---|---|---|---|---|---|
| Case | MXU135 | 101 | 119[a] | 16x16 (4) | 40[b] | 41,759 |
| Case | MXM130 | 98 | - | 18x6 (6) | 40 | 39,230 |
| Case | CVX1135 | 102 | - | CVT | 50 | 49,421 |
| John Deere | 6820 | 101 | - | 20x20 (4) | 40[c] | 55,877 |
| John Deere | 6820 AP | 101 | - | CVT | 50 | 62,058 |
| AGCO (MF) | 6475 | 101 | 108[d] | 32x32 (4) | 40 | 45,450 |
| AGCO (MF) | 7475 | 101 | 108[d] | CVT | 50 | 61,900 |
| AGCO (Fendt) | 714 | 103 | - | CVT | 50 | 71,250 |

a) Top 4 Gears and PTO only
b) 50km/h 17x16 transmission optional extra
c) 50km/h optional extra
d) Top 2 Gears only

## *8.4 Corporate Perspectives*

### 8.4.1 CNH Product Marketing

A group discussion was held with seven members of the CNH product marketing team, who between them were responsible for all tractor products for Europe, Africa and Asia. The purpose of the session, summarised below, was to ascertain their views on the importance of CVT tractors at present to the market and CNH, and to identify the likely future direction of the tractor market.

CVT tractors accounted for 10% of all European sales above 90kW last year according to CNH market estimates, and this percentage is increasing year on year. Western Europe is at present the principle market for CVT tractors. North America is not presently interested in this transmission type, mainly due to the 'broadacre' style of agriculture.

The key advantage of CVT tractors identified was the ability to infinitely change forward speed independently of engine and P.T.O. speed, making the tractor particularly suitable for applications relating to potentially high yielding root crops. CVT tractors were not promoted for acceleration and transport applications as, although there was some disagreement, the consensus of opinion was that a CVT transmission was less efficient than a conventional semi-powershift transmission at road speeds. Driveability was the other major benefit of a CVT tractor, with the transmission controller having the ability to 'filter' bad driver behaviour, protecting the implement from shock loading.

The future direction of tractor transmissions was an area which drew differing views. The majority of the group felt CVTs will eventually become the main technology used for agricultural tractors in the European market above 75kW engine power, for the reasons discussed, and because it provided a better platform to develop towards fully automated tractors. The remaining group members felt the inherent complexity, cost and inefficiency, as well a lack of actual need for infinite variable speed in many farming situations, would result in a move towards simple, electronically-controlled stepped transmissions. All agreed the CVT proportion of the market would become more significant as the unit cost of production reduced with increasing volumes - a situation not helped at CNH through the purchase of the CVT as a complete assembly (see Section 8.7) and because only 4% of CNH's total world tractor production being CVTs.

Historical brand image of some CNH tractors as well as other post-merger priorities resulted in a strategy whereby the CVT was initially restricted to its original Steyr and Case brands. The strategy of marketing products under three separate brands whilst maintaining the common platform approach to save costs has resulted in a New Holland CVT tractor being introduced during 2004, this factor alone helping to increase the proportion of future sales being CVTs.

Future drivers for transmission technology would be primarily cost and fuel efficiency, with the person buying the tractor often now driving it for longer periods, thus requiring better ride comfort and ease of operation.

### 8.4.2 CNH Tractor Dealer

An interview was held with the Doe Power group Case sales manager, Gerald Silvey, to determine the dealer perspective on the CVT and powershift issues. Doe Power is an independent dealer selling Case tractors through six branches across East Anglia. In 2003 10% of group sales were CVT tractors, a proportion which has increased every year. The view was that CVT sales could be increased substantially if the price was reduced from the present premium because customers liked the product and the flexibility of the tractor, but most non-sales were as a result of price or complexity of operation (a misconception about the CNH CVT which was rarely borne out once the tractor was driven). For this reason, the primary method of promotion of CVT tractors was through on-farm demonstrations where a member of the training team would initially accompany the tractor to explain the principles of operation and the basic controls to obtain optimum performance, before leaving the tractor with the farmer. Promotion was also undertaken through the annual Doe show along with an annual sales brochure, two channels which most dealers would not have at their disposal. Whenever possible trips to the CVT factory in Austria were arranged for recent and potential customers to try to maintain and extend brand loyalty.

When asked to identify key selling points of a CVT tractor the main factor was the ability to set any desired forward speed, together with easy to operate controls, especially compared to Fendt CVT controls. Selling CVT tractors resulted in the need to have specialist transmission training to allow a good backup service to be offered, although this was not more excessive than other modern products. It was predicted that the future would see significant increases in CVT tractor sales, but the transmission would only become dominant if a simple, less expensive version could be produced.

## 8.5 User Perspectives

As discussed (see Section 8.2), the number of CVT tractors is small but becoming more significant. Due to time constraints and the relatively limited numbers of CVT tractors in existence, detailed face to face interviews were undertaken with four owners who operated Case CVT tractors in addition to tractors with powershift-type transmissions to determine the reasons for choosing tractors fitted with a CVT.

### 8.5.1 Methodology

Following an initial approach by the CNH customer support manager, telephone contact was made with the owners to brief them on the objectives of the interview and allow the owners to gain staff views prior to the visit (in one case the main operator also sat in on part of the interview as the owner had not used the tractor himself). The interview itself was conducted on the business premises and included an introduction to the business as well as specific tractor fleet information, particularly the CVT tractor(s) within it (see Appendix A6.1-A6.4 for full questions and transcripts of responses).

Four owners were chosen to represent different types of businesses from specialist contractors, a large farming estate and a small family farm. This methodology did have limitations as all operated Case CVT tractors, and were satisfied CVT customers. However as the purpose of the study was to determine the factors behind choosing a transmission type this was not deemed to be a serious issue. The sample size restricted statistical analysis, limiting the information to an indication of customer views.

The information presented from these interviews, as would be expected is of a subjective nature and is used to demonstrate user opinions rather than quantifiable data. Nonetheless, it remains a valid indication of the key desirable criteria of any improved powershift-type transmission control system.

### 8.5.2 Interviewee Business Profiles

Briefly outlined, the main business activities of the four interviewees were:

1. Potato Contractor - South Yorkshire

In addition to the small area (26ha) they farm themselves, two brothers run an agricultural contracting business, specialising in potato work, where they prepare ground for and plant 325ha and harvest 400ha per annum. They also undertake 1200ha of cereal cultivation work and 7300ha of spraying a year. The business has five full time employees in addition to the brothers.

2. Large Estate - Nottinghamshire

A 2065ha estate run by a farm manager, a team of eight full time employees and up to ten seasonal workers. The main unit, 900ha of sandy soil, grows salad & crisping potatoes and a variety of vegetables including carrots and onions with barley and sugar beet grown in rotation. The remainder of the estate is heavy clay soil growing wheat, rape and beans with minimum tillage techniques.

3. Grassland Contractor - Oxford

A large contracting business centred around, but not limited, to grassland contracting with work ranging from hay or haylage making and pasture maintenance for small equestrian units to baling and solid manure spreading on large farms. A joint venture also created 1000ha of mowing and raking for silage each year. In addition to the owner there are two full time employees and up to six casual workers, employed as workload dictates.

4. Family Farm - Clacton on Sea

A 365ha all-arable farm, 2/3 owned by the father and son partnership. Principle cropping on the heavy Essex clay soil is wheat, barley, peas & oil seed rape, with a small area of sugar beet (34ha) and potatoes (16ha). In addition to the father and son, the farm provides work for one full time employee, plus seasonal help as required from a local farmer's son.

### 8.5.3 Tractor Fleet Profiles & Purchasing Decisions

Table 8.3 presents an overview of the interviewees' tractor fleets.

*Table 8.3 – Respondent tractor fleet profiles*

| Business Scenario | CVT | | Full Powershift | | Semi Powershift | |
|---|---|---|---|---|---|---|
| | Number | Power range | Number | Power range | Number | Power Range |
| Potato Contractors | 4 | 96-140kW | 1 | 210kW | 2 | 90-110kW |
| Large Estate | 6 | 101-125kW | 2 | 145-200kW | 4 | 96kW |
| Grassland Contractors | 1 | 125kW | 3 | 134-149kW | 5 | 93-130kW |
| Family Farm | 1 | 140kW | 0 | n/a | 2 | 96kW-114kW |

All respondents attempted to operate new tractors for their mainline fleet with pre-defined replacement periods of between three and four years (see Figure 8-3). However all had also experienced recent expansion and as a result some tractors had been kept longer than planned. One respondent was concerned about the potential impacts of the single farm payment on his business and, as a result, was making equipment last longer and purchasing good second-hand high power tractors for limited seasonal use with large square balers.

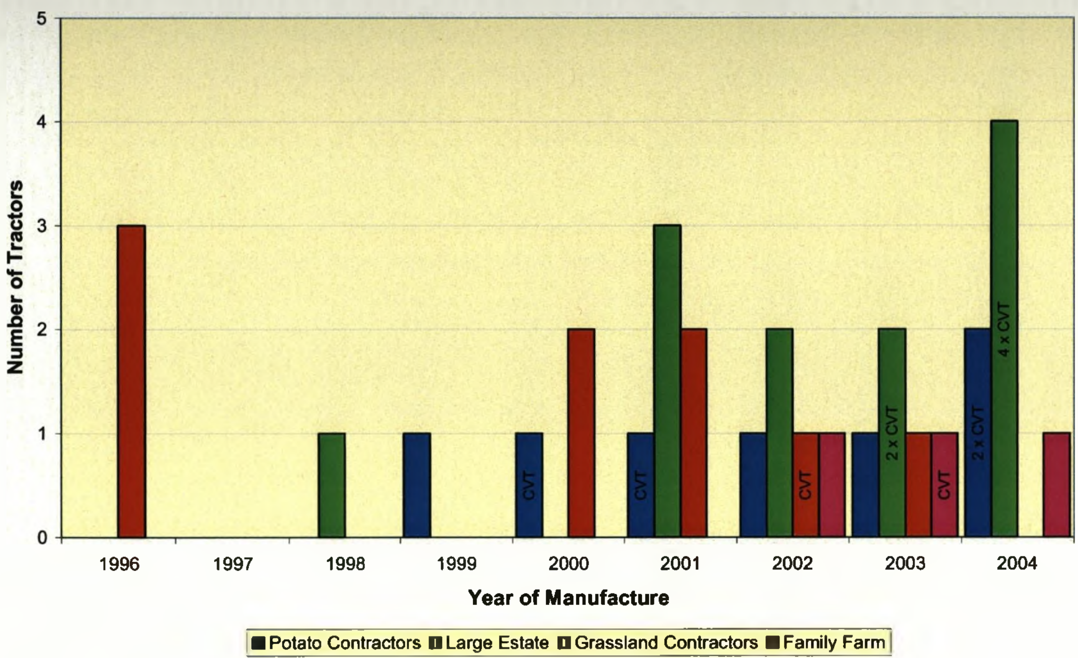



*Figure 8-3 – Respondent tractor fleet purchase data*

One respondent had recognised the potential benefits for their potato contracting business and so purchased a CVT tractor without a demonstration in 2000. The others all had a CVT tractor on demonstration alongside CNH and competitor products and found the tractor outperformed the others across a variety of operations, including ploughing, root crop harvesting, baling and road transport (the 50km/h transmission was a deciding factor for one respondent). A dislike for other current CNH products, but a high brand loyalty and a wish to maintain simplified servicing and parts stocks for a large tractor fleet was a factor in the move to CVT tractors for one respondent. It was apparent that all respondents had a high dealer and brand loyalty. Witney (1988) identified these were major factors in choosing a new tractor and causes of reluctance to switch between brands.

### 8.5.4 Tractor Selection for Specific Tasks

The two respondents with large areas of root crop work always used CVTs for operating harvesters and for bed preparation, although they also tended to use their CVT tractors wherever possible anyway. A CVT tractor was preferred for ploughing by one respondent (family farm), the grassland contractor used his CVT for most tasks, but always tried to use it with a combined baler-wrapper and a power-harrow drill. All respondents used CVT tractors for transport operations where possible. The reasons for selecting a CVT tractor for these tasks were due to ease of use in the case of transport operations, and maximisation of output for the other tasks.

With conflicting views to the other respondents, the family farm did not use their CVT tractor during root crop work as it was too heavy during autumn harvesting and not sufficiently manoeuvrable at the headlands. In this situation the respondent preferred a semi-powershift transmission tractor, albeit due to its size and weight rather than the transmission specification. Analysis of sales literature confirms the minimum weight of an unballasted Case CVX1135 is 6390kg compared to 5250kg for a MXM130 semi-powershift tractor.

### 8.5.5 Specific Benefits of CVT Tractors

All respondents stated the primary benefit of a CVT tractor was the ability to infinitely vary the forward speed of the vehicle allowing it to be matched to the conditions. Speed matching was highlighted as particularly important in root crop planting and harvesting to maintain progress, whilst allowing the machine to cope with the often large volume of material. One respondent had previously needed to specify semi-powershift tractors with a creeper gearbox to reduce forward speed sufficiently for carrot harvesting. During the dry 2003 potato harvest there was little soil to cushion the crop as it passed over the machine, therefore a strategy of keeping the harvester bed full of potatoes reduced the harvesting damage. The ability to adjust forward speed and P.T.O speed independently made this possible. The variable forward speed made operating the CVT tractors with trailers far easier, particularly when filling whilst moving. The respondents all felt the tractor accelerated faster and, aided by lack of gearshifts, this meant the CVT tractors had quicker cycle times - highlighted by one respondent during muck spreading duties. All mentioned the reduced fatigue experienced by the CVT operator after a long shift undertaking road transportation, seen as an important factor in encouraging operators to work longer hours during busy periods.

Operating the engine at lower speeds in part-load situations was identified as an aid to reducing fuel consumption. This was even a strategy used by one respondent for ploughing; where their view was a substantial fuel saving could be made (50% on a similar sized semi-powershift tractor) by operating the engine between 1500 and 1700rpm whilst maintaining the same forward speed. In destoning work, a side-by-side comparison between two 110kW tractors highlighted the fuel savings possible to one respondent: the semi-powershift tractor used 170 litres of diesel during the day and the CVT tractor used 140 litres.

Whilst a stepped ratio transmission has limitations in maximising output or minimising fuel consumption (nearest gear ratio), by improving the control system, and through automatic shifting and cruise control, it would be possible to maintain as close as possible to maximum output, or if desired, minimum fuel consumption, thus allowing powershift-type tractors to compete closer with CVT tractors. As was discussed in Section 2.7, no clear evidence exists for the superior performance of the transmission itself, but the ability to vary engine and transmission speed and more importantly, the superior electronic control system, allow CVT tractors to outperform powershift-type tractors.

### 8.5.6 Detrimental Aspects of the CVT Tractors

In terms of design and operation there were only two criticisms of the CVT. During the wet 2002 autumn potato harvest one respondent had a number of incidents where CVT tractors with laden trailers were getting stuck. From operator reports, as the tractor began to lose traction, the controller reduced transmission speed and increased engine speed, resulting in the tractor getting stuck before the operator could override the control system. The following year had been particularly dry and the interview was undertaken prior to the 2004 harvest so it is unclear whether this occurred again. The user interfaces were criticised in different ways by the respondents. Some felt there were too many options of achieving a forward speed, whilst others would like to see increased functionality from the pillar mounted interface.

The other major complaint was not specifically concerned with the CVT, but three of the respondents felt the tractor brakes were insufficient for a tractor with a road speed of 50km/h.

### 8.5.7 Ease of Use of Different Tractor Transmissions

The common consensus from the respondents was that for an experienced tractor driver, and provided some initial training was given, CVTs were the easiest transmission type to operate. It was stated by two of the respondents that there was a difference between being able to drive a CVT tractor and driving it to its optimum efficiency. None of the respondents allowed casual or inexperienced staff to drive their CVT tractors. This highlights the need for any automated control system for powershift-type transmissions needs to be developed in such a way as to make its operation intuitive to the operator and require as little operator input as possible.

### 8.5.8 Improvements to Powershift-Type Transmissions to match CVTs

All respondents indicated that the major benefit of a CVT, i.e. the ability to operate engine and forward speed independently, could not be replicated with a multi-ratio transmission. Some suggestions with respect to improving the shift behaviour of a powershift-type transmission to smooth the ride were suggested. It was suggested that if forward speed could not be variable then consideration should be given to provision of a variable speed P.T.O.

### 8.5.9 Price Premium for a CVT tractor

With the exception of the large estate manager, the other respondents paid a price premium of between four and six thousand pounds above a similar sized tractor with a semi-powershift transmission. The estate manager paid a "very small" premium, but this can be attributed to their purchasing power, having bought an average of three tractors per annum for the last three years (see Figure 8-3). All were adamant that the tractors provided value for money in terms of increased output and operator comfort. The initial extra outlay was also not seen as a significant issue as the tractors had a known higher resale value than a comparable powershift-type tractor. Where a CVT tractor had been traded in by a respondent the additional price gained was the same as the premium originally paid. All respondents stated they were willing to pay a higher premium if necessary for a CVT tractor.

The continuing decline in farm incomes in the UK, and more generally in Western Europe will lead to a greater need to reduce costs. Whilst those already using CVT tractors will continue to operate CVTs, the significant price premium is likely to restrict the number of farmers switching to CVTs

### 8.5.10 Next Tractor Choice

It is clear the advantages of the CVT tractors had been appreciated by all the interviewees. Those involved in large amounts of root crop work were intending to operate all Case CVT tractors as their older tractors are replaced. The other respondents intended to either just run one CVT tractor (family farm), or to add a second CVT to allow the two full time employees to operate one each (grassland contractor). In both cases the additional cost and weight of the CVT tractor was seen as a prohibitive factor. Interestingly, the respondents felt the range did not extend across the tractor power spectrum far enough. Those involved in large areas of cereal cultivations would like high-horsepower CVTs available to replace full powershift draught tractors, indeed past choices had been limited through no CVT model. The smaller family farm felt their next tractor would be a semi-powershift tractor as the current Case CVT tractor range was too large and heavy for some operations on their farm, but they did intend to retain one CVT tractor.

## 8.6 *Customer Satisfaction Survey*

### 8.6.1 Theory and Methodology

The CNH Marketing team recently undertook an extensive survey into tractor buyer preferences and influences across Europe based on 600 farmer responses across a range of business scenarios and operation of a variety of tractor brands. In addition to basic information to segment the customer, the questionnaire was split into two main sections. The first section required the respondent to score the importance of 30 questions relating to product features (see Appendix A6.5) between 5 for "extremely important" and 1 for "not at all important". Recipients were also asked to rank the three most important features from the 30 listed, these ranked scores were then processed to give a reflected sum of ranks (RSOR) for each feature.

The second part of the questionnaire was a Kano survey described in detail by Berger *et al* (1993). Kano questioned the traditional theory that customer satisfaction was proportional to how functional a feature is, i.e. the more something is present the more satisfied the customer is and the more they would be prepared to pay. Kano described these as "Linear" or "One-dimensional" requirements, but also identified that there are two other elements to customer satisfaction: "Must-have" and "Exciter" attributes. The Must-have attributes are required in order for the product to meet the basic needs of the customer, however increasing this attribute will not improve customer satisfaction. Exciter attributes are those which give customers great satisfaction and so they are therefore willing to pay a price premium. However, satisfaction will not decrease (below neutral) if the product lacks the feature. These features are often unexpected by customers and they can be difficult to establish as needs up front. The relationship between the factors and satisfaction suggested by the Kano model is shown in Figure 8-4.

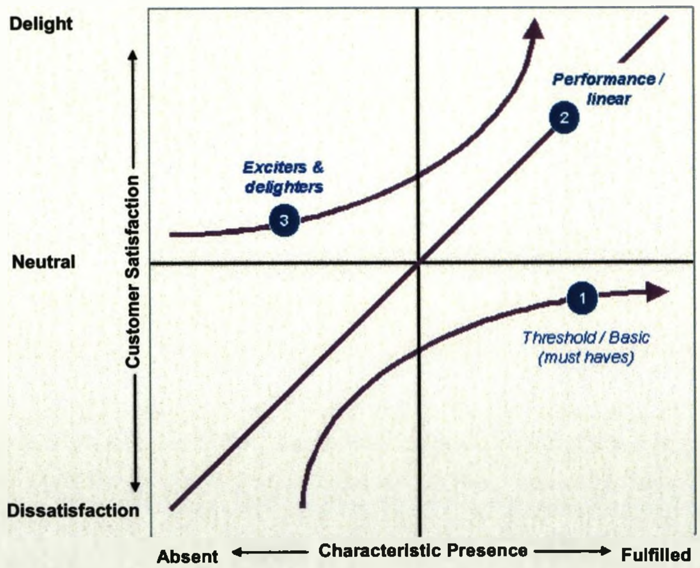



*Figure 8-4 – Kano model of customer satisfaction*

The same 30 questions from the initial part were asked for the Kano survey, this time in two ways: the first relating to whether the feature was present to a greater extent in the future and the second relating to whether the feature was present in a lesser extent in the future. Respondents marked a response ranging from "I'd be delighted to find it that way" to "It must not be that way" for each half of the question. Using a standard methodology described by Mello (2001) two scores (one for the functional and another for the dysfunctional half) were calculated for each question, based on all responses or on a segment by segment basis. These pairs were plotted onto the Kano chart to determine the customer response to that particular feature. Whilst the majority of questions were not relevant to this study, three questions were related to powertrain issues and so are included:

Q4. smoothness of transmission gear ratio changes;

Q10. maximum range of vehicle speeds;

Q27. maximised fuel efficiency.

### 8.6.2 Reflected Sum of Ranks

Figure 8-5 shows the top 15 results from the RSOR calculations from all respondents with the three questions of interest highlighted. It is clear the most important consideration identified was that the tractor was reliable, and then if there is an issue the operator can identify it quickly. Smoothness of changing transmission gear ratio was identified as the third most important factor of the 30. Fuel efficiency was ranked as the 6th most important factor, and one which it is suspected continues to rise with time as fuel prices become more of an issue to farmers.

The real surprise of the survey was the low RSOR for question 10, relating to the operator having the maximum range of vehicle speeds. This low RSOR could have resulted from the wording of the question The respondent was asked "how important is it if you have the maximum range of your vehicle speeds". It is unclear as to whether this relates to the overall range of speeds possible, i.e. maximum and minimum speeds, or whether it relates to the individual speed possibilities within the operating envelope (as would be desirable to know!).

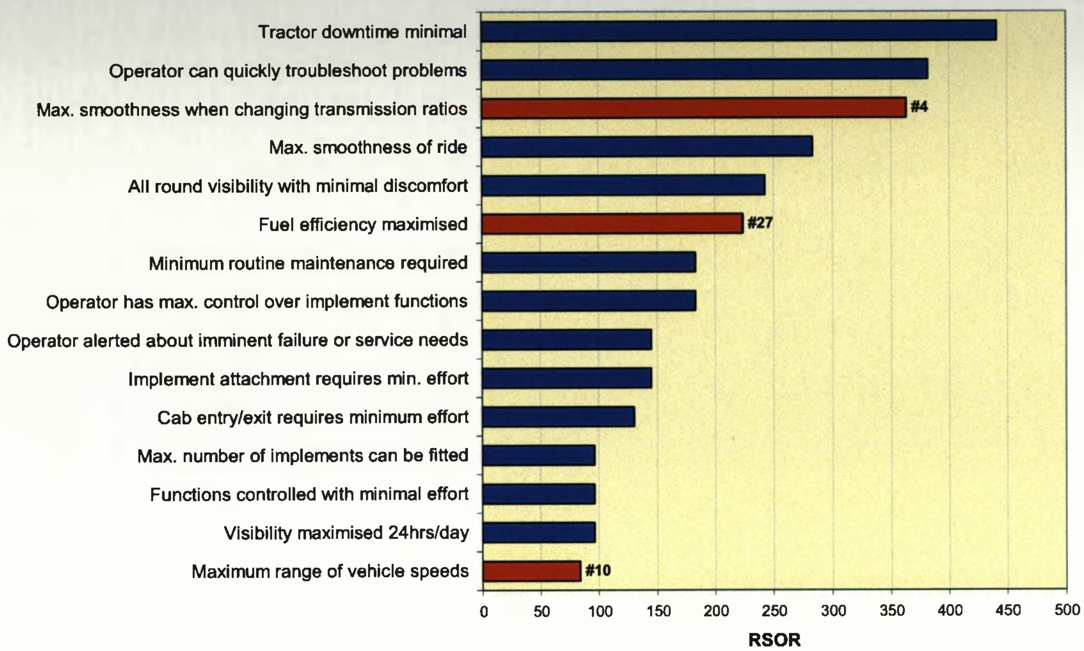



*Figure 8-5 – The top 15 reflected sum of rank scores*

### 8.6.3 Kano Survey

The Kano diagrams for the three powertrain related questions are shown in Figures 8-6, 8-7 and 8-8, with each diagram showing the response segmented by cropping type (see Table 8.4).

*Table 8.4 – Kano segmentation constituents*

| Segment | Constituent Parts |
|---|---|
| Cereal Crops | Wheat / Barley / Canola (Oilseed Rape) |
| Hay, Forage | All grassland related applications |
| Row/Root Crops | Root Crops / Maize / Soya / Cotton / Sugar Cane |
| Speciality Crops | Fruit / Flowers / Non-root Vegetables |
| Other | Forestry / Industrial / Construction |

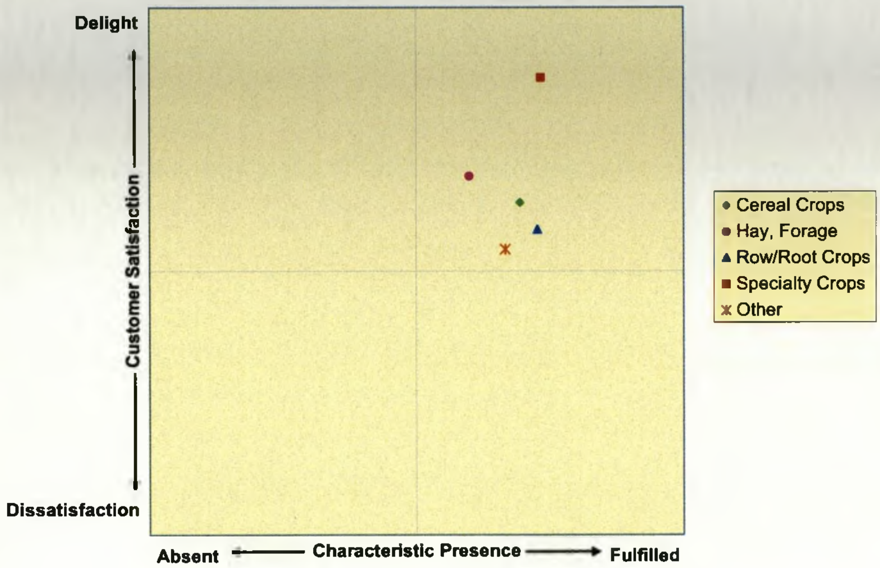



*Figure 8-6 – Kano analysis for Q4 – importance of gearshift smoothness*

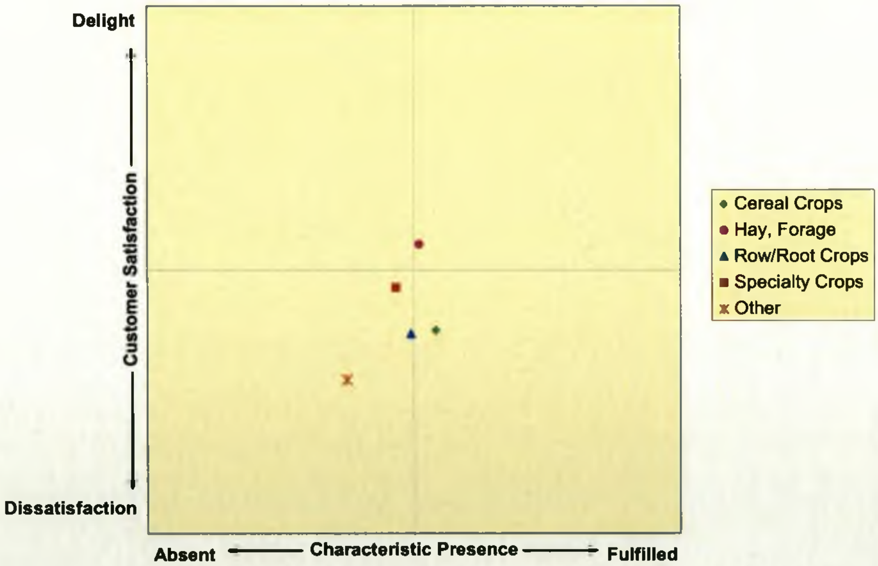



*Figure 8-7 – Kano analysis for Q10 – importance of maximum range of vehicle speeds*

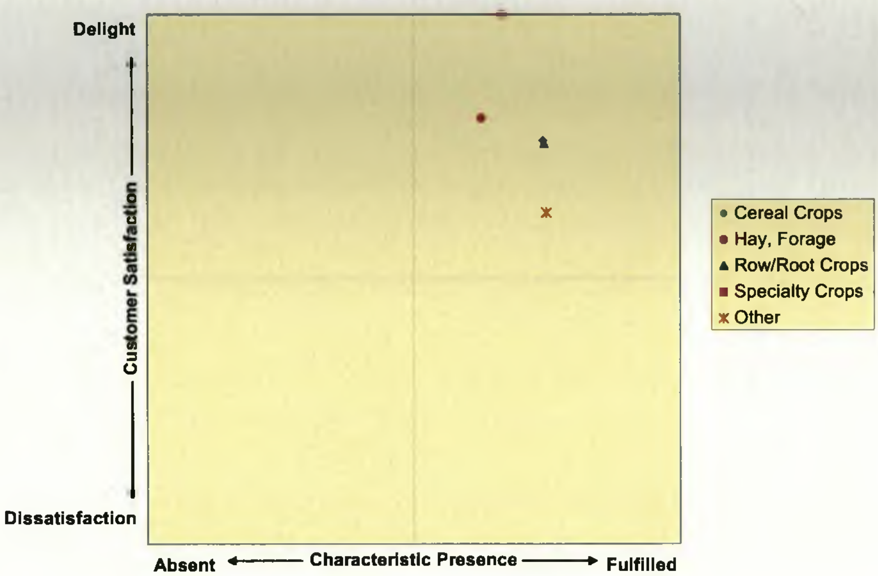



*Figure 8-8 – Kano analysis for Q27 - importance of maximised fuel economy*

The importance of gearshift smoothness (see Figure 8-6) was found to be a one-dimensional factor in determining satisfaction in all segments, implying respondents would be more satisfied with improved smoothness of shifting. The speciality croppers scored its importance higher than the other segments implying this was a more important characteristic to these respondents. These results show that smoothness of gearshifts is not considered a must-have requirement –customers are used to the traditional trade-off between smoothness and ease of gearshift.

The low RSOR previously identified for maximising the range of vehicle speeds available was backed up by the Kano results (see Figure 8-7). Here the segments are clustered around the middle of the diagram making it difficult to distinguish between the characteristic traits. The "other" users can be considered indifferent to maximising the range of gears available, reflecting the basic types of tasks carried out by these users. The cereal and row/root crop users tended towards considering a maximised range of speeds to be a "must-have" quality of a tractor rather than a one dimensional factor. This generally indifferent response is potentially explained by the wording of the questions which were similar to the ranking question described in Section 8.6.2.

The potential to maximise fuel economy (see Figure 8-8) was seen as an important satisfaction factor, with all sectors scoring this question highly in the upper right quadrant. The cereal and row/root crop sector respondents' satisfaction was highly linear in relation to fuel consumption. This means operators would be more satisfied by better fuel economy and reflects the higher usage these tractors would have and so the owners' focus would be on reducing machine operating costs. The grassland and specialist sectors considered maximised fuel consumption to tend towards an exciter factor, reflecting the low expectation on fuel economy as a result of generally lower operating hours in these sectors. This result backs up the responses given during the user interviews regarding the importance of fuel consumption.

The results identified through this survey show that the comfort of the operator, especially during gearshifting, and efficient operation of the tractor to reduce fuel consumption are important factors to the operator and future developments should be aimed towards ensuring CNH products slightly exceed the best in the class. No satisfactory conclusions can be drawn with respect to the need to maximise the range of vehicle speeds available to the operator.

## 8.7 Manufacturing Issues

Manufacturing costs are commercially sensitive and were not available for this part of the investigation. Therefore, suggestions as to the likely manufacturing implications from transmission choice have been identified.

The semi-powershift 16x16 transmission used in the test tractor was assembled in the CNH factory in Modena, Italy from components either bought-in complete or machined in-house in Modena or at a CNH plant in Anwerp, Belgium, where gears, shafts and transmission casings are machined. The assembled transmission is then shipped to a tractor plant, such as Basildon, UK, for tractor construction. As with any manufacturing situation, the mix between made and bought components is established by a variety of factors, primarily the total cost to produce each part against the supplier price. The typical shape of this curve with increasing production volume is shown in Figure 8-9.

For low production volumes it often makes economic sense to purchase in rather than manufacture, mainly due to the high fixed costs which can only be distributed across a small volume of parts. As manufacturing volumes increase, the fixed costs are spread and the economic benefit shifts towards manufacturing. The actual volumes associated with these trends are dependent on a variety of factors related to the complexity of both the component and the manufacturing process. There may be other considerations to be made when deciding on the best mix of make-or-buy components, such as the availability of machine capacity and labour to make the component and the potential supplier location and reliability.

These make-or-buy decisions can be extended to consider whole assemblies such as the tractor transmission. Whilst it makes economic sense to manufacture and assemble the transmission in the test tractor due to the volume required for worldwide production for the TSA range (approximately 15,000 units per annum), the specialist nature and currently limited market for CVT tractors, with a total volume less than 10% of TSA sales, means at present it makes economic sense to outsource the transmission assembly, in this case from ZF. The control software and associated control hardware responsibilities are retained by CNH and are unique to their tractors, but the hardware can be, and is, used by competitors.

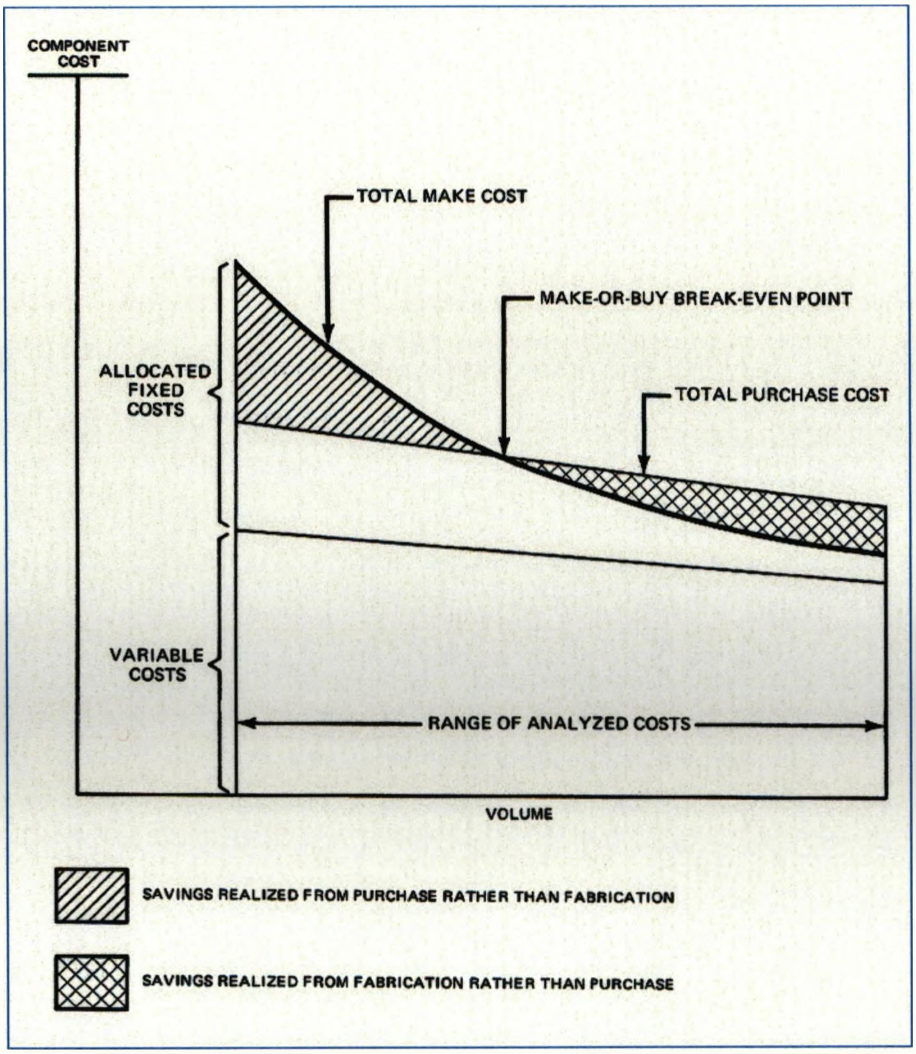



*Figure 8-9 – Hypothetical make-or-buy cost curve (MacDonald et al, 1969)*

Purchasing the CVT results in a higher total tractor cost of production compared to a standard powershift-type transmission. Therefore the finished product is more expensive. By purchasing the CVT, the high development costs associated with new transmissions, particularly ones different to conventional transmission technology are avoided and using an established product from a supplier with a good reputation enhances the product desirability and reliability. Whilst buying-in is acceptable for present production volumes, if the CVT route is to be developed into the main transmission option it would be desirable for CNH to bring the assembly and some manufacture in-house. If this was the case, it would be necessary to pursue mass adoption of CVT technology across the product range to justify the huge capital investment required. New transmission designs are inherently expensive and so the manufacturer must ensure volume and life are extended to the maximum possible to justify the investment. This can be seen in many examples in the tractor industry, for instance the TSA 16x16 transmission has had the same basic form for nearly 15 years since first introduced in Ford 40 series tractors. Massey Ferguson resorted to a joint venture with Renault to ensure sufficient volume for the "dynashift" transmission featured in, amongst others, the current MF 6400 series and the Renault Aries, but has been in existence since the mid 1980s.

Comparisons of the average cost to purchase a powershift-type and a CVT for tractors between 75 and 110kW were obtained from ZF (Stobinski, 2004). They would supply a powershift type transmission for €5,900 and a CVT for €7,200, the actual price depending on exact customer requirements. This still does not give a definitive answer as to which is the more expensive to produce as the ZF pricing policy could reflect the, presumably, higher research and development costs associated with the CVT transmission.

## 8.8 Conclusions

The recent advent of CVTs in agricultural tractors as an alternative to powershift-type transmissions raised the question as to the likely future direction of driveline technology. This part of the investigation focused on the key issues likely to influence this direction. The following conclusions can be made:

- there are obvious technological advantages using a CVT, in terms of variable forward speed, smoother speed changes and potential fuel savings through optimum loading of the engine;
- these advantages must be balanced with the additional purchase price, weight implications and whether the CVT benefits are required in a given situation;
- electronic control of powershift-type transmissions, whilst unlikely to allow continual variation in forward speed, could provide increased refinement to gearshift control and smoothness as desired by tractor operators;
- whilst political and economic issues will influence the future demand for, and size of, agricultural tractors; it is unlikely to influence either transmission choice exclusively;
- manufacturing economies of scale could help to reduce the price premium for a CVT and therefore lead to an increase in their share of the overall tractor market in developed European countries;
- it is unlikely either one of the two transmission systems will completely dominate in the future.

# 9 Control Strategy Improvements

## 9.1 Introduction

The primary focus of this investigation was to quantify the characteristics of the test tractor powertrain in response to a variety of operational scenarios. A number of key areas have been highlighted where potential improvements could be made to the vehicle performance and behaviour, through the use of intelligent control strategies, these made possible by the ability to communicate between vehicle sub-systems through the use of CAN-bus messages.

It is necessary to enhance control strategies, not only to improve the operational characteristics of the vehicle but also, as identified in Section 8 to allow semi-powershift tractors to compete in the future with CVT equipped vehicles.

In addition to the areas for improvement identified from the fieldwork and axle dynamometer investigations, the CVT user interviews (see Section 8.5) provided details of key CVT operational features which would be desirable to replicate on other (stepped-ratio) vehicles.

The majority of the potential control strategy improvements fall within three main categories, namely engine power boost, gearshifts and vehicle speed control. This part of the investigation discusses each of these categories and highlights potential methods to enhance vehicle operation.

## 9.2 *Engine Power Boost*

### 9.2.1 Theoretical Power Boost Operation

Engine power boost is an extremely useful feature of the test tractor allowing the engine to generate additional torque during P.T.O. operations to achieve an increased output. As was demonstrated during the change in power harrow working depth (see Section 5.6), as the working depth increased, the resulting additional P.T.O. torque requirement caused the vehicle controller to increase the allowable engine torque output up to the boosted full-load curve, thereby allowing engine speed and tractor forward speed to return to pre-depth increase levels.

Power boost operation is determined by two separate control algorithms. The first determines the boost status ($B_S$), i.e. whether the engine controller can switch to boosted mode, based on the torque demand from the P.T.O. driveline. The power boost logic used to make this decision is presented in Figure 9-1.

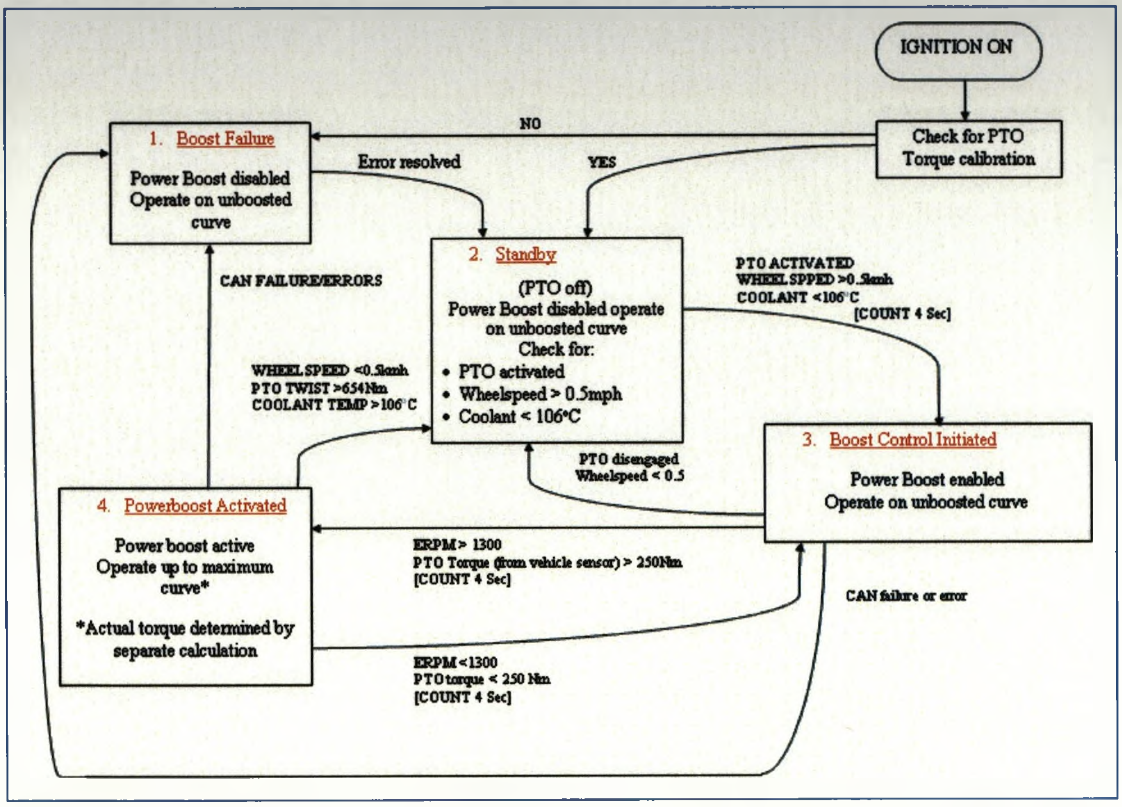



*Figure 9-1 – Power boost status logic diagram*

The second algorithm determines the actual increase from the unboosted torque curve, up to the maximum boosted curve, based on the division of engine power between the P.T.O. driveline and the traction driveline. The overriding criterion used in this calculation is that the P.T.O. driveline must not receive more than the maximum rated power of the unboosted engine. The allowable level of boost is therefore determined by calculating the approximate transmission power according to Equation 9-1.

$$\textit{Transmission Power} \approx \textit{Engine Power} - \textit{P.T.O. Power} \qquad \textit{Equation 9-1}$$

Engine power is determined from the flywheel torque sensor (see Section 3.3.1). P.T.O. power is determined from the vehicle P.T.O. torque sensor (see Section 3.3.6). A lookup table (see Appendix A7) is then used to determine the permissible additional torque, as a percentage of the maximum engine torque ($B_{\%}$), to be added to the unboosted torque curve. The minimum of either this value or the full-load boosted curve is implemented for each engine speed.

To avoid rapidly changing the engine power output the rate of change of $B_{\%}$ is restricted to ±1% every 2.5 seconds. However, a ring buffer is used to re-introduce the same value of $B_{\%}$ when the power boost feature is re-enabled after an interruption, for instance after a headland turn.

### 9.2.2 Actual Power Boost Operation

During the power harrow field investigation (see Sections 5.5 and 5.6) it was discovered that in practical field operations, the behaviour of the power boost system did not follow the theoretical system design. The tractor was operating in boosted mode in a number of instances where P.T.O. torque was below the minimum threshold (see Appendix A4.2). In the majority of tractor-power harrow configurations investigated, the power consumed by the transmission driveline was sufficient to allow maximum power boost, provided the logic allowed boosted mode to be used.

The field tests also highlighted that the vehicle P.T.O. torque transducer indicated a consistently higher P.T.O. derived torque value than the external British Hovercraft transducer (see Appendix A4.2). A steady state investigation of the two transducers was undertaken with an eddy current dynamometer to provide a series of P.T.O. torque loads. The output of the two transducer systems was then compared (see Figure 9-2). As can be seen, the vehicle torque transducer does indicate higher P.T.O. torque, with a R.M.S. torque error of 23Nm when compared with the output from the external transducer. The calibration of the vehicle sensor was checked and the test repeated, with the same result.

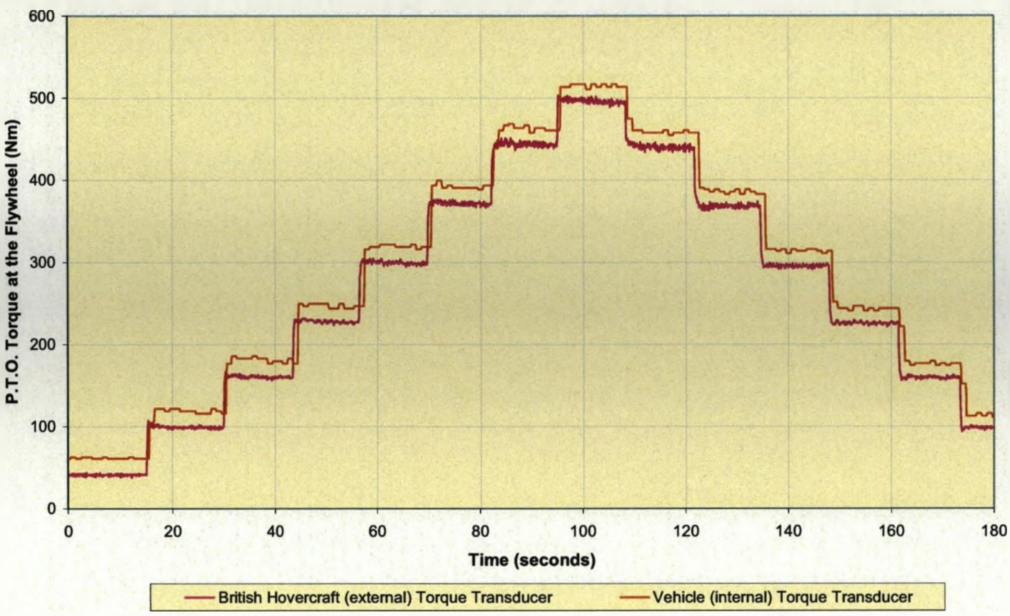



*Figure 9-2 – A comparison between the external torque transducer (British Hovercraft) and the vehicle internal torque transducer*

A number of dynamic boost investigations were then undertaken using the axle dynamometer in conjunction with a P.T.O. dynamometer (see Figure 9-3). An example is presented in Figures 9-4 and 9-5. The test allowed the torque division between the traction driveline and the P.T.O. driveline to be varied during the test, and therefore to vary the Boost % calculated and added to the engine unboosted torque curve.

*Figure 9-3 – Test tractor undertaking dynamic power boost investigation with axle and P.T.O. dynamometers*

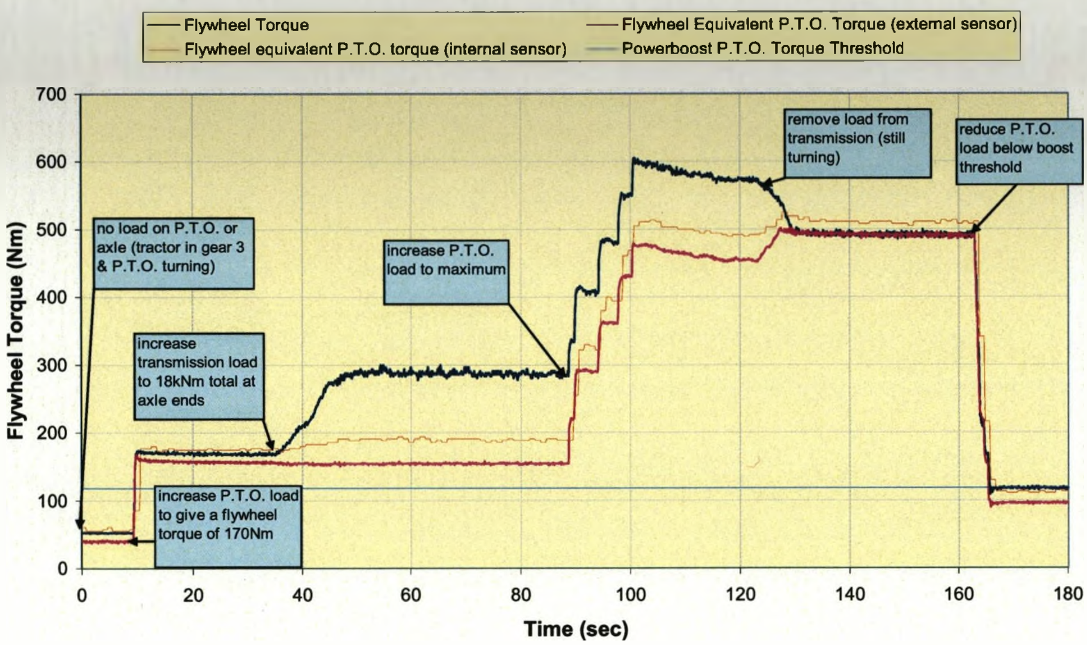



*Figure 9-4 – Dynamic power boost investigation - Torque loads*

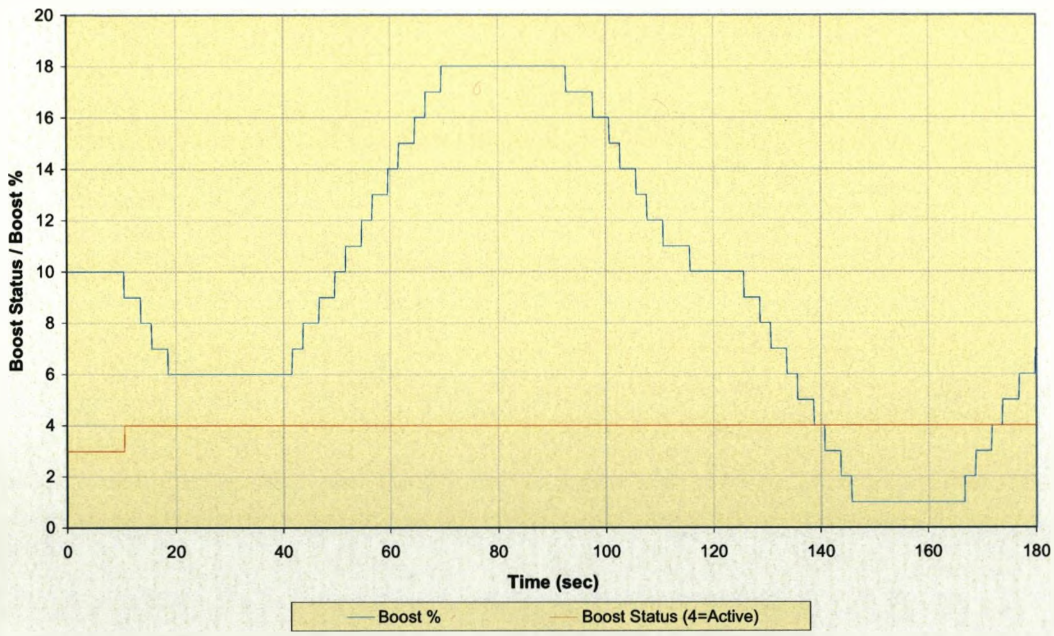



*Figure 9-5 – Dynamic power boost investigation – boost status and boost % (always calculated, regardless of boost status)*

From Figure 9-4 it can be seen that the P.T.O. torque load was increased at 10 seconds, crossing the threshold which permits power boost to occur. The tractor did switch modes, although almost at precisely the instant that the P.T.O. load was increased. According to the power boost status logic (see Figure 9-1) there should be a 4 second delay before the boost status changes. This therefore implies the tractor would have gone into boosted mode even if the P.T.O. torque had not increased above the threshold. This corresponds with vehicle behaviour encountered during the field experiments, where boosted mode was being activated with a far lower P.T.O. torque demand than the specified threshold. From Figure 9-5 it can be seen that even when power boost is disabled, the $B_{\%}$ calculation is still performed, although the value is not added to the unboosted torque curve.

During the next period (10-35 seconds) the tractor was operated with no torque applied to the axle ends; therefore the transmission power was minimal. This resulted in $B_{\%}$ reducing to 6%. A greater reduction was expected as 6% Boost relates to the transmission consuming 9kW (12hp).

The increase in transmission load (at t=35 seconds) increased the level of power transferred through the transmission and subsequently allowed the level of $B_{\%}$ to increase, which it duly did.

As the P.T.O. load was increased again (at t=90 seconds) the tractor $B_{\%}$ began to reduce again as the transmission power reduced, due to engine speed reduction as a result of P.T.O. torque load increasing. The removal of the axle torque load (at t=125 seconds) further reduced the value of $B_{\%}$. This time, unlike the period from 10 to 35 seconds, $B_{\%}$ did reduce to 1%. The reduction of the P.T.O. load (at t=160 seconds), and the subsequent increase in engine speed, resulted in $B_{\%}$ increasing again. This was despite the fact P.T.O. torque was below the minimum boost threshold, and there was no axle torque applied, implying no transmission power consumption.

### 9.2.3 Improvements to the Power Boost Control Feature

From experience and data gained during the power harrow field investigation it is questionable whether there is a need for graduated power boost levels, given that once there was sufficient P.T.O. torque demand to allow power boost to activate, there was generally a sufficient level of power consumption through the traction driveline to allow full power boost to be implemented. Non soil-engaging P.T.O. operations are likely to require lower levels of traction driveline power, resulting in a potential requirement to limit the additional $B_{\%}$. However, they are also less likely to require the additional power which can be made available, unlike power harrowing, which has a very high power requirement.

As previously discussed, during the road transport investigation (see Section 5.7) power boost would improve performance during haulage duties. Since this investigation was undertaken, the power boost feature has been extended by the manufacturers to include the upper range of gears (13-16). As the limitations of the driveline are the torque capacity of the differential crown wheel and pinion, and the axle epicyclic reduction units, it should be possible to extend the power boost feature further down the upper range of gears to further aid transportation, and particularly accelerating with a heavy payload. The torque transmitted will still be below the levels reached in the lower range of gears.

An alternative system to control the level of $B_{\%}$ is proposed. For P.T.O. applications, the minimum threshold could remain in place, but the over-complicated method of determining $B_{\%}$ could either be removed, or simplified to reduce the graduations. It is also proposed that the limitations imposed in changing the level of $B_{\%}$ be removed; the small fluctuations in engine speed likely to be experienced as a result will almost certainly be lower than the fluctuations experienced when undertaking a high-draught task such as ploughing. For non-P.T.O. applications, the full power boost could be made available for gears 13 to 16. In addition, an intermediate level of boost could be made available for gears 9 to 12. This would assist with acceleration performance during transport tasks and also assist during the ascent of steep on-road slopes, where it is necessary to operate below gear 13.

It may also be possible to extend the power boost feature to gears 5 to 8, but in a time/duty-cycle limited manner (such as the one suggested by Scarlett and Lowe, 1997), for example when required to overcome a difficult patch during ploughing. By using the tractor-implement model, it is possible to predict the likely improvement in engine and forward speed possible. A simulation was undertaken to show the recovery in forward speed possible by switching to the boosted full-load curve, the results of which are presented in Figures 9-6 and 9-7.

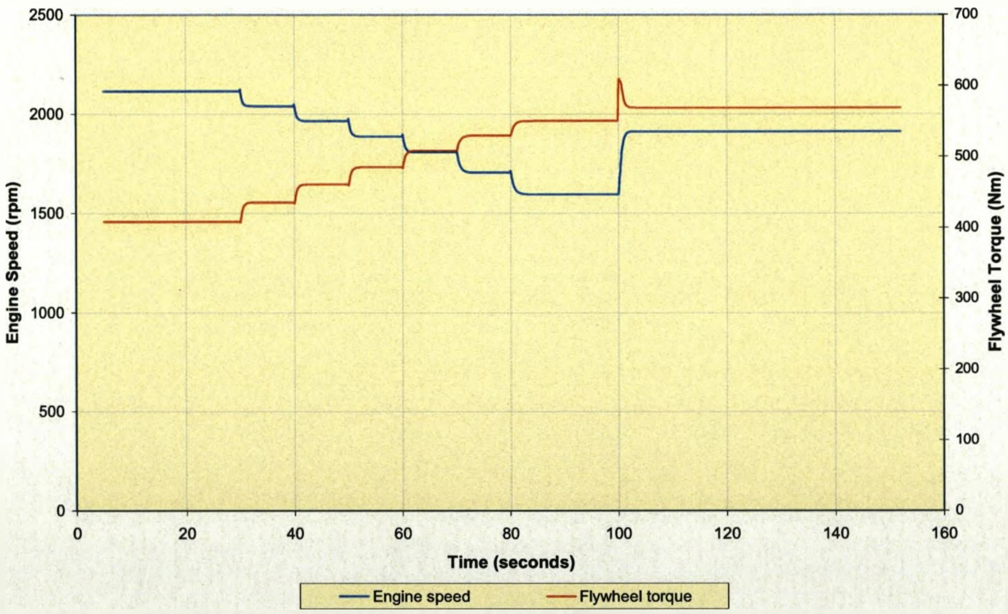



*Figure 9-6 – The effect of increasing soil resistance followed by a switch to the boosted full-load curve on engine torque and speed (simulation)*

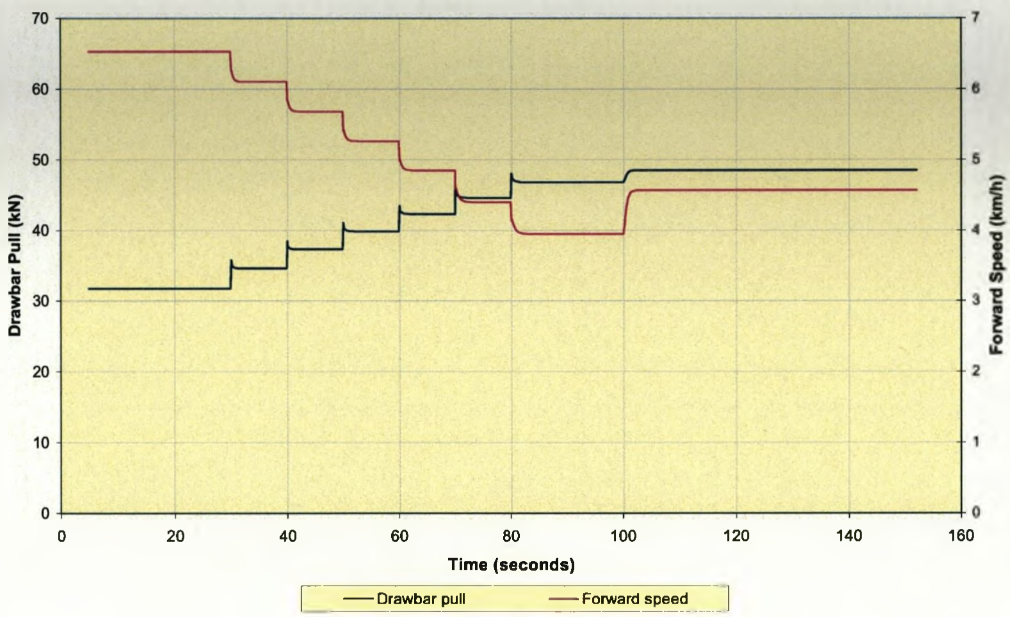



*Figure 9-7 – The effect of increasing soil resistance followed by the switch to the boosted full-load curve on forward speed and drawbar pull (simulation)*

The soil strength factor (Fi) was increased incrementally, causing the drawbar pull requirement to increase. Upon switching to boosted mode, the engine speed was able to increase from 1595rpm to 1912rpm. As a result, forward speed also increased from 3.95km/h to 4.57km/h. Theoretical workrate would increase by 15% from 0.8ha/hr to 0.92ha/hr.

It would be possible to implement these and other improvements to the power boost feature, but in order to do so confidence in accurate measurements of torque and power flow through the P.T.O. and traction drivelines must be gained. It is therefore proposed that an improved tractor P.T.O. torque transducer system is required.

## 9.3 Gearshifts

The smoothness of making changes to forward speeds was highlighted during the user interviews as one of the benefits of CVT-equipped tractors. The absence of step changes in gear ratios is the main reason for this smoothness of speed change. A semi-powershift transmission, by design, features stepped-ratios, making seamless changes in speed more difficult to achieve.

For the powershift gear changes in the test tractor transmission, the fluctuations in engine speed and torque during the change are directly related to the oncoming and offgoing clutch pressures. Figure 9-8 shows a 15-16 gearshift with the operator requesting the shift at t=0 seconds.

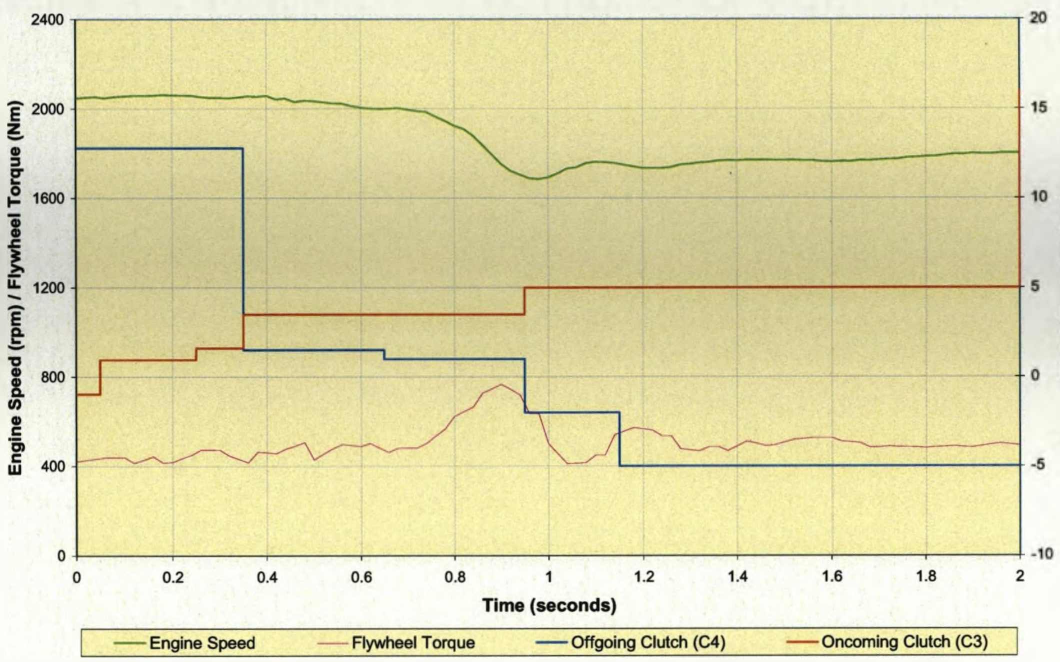



*Figure 9-8 – Engine speed and torque profiles with the clutch pressure profiles for a 15-16 gearshift*

As can be seen, the oncoming clutch pressure begins to increase almost immediately, whilst the offgoing clutch is still operated at full pressure. 0.35 seconds after the start of the shift, the offgoing clutch pressure is reduced to a lower pressure, at the same time the oncoming clutch pressure is increased. From this point the drive beings to be transferred from one transmission path to the other, resulting in engine speed reducing and flywheel torque increasing. The flywheel torque peaks just prior to the offgoing clutch pressure being completely removed (at t=0.95 seconds), after which the torque reduces and the engine speed starts to increase, as the vehicle accelerates. The time to change the pressures between the two clutches gives the overall gearshift time, but it also indicates how quickly the input and output shafts have to be synchronised. The faster this is done, the harsher the shift is, and the more noticeable it is to the operator. There is also a conflict between the obtaining a smooth shift and making the gearchange quickly to maintain vehicle momentum.

The gearshift system on the test tractor implements the shifts by means of time vs. pressure lookup tables, with adjustments made for oil temperature. Each gear change has its own offgoing and oncoming clutch pressure vs. time profile. The inherent problem is that this is essentially an open loop system; therefore the effects of variations in engine loading and the effective inertia of different tractor and implements are not accounted for. A shift which is smooth under heavily loaded conditions will be harsh in lightly loaded situations.

Flywheel torque and engine speed data could be used as the basis for a shift control strategy, whereby the clutch time vs. pressure profiles are adjusted to suit the load. The engagement rate of the clutches can be reduced in a lightly loaded situation, thereby allowing the shift to occur over a longer time period. This reduces the acceleration rates of the input and output shafts, making the shift smoother. Under heavy load, the priority is to make the shift quickly to avoid a loss of vehicle momentum; therefore a rapid engagement rate is required. The high load effectively damps the rate of acceleration, so the shift is not as noticeable as a lightly loaded shift. It would be desirable to make this a self-learning control system whereby the smoothness of each shift is measured (through measuring the rate of change in engine speed) and used to adjust that gearshift when it next occurs.

Additional improvements to the gearshift control strategy could also include the use of engine speed adjustments during gearshifts, particularly for the non-powershift change between gear 12 and 13. During this change there is no continual power-flow between the engine and the wheels (unlike the powershift changes): as the offgoing clutches are disengaged prior to the synchromesh change, the sudden reduction in engine load causes the engine to accelerate. After the synchromesh change is made, and the oncoming clutches are re-engaged, the engine speed is reduced again under load. This makes the 12-13 gearshift uncomfortable and very noticeable to the operator. As the request to make a gearshift is transmitted on the CAN-bus, the engine controller could momentarily reduce the engine speed as the clutches are disengaging, to prevent the engine accelerating during the gearshift. More noticeable was the speed fluctuation during a downshift (13-12), as was experienced during the deceleration experiments (see Section 5.8). During this downshift it would be necessary to increase engine speed, to allow the shaft speeds to be synchronised as the oncoming clutches pressurise.

There is also the potential to use the engine power boost feature to aid gearshifts in heavily-loaded situations, to speed the transition to the new gear; for example making an upshift whilst operating a large square baler. Flywheel torque and engine speed levels would be used to determine the level of boost allowable and the duration required.

The other potential improvement to transmission operation would the inclusion of a facility to automatically skip gearshifts in lightly-loaded situations. During the acceleration tests with the solo-tractor, it was possible to make upshifts as soon as the previous one had been made as a result of the light loads and closely matched ratios. An improvement would be when instances of light-load operation could be identified by the level of flywheel torque required to accelerate away from rest, the transmission control system could then change from 9$^{th}$ gear straight to 11$^{th}$ or even 12$^{th}$. This would not only save time, but also serves to reduce operator fatigue, by reducing the need to make excessive, unnecessary gearshifts.

## *9.4 Vehicle Speed Control*

The ability of CVT equipped tractors to be driven using the foot throttle as a forward speed control pedal, is a desirable feature to the operators. In addition, it is one which could be implemented on a tractor equipped with a stepped-ratio powershift transmission, by use of the 'autoshift' feature. In an operator-selected mode, foot throttle position would determine the desired vehicle forward speed: engine speed and transmission ratio could be adjusted automatically as required. Flywheel torque and engine speed transducers would be used to determine the vehicle load. The rate of change of pedal position would be used to indicate the desired rate of acceleration. The vehicle load and pedal operation would form the basis of the control strategy in determining how to adjust engine speed and transmission gear ratio to achieve the desired forward speed. Whilst this system would not be as seamless as on a CVT tractor, it would be entirely possible to implement.

# 10 Conclusions & Recommendations

## *10.1 Conclusions*

This research investigation has extended the knowledge and understanding of the test tractor dynamic performance and has permitted the development of improved control strategies. In particular the following conclusions can be made:

1. A mathematical model to predict the dynamic behaviour, based on Newtonian physics has been developed to represent the test tractor powertrain, and its sub-systems. When validated against measured field data the model estimated engine speed to within 1% (20 rpm) and flywheel torque to within 5-10% (30 to 60 Nm) depending on the application.

2. From the above model, and with data obtained during this investigation, it was possible to investigate potential improvements to powertrain control strategies. Improvements to engine power management, gear shifting and automated driving strategies were identified and solutions proposed to enhance vehicle performance and behaviour. These include a revised powerboost strategy with reduced complexity and an extended operating range, combined engine-transmission closed loop gearshift control and automated skip-shifting based on flywheel torque loads.

3. A procedure to accurately quantify the losses through the tractor driveline across the full vehicle operating range has been developed. An axle dynamometer was used to load the driveline at the axle ends and by using the flywheel torque transducer to measure the torque produced by the engine, the losses through the driveline were determined and optimum operating gears and engine speed regions were identified, for example gear 6 was much more efficient than gear 5 or 7.

4. From the above, a multiple regression model was developed to determine driveline torque losses as a function of equivalent flywheel torque, transmission output speed and the number of gear pairs transmitting torque. This model accounted for 99.8% of the experimental variation with a standard error of prediction of 6.77 Nm. Peak efficiency achieved with this transmission was 96.7%.

5. A novel technique has been developed to determine equivalent driveline inertia in each gear through a limited number of measurements and use of matrix algebra. It was found that 50% of the inertia can be attributed to the tractor rear wheels, causing a high torque requirement during acceleration in the upper gear range.

6. Investigations into powertrain loading with this tractor-implement configuration have determined that:
   a) Soil type, forward speed and working depth had a significant impact on flywheel torque levels during cultivation tasks. Increasing the plough working width did not increase flywheel torque proportionally.
   b) The power distribution between traction and P.T.O. drivelines when power harrowing was influenced by forward speed and working depth.

7. Powertrain loads when undertaking transportation were directly influenced by terrain. The gradients encountered (<10%) were found to be sufficient to require non-powershift gear changes which severely affected momentum.

8. Gearshift characteristics and shift quality were influenced by the number of clutches being engaged and disengaged as well as the type of operation being undertaken. High draught forces resulted in severe torque spikes during gearshifts whereas P.T.O. loading did not.

9. Powershift-type transmissions are likely to remain popular in the foreseeable future due to their price advantage against CVTs (10-15%) and because no conclusive data exists to prove the claimed improvements in performance and fuel economy for continuously variable transmissions.

10. It was possible to utilise the standard vehicle sensors and CAN bus on an electronically controlled tractor to provide information on powertrain loading characteristics once a suitable calibration algorithm was implemented. The issue of sensor accuracy was identified with respect to the P.T.O. torque transducer, and consideration must be given to an improved measuring device if it is to be used as control parameters.

11. A technique was developed, using an axle dynamometer, to replicate in a controlled manner measured field data of powertrain loading in real time to within 5% of the setpoint flywheel torque.

12. Utilising the model and axle dynamometer would allow the validation of proposed improvements to be made during the controller development phase, rather than during vehicle field testing.

## 10.2 Recommendations

1. Further development of the mathematical model is required; particularly the addition of a sub-model to determine the behaviour of the powertrain during powershift gear changes, by modelling the clutch engagement and disengagement behaviour.

2. The development of a CVT model, for substitution into the overall tractor-implement model, would allow detailed comparisons of the two driveline systems to be undertaken.

3. Additional development and evaluation of the control strategy improvements is required to prove the suggested concepts proposed.

4. Following further development, incorporate model-referenced adaptive control software on a test vehicle, using inputs to the model from the tractor-implement sensors and using the model to control the vehicle sub-systems.

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# APPENDICIES

# A1 Test Equipment Specifications

## *A1.1 Tractor Specification*

Shown below is key technical data on TS135A model, taken from the technical specifications. It should be noted that ISO TR14396 is measured at the engine crankshaft without the fan or other power absorbing accessories (International Standards Organisation, 1996).

| | |
|---|---|
| **Engine (Measured to ISO TR14396)** | |
| Rated Engine Power | 100kW (134hp) |
| Maximum Engine Power | 106kW (142hp) |
| Rated Boosted Power | 119kW (160hp) |
| Maximum Boosted Power | 124kW (166hp) |
| | |
| Rated Engine Speed | 2200rpm |
| Maximum Torque (at 1400rpm) | 612Nm |
| Maximum Boosted Torque (at 1600rpm) | 654Nm |
| Torque Rise | 41% |
| | |
| Idle Speed | 650rpm |
| Number of Cylinders | 6 |
| Aspiration | Turbocharged with Intercooler |
| Emission Level | Tier II |
| | |
| **Transmission** | |
| Transmission Variant | Electro Command™ |
| Number of Gears (FxR) | 16x16 |
| Number of Powershift Gears | 4 |
| Top Speed | 40km/h |
| P.T.O. Ratio ($r_{pe}$) | 2.12:1 |

**Hydraulic System**

| | |
|---|---|
| Type | Closed Centre, load sensing |
| Main pump flow | 113 litres/minute |
| Main pump pressure | 210 bar |
| Draught Control | Electronic |
| Max lift at ball ends with arms horizontal | 7864kg |
| Max lift through full range | 6580kg |

**General Information**

| | |
|---|---|
| Overall length to link ends (inc weights) | 4532mm |
| Wheelbase (std axle) | 2650mm |
| Minimum Width | 1913mm |
| Height to top of cab | 2920mm |
| Ground Clearance | 478mm |
| Tractor mass (std axle, unballasted) | 4950kg |
| Tractor mass (as used for experiments) | 6752kg* |
| Cab Suspension | Yes |
| Front Axle Suspension | No |
| Cab Noise (OECD) | 72dBa |
| Tyre Sizes (Front) | 480/65 R28 |
| (Rear) | 600/65 R38 |

* 5752kg for Power Harrowing

## *A1.2 Plough Specification*

| | |
|---|---|
| Make | Dowdeswell |
| Model | 100 Series Delta Furrow HA |
| Operated Mode | Semi-mounted |
| No of Furrows | 5 (3+1+1) |
| Leg Protection | Shearbolt |
| Furrow Width | 305 – 457mm (12 – 18”) |
| Adjustment Method | Hydraulic |

| | |
|---|---|
| Underbeam Clearance | 660mm |
| Interbody Clearance | 910mm |
| Body type | UCN |
| Skim type | K |
| Depth wheel | pneumatic tyre (rear of beam) |
| Plough Mass | 1524kg |

## A1.3 Power Harrow Specification

| | |
|---|---|
| Make | Dowdeswell |
| Model | PH 400 |
| Working width | 4.0m |
| Overall width | 4.2m |
| P.T.O. speed | 1000rpm |
| No. of Rotors | 16 |
| Theoretical Rotor Speed (at 1000 P.T.O. rpm) | 335rpm |
| Roller type | Packer |
| Adjustment method | Hydraulic |
| Power Harrow mass | 1855kg |

## A1.4 Trailer Specification

Marston 'Ace' 10 tonne tandem axle monocoque grain trailer, fitted with hydraulically-actuated brakes and multi leaf semi-elliptic springs on axles. Fixed drawbar. Recorded load distribution (tractor and trailer attached):

| | | |
|---|---|---|
| Tractor | | 6,752kg |
| Trailer empty | | 2,600kg |
| Trailer laden | | 12,412kg |
| Distributions: | Front | 2450kg |
| | Rear | 6251kg |
| | Trailer | 10463kg |
| Gross train mass | | 19,164kg |

# A2 Additional Model Data

## A2.1 Model Input Data (M-file text)

```
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%                                    %%%
%%%    ENGINEERING DOCTORATE MODEL     %%%
%%%                                    %%%
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%

% Global Variables for use in EngD model
% Last Updated : 14th May 2005
% Created by David Sayer
% Model and Data based on New Holland TSA135 tractor
% Electronic engine and 16x16 transmission (40km/h)
% Tyres    Rear:    600/65R38
%          Front:   480/65R28


% Open the Simulink Master Model file:
engd3
% Set initial engine speed (rpm):
IES = [700];


      %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
      %   Vehicle General Properties    %
      %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% Tractor mass (kg):
mass_tr = [6752];
% Engine inertia, 6cyl cr inc flywheel (kg.m^2):
IE = [0.8187];
% Total Inertia of the driveline for each gear, referenced to
% the flywheel, including the rear wheel inertia.  Used in
% conjunction with Gear number (G), for gears 0 to 16 (kg.m^2):
IDF = [0;0;0;0;0;0;0;0;0;0.12;0.16;0.26;0.37;0.55;0.82;1.23;1.83];
% Loaded rear wheel radius (m):
wh_rad = [0.82];


      %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
      %   Additional Driveline Information   %
      %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% Number of meshed gear pairs (for use in the driveline loss model):
Gm = [0;5;3;7;5;3;1;5;3;3;3;5;5;1;1;3;3];
% Basic Transmission Ratios for the 16 gears (and 0)
% Note: Reverse ratios are not included:
G = [0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16];
% Transmission gear ratios in column for direct lookup function:
rt = [0;8.514;6.941;5.700;4.647;3.624;2.955;2.426;1.978;2.177;1.775;
     1.458;1.188;0.927;0.756;0.621;0.506];
% PTO reduction ratio:
PTO_ratio = [2.12];
% Rear axle ratio (product of differential and hub reduction):
rar = [6.75*47/9];
```

```
        %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
        %   FOOT THROTTLE TO SET POINT SPEED   %
        %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% Used to determine the relationship between
% percent foot throttle and set point speed:
% esps = set point speed (rpm-foot throttle)
% fth_per = foot throttle percent
esps = [700 700 700 710 720 735 750 765 790 820 860 920 980 1080 ...
        1180 1280 1390 1530 1680 1830 1980 2150 2270 2350 2400];
fth_per = [0 3 6 9 12 15 18 21 24 27 30 35 40 45 50 55 60 65 70 ...
          75 80 85 90 95 100];


        %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
        %   HAND THROTTLE TO SET POINT SPEED   %
        %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% Used to determine the (linear) relationship between
% percent hand throttle and set point speed:
% spd_sp_hth = set point speed (rpm-hand throttle)
% hth_per = hand throttle percent
spd_sp_hth = [700 2400];
hth_per = [0 100];


        %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
        %   GOVERNOR DROOP RELATIONSHIP   %
        %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% Governor feedback parameters (rpm per mg/stroke)
% relating to 0,1,2 modes (0, 5 and 12% droop)
% 0 = Isochronous governor (0%)
% 1 = 5% governor
% 2 = 12% governor
fbf = [0;1.49999390;3.2000732];


        %%%%%%%%%%%%%%%%%%%
        %   CONVERSIONS   %
        %%%%%%%%%%%%%%%%%%%
% General conversions used in the model
% rpm_rad = converts revolutions per minute to radians per second:
rpm_rad = [2*pi/60];
% rad_rpm = converts radians per second to revolutions per minute:
rad_rpm = [60/(2*pi)];


        %%%%%%%%%%%%%%%%%%%%%%%%%%%%%
        %  No Load Fuel Quantities  %
        %%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% Curves used to determine the quantity of fuel required to achieve
% a given no-load engine speed with no engine load from a bare engine
nl_rpm = [700,800,900,1000,1100,1200,1300,1400,1500,1600,1700, ...
         1800,1900,2000,2100,2200,2300,2400];
nl_fuelling = [5.36,5.67,5.98,6.29,6.59,6.88,7.18,7.47,7.76,8.06, ...
              8.71,9.32,9.85,10.38,10.92,11.46,12,12.7];
```

```
        %%%%%%%%%%%%%%%%%%%%%%%%
        %  Fuel Limit Curves  %
        %%%%%%%%%%%%%%%%%%%%%%%%
%  Curves are used to limit the maximum fuel on each engine cycle
%  Compared to the PI governor output, minimum value chosen
%  Modelled as engine speed (rpm) v's fuelling (mg/stroke)
%  Data from CNH bare engine Dyno tests
%  UNBOOSTED DATA:
ub_rpm = [1001,1101,1200,1302,1400,1500,1600,1700,1800,1900,2000, ...
          2100,2200,2246,2275,2292,2310,2331,2346,2362,2369];
ub_fuelling = [60.7,63.8,67.5,69.9,71.8,71.7,70.5,68.5,66.4,64.3, ...
               61.9,58.7,55.8,52.6,47.0,41.5,36.1,31.3,26.3,20.7,18.1];
%  BOOSTED DATA:
b_rpm = [1001,1101,1200,1302,1398,1499,1601,1800,1997,2095,2196, ...
         2216,2238,2256,2274,2290,2308,2329,2346,2361,2372];
b_fuelling = [60.7,63.8,67.5,69.9,72.7,74.7,78.5,78.6,74.7,72.0, ...
              68.3,64.7,58.6,52.9,47.2,41.8,36.4,31.4,26.1,21.0,16.7];


        %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
        %  Engine Torque Production  %
        %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%  Uses the engine speed (rpm) & fuelling (mg/stroke)
%  to give torque output (Nm) - from CNH bare engine fuel loop data
%  Each point is an average of three readings
%  Have loop curves for approx 1000, 1600, 1800 and 2200 rpm
erpm_et = [996,1596,1795,2195];
fuelling_et = [5,10,15,20,25,30,35,40,45,50,55,60,65,70,75,80,85,90,95];
eng_torque = [-12.7,37.2,86.6,135.4,183.9,231.5,278.7,325.5,371.6, ...
              417.3,462.5,507.1,551.2,594.8,637.8,680.3,722.3,763.8,804.7
              -31.3,20.3,71.1,121.1,170.2,218.4,265.7,312.2,357.8, ...
              402.6,446.5,489.5,531.7,573.0,613.4,653.0,691.7,729.5,766.5
              -46.9,7.7,61.1,113.3,164.4,214.4,263.2,311.0,357.5, ...
              402.9,447.2,490.4,532.4,573.3,613.1,651.7,689.2,725.5,760.7
              -69.7,-15.2,38.0,89.7,140.0,188.9,236.4,282.5,327.1, ...
              370.4,412.2,452.6,491.6,529.2,565.3,600,633.4,665.3,695.8];


        %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
        %  Implement Specific Information  %
        %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
mass_pharrow = [1855+409];      %  power harrow mass (kg)
mass_plough = [1524+409];       %  plough mass (kg)
mass_trailer = [12412];         %  trailer and load mass (kg)

        %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
        %  Power Harrow Draught Force and PTO Torque  %
        %%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
%  lookup table data for Power harrow draught force and PTO torque
%  input is working depth in mm (row) and forward speed in km/h (column)
%  Power Harrow working depth (mm):
phwd = [75,100,125];
%  Forward speed (km/h):
phsp = [3,4,5,6];
%  Clay soil PTO torque (Nm at the PTO):
clay_PTO = [187,245,302,360;318,385,452,518;401,480,560,639];
%  Sandy soil PTO torque (Nm at the PTO):
sand_PTO = [505,523,542,560;350,436,521,606;629,694,760,825];
%  Clay soil draught force (kN):
clay_dr = [6.8,8.6,10.3,12.1;7.5,10,12.6,15.2;10.4,12.8,15.2,17.7];
%  Sandy soil draught force (kN):
sand_dr = [11.5,14,16.6,19.1;6.8,10.6,14.5,18.3;11.9,16.4,20.9,25.3];
```

## A2.2 Model Input Data

At a constant engine speed, there is a linear relationship between the quantity of fuel injected and the torque output of the engine. Figure A2.1 shows the relationship for three engine speeds, taken from a bare engine on a dynamometer:

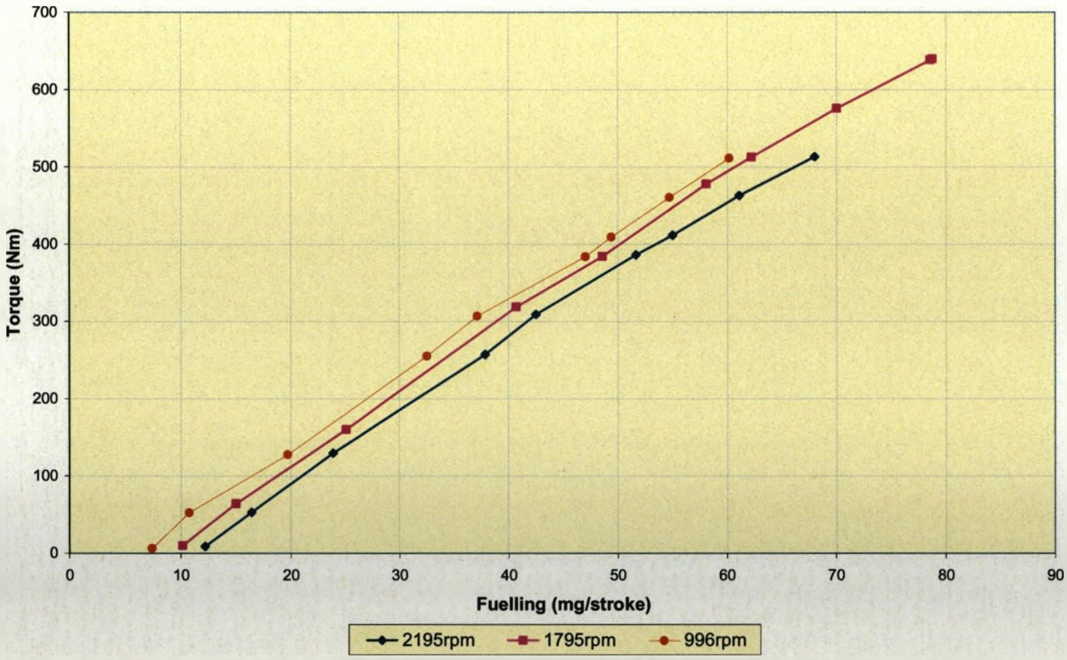


*Figure A2.1 – Flywheel torque determined by fuel quantity and engine speed*

## A2.3 Additional Model Block Diagrams

A number of sub-model block diagrams are shown to complete the model from Section 4:

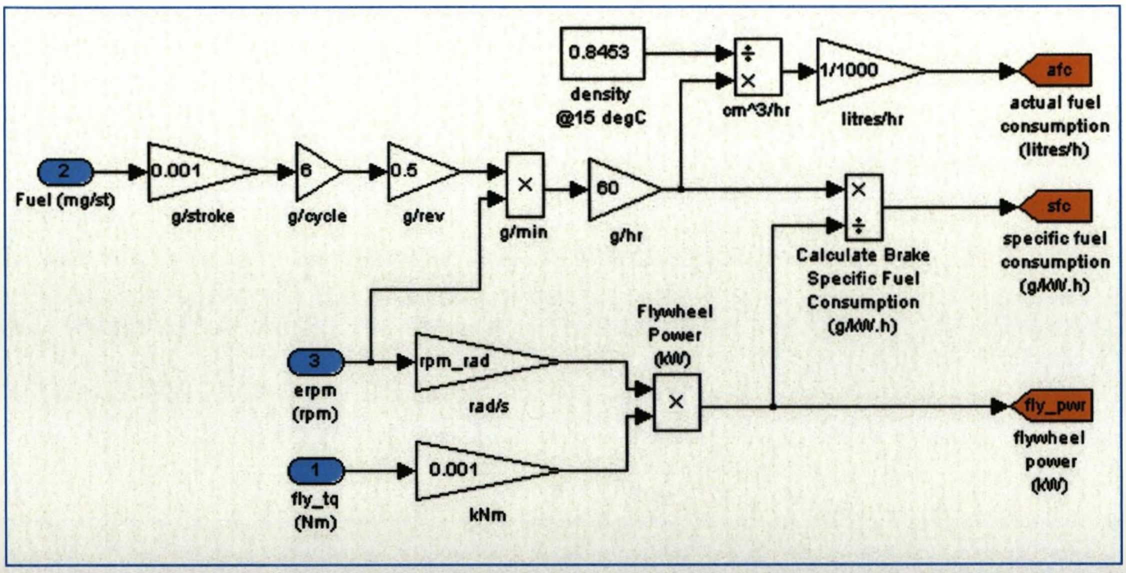


*Figure A2.2 – Engine power and fuel consumption block diagram*

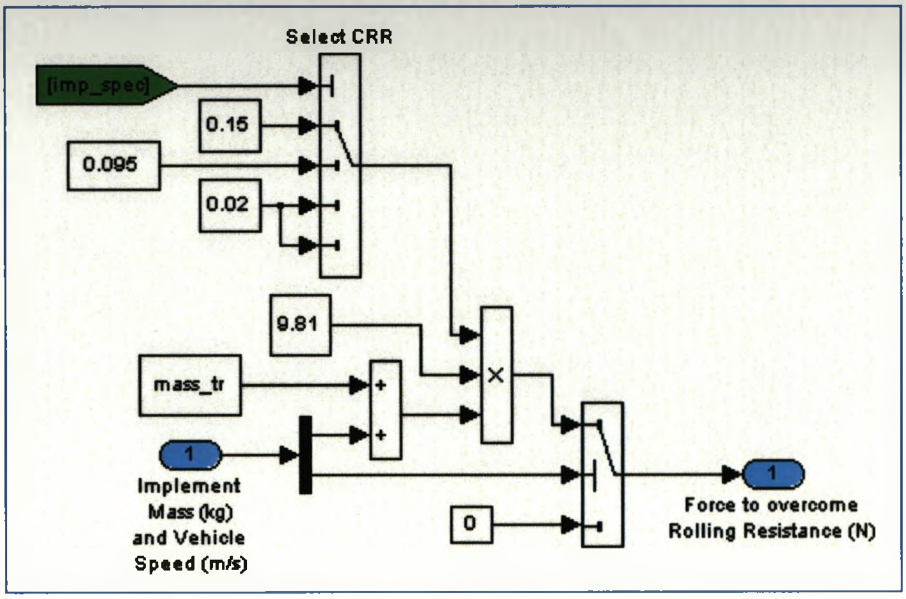


*Figure A2.3 – Rolling resistance force block diagram*

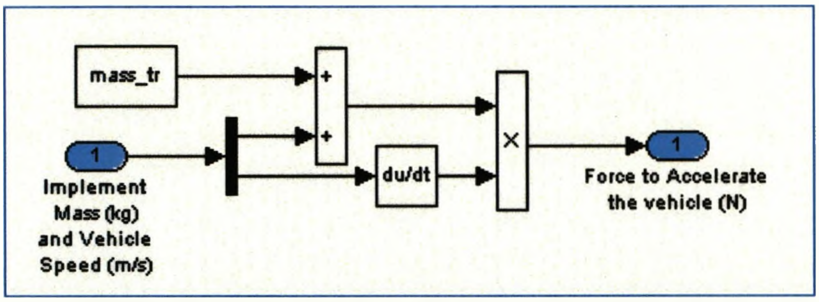


*Figure A2.4 – Vehicle Acceleration forces block diagram*

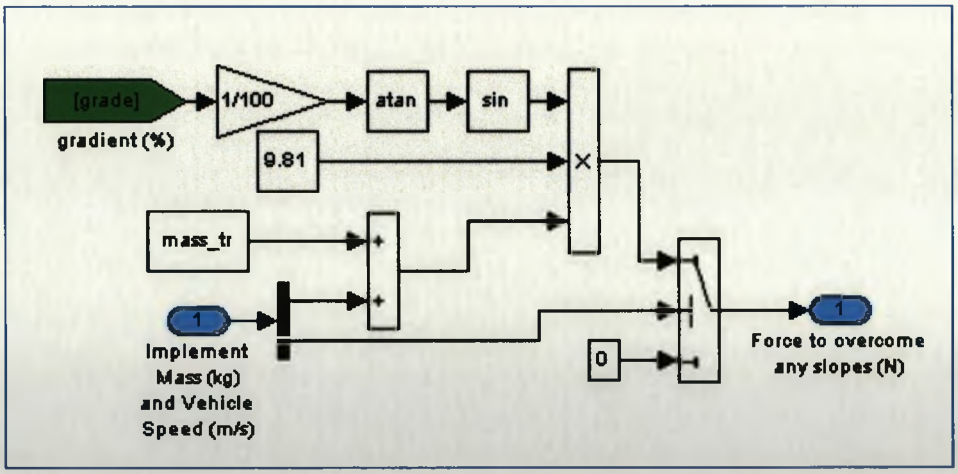


*Figure A2.5 – Forces due to slope block diagram*

# A3 Field Data Sensor Information and Calibration

## A3.1 T1 : Engine Speed

Engine speed ($\omega_E$) was measured with a variable reluctance sensor picking up on a 60 tooth tone wheel attached to the crankshaft, therefore giving a resolution of 6° of crankshaft rotation per count. Engine speed was recorded during the flywheel torque calibration process and the sensor output compared to the P.T.O. dynamometer output. The mean error of all measurements (approximately 150) was 0.02% of full scale (2370rpm) and the R.M.S. error was 0.6rpm.

## A3.2 T2 : Flywheel Torque

The calibration procedure and equations were discussed in Section 3.3.

Tables A3.1 to A3.4 show the regression data for the flywheel torque calibration:

*Table A3.1 – Linear regression analysis*

```
***** Regression Analysis *****

 Response variate: TF%
     Fitted terms: Constant, TF


*** Summary of analysis ***

                   d.f.          s.s.          m.s.       v.r.   F pr.
Regression            1        78479.      78478.89    6921.02   <.001
Residual            141         1599.         11.34
Total               142        80078.        563.93

Percentage variance accounted for 98.0
Standard error of observations is estimated to be 3.37

*** Estimates of parameters ***

                        estimate          s.e.     t(141)   t pr.
Constant                  28.468         0.578      49.23   <.001
TF                       0.12304       0.00148      83.19   <.001
```

*Table A3.2 – Linear regression analysis including engine speed*

***** Regression Analysis *****

Response variate: TF%
Fitted terms: Constant, TF, wE

*** Summary of analysis ***

| | d.f. | s.s. | m.s. | v.r. | F pr. |
|---|---|---|---|---|---|
| Regression | 2 | 78493. | 39246.64 | 3467.82 | <.001 |
| Residual | 140 | 1584. | 11.32 | | |
| Total | 142 | 80078. | 563.93 | | |

Percentage variance accounted for 98.0
Standard error of observations is estimated to be 3.36

*** Estimates of parameters ***

| | estimate | s.e. | t(140) | t pr. |
|---|---|---|---|---|
| Constant | 26.75 | 1.63 | 16.39 | <.001 |
| dynoT | 0.12353 | 0.00154 | 80.19 | <.001 |
| EngSp | 0.000876 | 0.000776 | 1.13 | 0.261 |

*Table A3.3 – Two stage calibration - regression analysis (lower model)*

***** Nonlinear regression analysis *****

Response variate: TF%
Explanatory: TF
Fitted Curve: A + B*(R**X)
Constraints: R < 1

*** Summary of analysis ***

| | d.f. | s.s. | m.s. | v.r. | F pr. |
|---|---|---|---|---|---|
| Regression | 2 | 9223.74 | 4611.8684 | 5649.33 | <.001 |
| Residual | 60 | 48.98 | 0.8164 | | |
| Total | 62 | 9272.72 | 149.5600 | | |

Percentage variance accounted for 99.5
Standard error of observations is estimated to be 0.904

*** Estimates of parameters ***

| | estimate | s.e. |
|---|---|---|
| R | 0.992264 | 0.000270 |
| B | -42.568 | 0.502 |
| A | 64.947 | 0.560 |

*Table A3.4 – Two stage regression analysis (upper model)*

***** Regression Analysis *****

Response variate: TF%
Fitted terms: Constant, TF

*** Summary of analysis ***

| | d.f. | s.s. | m.s. | v.r. | F pr. |
|---|---|---|---|---|---|
| Regression | 1 | 12218.4 | 12218.393 | 6353.61 | <.001 |
| Residual | 78 | 150.0 | 1.923 | | |
| Total | 79 | 12368.4 | 156.562 | | |

Percentage variance accounted for 98.8
Standard error of observations is estimated to be 1.39

*** Estimates of parameters ***

| | estimate | s.e. | t(78) | t pr. |
|---|---|---|---|---|
| Constant | 11.921 | 0.972 | 12.26 | <.001 |
| TF | 0.15572 | 0.00195 | 79.71 | <.001 |

## A3.3 T3 : Engine Torque

This message is the calculated actual engine output torque transmitted as a percent of the engine reference torque ($T_{E\%}$), including the torque required to overcome cylinder friction. This value is clipped to zero for negative torques. The calculation is undertaken based on engine speed and the fuelling quantity, whilst taking into account factors in the engine software limiting engine output engine such as smoke control. The J1939 message has a maximum resolution of 1% and is transmitted from the Bosch engine controller every 10 milliseconds.

This signal ($T_{E\%}$) was recorded during the flywheel torque calibration process. Linear regression was used to calibrate the CAN-bus output against the P.T.O. dynamometer data to give a value in Newton-metres for engine torque ($T_E$), the derived relationship is given by Equation A3-1:

$$T_E = \frac{T_{E\%} - 13.083}{0.1362} \qquad \textit{Equation A3-1}$$

Figure A3.1 shows the dynamometer data together with the linear prediction. As can be seen, a degree of variation occurs in $T_{E\%}$ for a given constant load applied with the P.T.O. dynamometer. These constant dynamometer loads were applied at different engine no-load speeds (as shown in Figure 3-2), some of the variation exhibited in this signal is as a result of changes in engine friction (which is included in this signal). Kimberley (2004) states that engine frictional losses increase with engine speed. As a result, the linear regression model is of limited accuracy. The R.M.S. error was 29.3Nm and the average error as a percentage of full scale (603Nm) was 4.2%. If an account for friction was made, this output could prove a useful source of engine loading data, especially in the absence of a flywheel torque sensor. In the case of this investigation, primarily the flywheel sensor has been used as a torque indicating device due to better accuracy and the provision for negative torque.

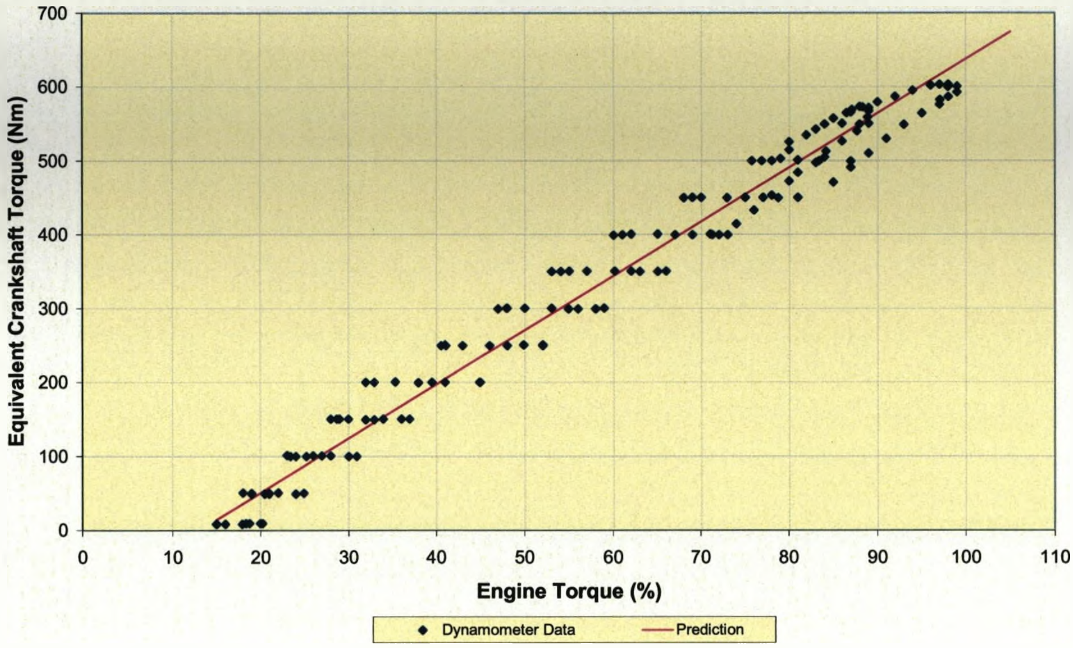


*Figure A3.1 – Engine torque data and calibration*

## A3.4 T4 : Gear

Gear number (G) is a calculated parameter determined by comparing the engine speed ($\omega_E$) to the transmission output speed ($\omega_T$). When the gearshift button is pressed the gear number changes immediately. Following clutch engagement a calculation is performed to check the correct integer is being displayed. No calibration was necessary or performed.

## A3.5 T5 : Transmission Output Speed

Transmission output speed ($\omega_T$) was measured with a Hall-effect sensor picking up on a 54 tooth gear at the rear of the transmission, therefore giving a resolution of 6.6° of rotation per count. The sensor output was recorded across the operating range at 12 different speeds, each replicated three times whilst the tractor was driven on a concrete test track. The sensor output, multiplied by the appropriate transmission ratio, was then compared to engine speed. The mean error was 0.01% of full scale (4685rpm) and the R.M.S. error was 0.75rpm.

## A3.6 T6 : Theoretical Forward Speed

Theoretical forward speed ($v_t$) was calculated from the transmission output speed, rear axle ratio and tyre specifications, with the user entering the loaded radius of the tyre. This was done and the output then checked on the concrete test track at 12 forward speeds, each replicated three times. The mean error was 0.11% of full scale (40km/h) and the R.M.S. error was 0.06km/h.

## A3.7 T7 : True Forward Speed

True forward speed ($v_a$) was measured using a 23 GHz radar fitted to the underside of the tractor with an output of 44.21Hz/MPH. Calibration was undertaken on the concrete test track by timing the tractor over 100m and comparing the actual forward speed with the transmitted CAN-bus value. 12 forward speeds throughout the operating range (0-40km/h) were each replicated three times. A small correction factor was applied to the signal resulting in a reduction of the mean error to 0.22% of full scale (40km/h) and the R.M.S. error was 0.04km/h.

## A3.8 T8 : Wheelslip

Wheelslip (S) is a calculated parameter, defined as a percentage:

$$S = \frac{v_t - v_a}{v_t} \times 100 \qquad \textit{Equation A3-2}$$

## A3.9 T9 : Foot Throttle Position

The foot throttle position ($\varepsilon_t$) was measured by a potentiometer attached to the foot throttle arm and expressed on the CAN-bus as a percentage, with 100% relating to the throttle fully depressed. A calibration check was made of this parameter, the results of which are shown in Figure A3.2. This was undertaken dynamically so the majority of the error is as a result of engine inertia and subsequent acceleration or deceleration.

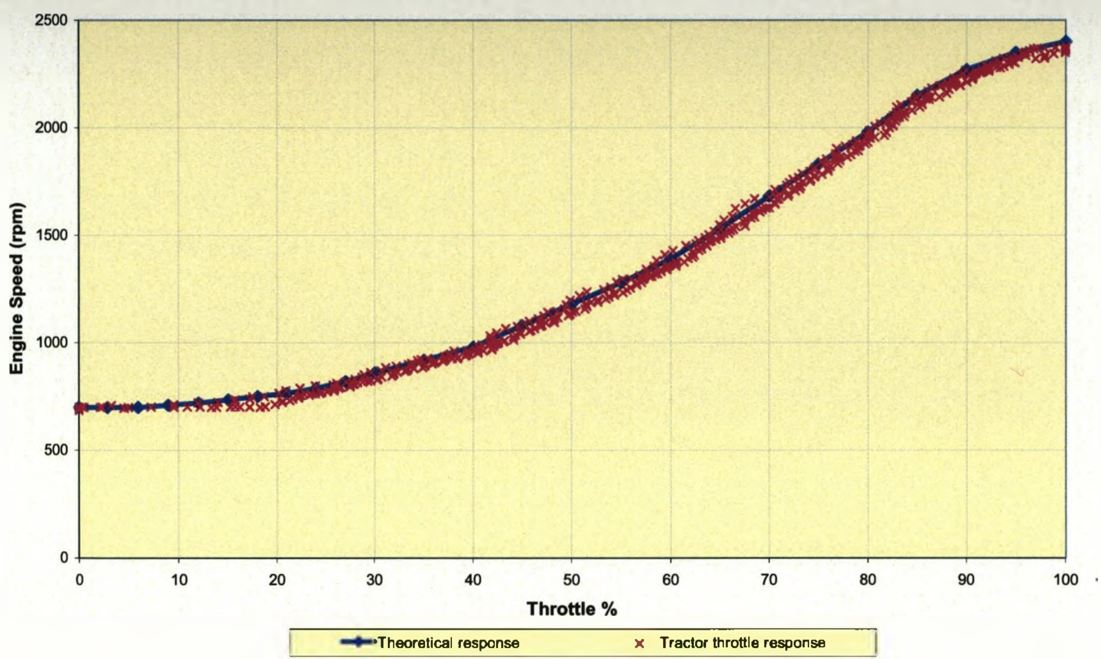


*Figure A3.2 – Ideal and actual foot throttle response*

## A3.10 T10 : Rockshaft Position

Rockshaft position ($\varepsilon_r$) was measured by a potentiometer and linkage attached to the rockshaft. The signal was expressed on the CAN-bus as a percentage, with 100% relating to the hitch being fully raised. No formal calibration was possible as the idealised response was not known. A check of performance indicated a steady change in response as the hitch travelled through its operational arc.

## A3.11 T11 : Boost Percentage

The boost percentage signal ($B_{\%}$) is a calculated parameter undertaken in the vehicle controller and transmitted via a CAN-bus message to the engine to add to the baseline torque curve. The actual value of boost is determined by the level of power being used in the traction driveline. The boost feature is detailed in Section 9.2.

## A3.12 T12 : Boost Status

Boost status ($B_S$) is transmitted as an integer between one and four, each representative of the current status of the power boost feature:

1. boost failure – sensor calibration or CAN-bus error (engine is restricted to the baseline curve);
2. boost standby – sensors ok and no errors detected but the P.T.O. is not engaged and/or vehicle is moving less the 0.5km/h (engine restricted to the baseline curve);
3. control initiated – P.T.O. is in operation, but the levels of P.T.O. torque or engine speed are below the boost threshold (engine restricted to the baseline curve);
4. power boost active – all conditions met (engine able to boost up to the maximum boosted curve).

## A3.13 T13 : Vehicle Torque Demand

This parameter ($T_{DV}$) was provided to the engine (via a CAN-bus message) by the vehicle for controlling or limiting the output torque. The parameter was expressed as a percentage of the reference torque value for the engine.

## A3.14 T14 : P.T.O. Torque (Vehicle)

The determination of P.T.O. torque from the vehicle ($T_{PTO}$) is detailed in Section 3.3.6, further information on calibration and accuracy of this parameter is discussed in Section 9.2.

## A3.15 T15 : Engine Droop

This parameter was an integer ($\delta_I$) output from the engine controller relating to the engine droop setting, and is detailed in Section 3.2. Correct change in the CAN-bus message at 12 km/h was checked during the test track calibration tests.

## A3.16 I1&2 : Draught and Vertical Forces

Calibration for the five draught and vertical force components of the Scholtz linkage was undertaken with a universal testing machine (*type ESH250, s/n 84/74*).

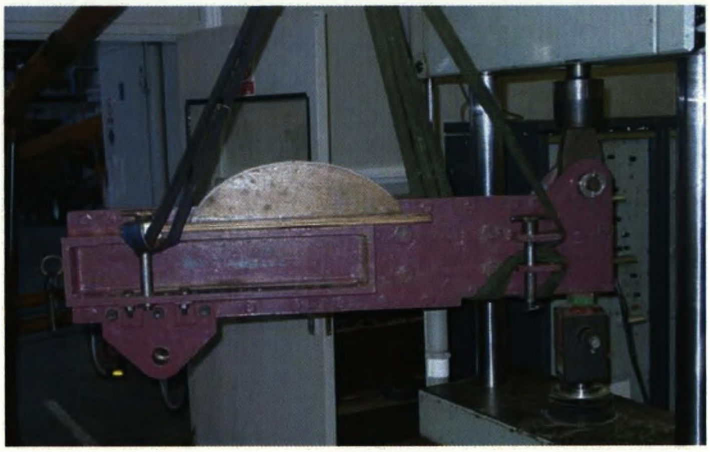

*Figure A3.3– Left leg of Scholtz linkage ready for calibration of $H_L$*

In addition to the absolute accuracy of each channel, the cross-torque in each of the legs was also considered, i.e. vertical forces being generated when the implement is loaded purely in the horizontal plane or vice versa. The resultant errors, both in terms of the applied load measurement and the cross torque are shown as a percentage of full scale (FS). R.M.S. error is also shown. (see Table A3.5).

*Table A3.5 – Scholtz linkage dynamometer calibration accuracy*

| Parameter | Full Scale (kN) | Measured Error | | Cross-Torque | | |
|---|---|---|---|---|---|---|
| | | % Full Scale | R.M.S (kN) | Source | % Full Scale | R.M.S. (kN) |
| $H_L$ | 100 | 0.04 | 0.04 | LV | 1.46 | 1.81 |
| $H_R$ | 100 | 0.06 | 0.07 | RV | 1.23 | 1.54 |
| $V_L$ | 37 | 0.46 | 0.2 | LD | 0.52 | 0.23 |
| $V_R$ | 37 | 0.55 | 0.22 | RD | 0.54 | 0.27 |
| $H_T$ | 100 | 0.11 | 0.15 | - | n/a | n/a |

## A3.17 I3 : Plough Depth Measurement

The tractor was driven onto blocks to simulate normal in-furrow operation, then a standard measure was used to calibrate the depth skid. There was a mean error of 0.6% of full scale (350mm) and the R.M.S. error was 2.9mm.

## A3.18 I4 : Plough Width Measurement

Calibration was undertaken with a standard measure. The sensor had a mean error of 0.7% of full scale (500mm) and a R.M.S. error of 4.3mm.

## A3.19 I5 : Power Harrow Depth Measurement

Calibration was undertaken against a standard measure across the full range of the depth controlling hydraulic actuators. The left sensor mean error was 0.7% of full scale (200mm) with a R.M.S. error of 1.7mm. The right sensor achieved 1.7% of full scale with a R.M.S. error of 3.8mm.

## A3.20 I6 : P.T.O. Torque Transducer

Calibration was undertaken with an in-house static torque calibration rig used to apply a force via a moment arm onto the transducer, with the provision to correct for sine errors as the unit rotates under load. The mean error was 0.16% of full scale (2000Nm) with a R.M.S. error of 2.6Nm.

## A3.21 Secondary Ploughing Parameters

In addition to the recorded parameters, a number of secondary parameters were calculated with the field data obtained.

Total horizontal force (kN):

$$H_A = H_L + H_R + H_T \qquad \textit{Equation A3-3}$$

Total vertical force (kN):

$$V_A = V_L + V_R \qquad \textit{Equation A3-4}$$

Flywheel draught torque (Nm) (that is the total draught force translated to the effective torque requirement at the engine flywheel):

$$T_D = \frac{H_A \times r}{r_d} \qquad \textit{Equation A3-5}$$

Flywheel power (kW):

$$P_F = \frac{T_F \times \omega_E}{9550} \qquad \textit{Equation A3-6}$$

Drawbar power (kW):

$$P_D = \frac{H_A \times v_a}{3.6} \qquad \textit{Equation A3-7}$$

Tractive efficiency (%), the ratio of drawbar power to flywheel power, and hence how efficient the tractor-implement system is at gaining useful work from the generated engine power:

$$\eta_T = \frac{P_D}{P_F} \times 100 \qquad \textit{Equation A3-8}$$

Theoretical worked area (ha/h) is the true forward speed multiplied by the working width, in reality a field efficiency factor would need to be included:

$$A_{PT} = \frac{W_P \times v_a}{10{,}000} \qquad \textit{Equation A3-9}$$

## A3.22 Secondary Power Harrowing Parameters

In addition to the recorded parameters, a number of secondary parameters were calculated with the field data obtained. As with the ploughing data, total horizontal force (see Equation A3-3), total vertical force (see Equation A3-4), flywheel power (see Equation A3-6) and drawbar power (see Equation A3-7) were calculated. In addition, the following parameters were also calculated:

Mean Harrow Depth (mm):

$$D_{HA} = \frac{D_{HL} + D_{HR}}{2} \qquad \textit{Equation A3-10}$$

P.T.O. Speed (rpm):

$$\omega_P = \frac{\omega_E}{2.12} \qquad \textit{Equation A3-11}$$

P.T.O. Power (kW):

$$P_P = \frac{T_{BH} \times \omega_P}{9550} \qquad \textit{Equation A3-12}$$

Lost power (kW):

$$P_L = P_F - P_D - P_P \qquad \textit{Equation A3-13}$$

Theoretical worked area (ha/h):

$$A_{HT} = \frac{W_H \times v_a}{10} \qquad \textit{Equation A3-14}$$

## A3.23 Secondary Transport Parameter

In addition to the recorded parameters, flywheel power (see Equation A3-6) was calculated with the data obtained.

# A4 Additional Field Experimental Results

## A4.1 Steady State Ploughing Data

| Target Width (mm) | Gear | Rep | | Draught Force (kN) | Vertical Force (kN) | Plough Depth (mm) | Plough Width (mm) | Rock shaft (%) | True Speed (km/h) | Wheel Speed (km/h) | Slip (%) | Engine Speed (rpm) | Trans Speed (rpm) | Draught Torque (Nm) | Engine Torque (%) | Flywheel Torque (%) | Flywheel Torque (Nm) | Flywheel Power (kW) | Drawbar Power (kW) | Tractive Efficiency (%) | Theo. Area (ha/hr) | Run |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 5 x 305 | 5 | 1 | Mean | 17.3 | 10.8 | 205 | 1525 | 10 | 5.1 | 5.3 | 4.3 | 2203 | 608 | 111 | 38.7 | 34.4 | 178 | 41.1 | 24.4 | 60 | 0.78 | 27 |
| | | | s.d. | 1.5 | 0.5 | 20 | 1 | 0.0 | 0.0 | 0.0 | 0.9 | 7 | 2 | 9 | 1.4 | 0.8 | 8 | 1.7 | 2.0 | 5 | 0.01 | |
| | | 2 | Mean | 16.5 | 11.1 | 210 | 1524 | 12 | 5.1 | 5.3 | 4.0 | 2187 | 604 | 106 | 38.5 | 34.2 | 175 | 40.1 | 23.3 | 58 | 0.77 | 20 |
| | | | s.d. | 2.0 | 0.5 | 17 | 2 | 0.5 | 0.1 | 0.0 | 1.5 | 8 | 1 | 13 | 2.4 | 1.3 | 13 | 2.9 | 2.5 | 4 | 0.01 | |
| | | 3 | Mean | 15.8 | 11.0 | 207 | 1524 | 11 | 5.1 | 5.3 | 3.5 | 2193 | 605 | 101 | 37.3 | 33.7 | 170 | 39.1 | 22.4 | 57 | 0.78 | 22 |
| | | | s.d. | 1.4 | 0.4 | 14 | 2 | 0.2 | 0.1 | 0.0 | 0.9 | 8 | 2 | 9 | 1.7 | 0.8 | 9 | 2.0 | 1.8 | 4 | 0.01 | |
| Mean of Three Replications | | | | 16.5 | 10.9 | 207 | 1524 | 11 | 5.1 | 5.3 | 3.9 | 2194 | 606 | 106 | 38.1 | 34.1 | 175 | 40.1 | 23.4 | 58 | 0.78 | |
| 5 x 305 | 7 | 1 | Mean | 18.4 | 11.5 | 206 | 1524 | 11 | 7.5 | 7.9 | 4.6 | 2188 | 902 | 176 | 56.1 | 60.1 | 294 | 67.3 | 38.5 | 57 | 1.15 | 25 |
| | | | s.d. | 1.7 | 0.5 | 13 | 2 | 0.5 | 0.1 | 0.1 | 0.9 | 11 | 4 | 17 | 2.6 | 3.1 | 20 | 4.3 | 3.5 | 4 | 0.01 | |
| | | 2 | Mean | 17.7 | 11.4 | 207 | 1522 | 11 | 7.6 | 8.0 | 4.4 | 2207 | 910 | 169 | 56.6 | 58.9 | 288 | 66.5 | 37.4 | 56 | 1.16 | 21 |
| | | | s.d. | 1.9 | 0.5 | 13 | 2 | 0.5 | 0.1 | 0.0 | 1.0 | 11 | 4 | 18 | 3.0 | 3.7 | 22 | 4.8 | 3.7 | 4 | 0.01 | |
| | | 3 | Mean | 17.8 | 11.5 | 210 | 1524 | 12 | 7.6 | 7.9 | 4.4 | 2196 | 905 | 170 | 56.2 | 59.7 | 292 | 67.2 | 37.4 | 56 | 1.16 | 23 |
| | | | s.d. | 1.6 | 0.5 | 13 | 2 | 0.4 | 0.1 | 0.0 | 1.2 | 11 | 4 | 15 | 2.9 | 3.6 | 22 | 4.8 | 3.2 | 4 | 0.02 | |
| Mean of Three Replications | | | | 17.9 | 11.5 | 208 | 1523 | 12 | 7.6 | 7.9 | 4.5 | 2197 | 906 | 172 | 56.3 | 59.6 | 291 | 67.0 | 37.8 | 56 | 1.16 | |
| 5 x 305 | 8 | 1 | Mean | 20.5 | 11.9 | 212 | 1524 | 11 | 9.2 | 9.8 | 5.6 | 2208 | 1116 | 241 | 69.9 | 75.3 | 393 | 90.9 | 52.6 | 58 | 1.41 | 26 |
| | | | s.d. | 1.5 | 0.6 | 14 | 2 | 0.4 | 0.1 | 0.1 | 0.9 | 20 | 10 | 18 | 2.1 | 2.7 | 18 | 3.4 | 3.7 | 4 | 0.02 | |
| | | 2 | Mean | 21.2 | 11.7 | 216 | 1524 | 12 | 9.1 | 9.6 | 5.9 | 2174 | 1099 | 250 | 70.5 | 76.3 | 400 | 90.8 | 53.3 | 59 | 1.38 | 19 |
| | | | s.d. | 1.8 | 0.6 | 17 | 2 | 0.1 | 0.3 | 0.2 | 1.8 | 42 | 21 | 21 | 3.6 | 4.4 | 29 | 5.1 | 3.4 | 4 | 0.05 | |
| | | 3 | Mean | 20.9 | 11.8 | 216 | 1522 | 11 | 9.2 | 9.8 | 5.6 | 2205 | 1115 | 246 | 70.0 | 75.5 | 395 | 91.1 | 53.6 | 59 | 1.41 | 24 |
| | | | s.d. | 1.6 | 0.6 | 13 | 2 | 0.3 | 0.2 | 0.1 | 1.0 | 29 | 14 | 19 | 2.7 | 3.3 | 21 | 3.9 | 4.0 | 4 | 0.03 | |
| Mean of Three Replications | | | | 20.9 | 11.8 | 214 | 1523 | 11 | 9.2 | 9.7 | 5.7 | 2196 | 1110 | 246 | 70.1 | 75.7 | 396 | 90.9 | 53.2 | 59 | 1.40 | |
| | | | | $H_A$ | $V_A$ | $D_P$ | $W_P$ | $\varepsilon_r$ | $V_a$ | $V_t$ | S | $\omega_E$ | $\omega_T$ | $T_D$ | $T_{E\%}$ | $T_{F\%}$ | $T_F$ | $P_F$ | $P_D$ | $\eta_T$ | $A_{PT}$ | |

*Table A4.1 – Field results: steady state ploughing, 305mm furrow width (sandy soil)*

| Target Width (mm) | Gear | Rep | | Draught Force (kN) | Vertical Force (kN) | Plough Depth (mm) | Plough Width (mm) | Rock shaft (%) | True Speed (km/h) | Wheel Speed (km/h) | Slip (%) | Engine Speed (rpm) | Trans Speed (rpm) | Draught Torque (Nm) | Engine Torque (%) | Flywheel Torque (%) | Flywheel Torque (Nm) | Flywheel Power (kW) | Drawbar Power (kW) | Tractive Efficiency (%) | Theo. Area (ha/hr) | Run |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 5 x 356 | 5 | 1 | Mean | 21.5 | 11.1 | 230 | 1772 | 10 | 4.9 | 5.3 | 7.3 | 2199 | 607 | 138 | 44.4 | 39.2 | 209 | 48.1 | 29.4 | 61 | 0.87 | 1 |
| | | | s.d. | 2.0 | 0.5 | 13 | 3 | 0.3 | 0.1 | 0.0 | 1.7 | 12 | 3 | 13 | 2.7 | 3.9 | 19 | 4.2 | 2.4 | 4 | 0.02 | |
| | | 2 | Mean | 19.1 | 11.1 | 211 | 1774 | 11 | 5.0 | 5.3 | 5.3 | 2190 | 604 | 123 | 40.5 | 35.4 | 185 | 42.4 | 26.6 | 63 | 0.89 | 5 |
| | | | s.d. | 2.2 | 0.4 | 15 | 2 | 0.2 | 0.1 | 0.0 | 1.5 | 10 | 3 | 14 | 3.0 | 2.3 | 16 | 3.6 | 2.7 | 4 | 0.02 | |
| | | 3 | Mean | 18.8 | 11.1 | 206 | 1773 | 11 | 5.1 | 5.3 | 4.9 | 2205 | 608 | 121 | 40.0 | 34.8 | 181 | 41.9 | 26.5 | 63 | 0.90 | 9 |
| | | | s.d. | 1.7 | 0.5 | 13 | 2 | 0.0 | 0.1 | 0.0 | 1.5 | 8 | 2 | 11 | 1.8 | 1.3 | 11 | 2.4 | 2.1 | 4 | 0.01 | |
| Mean of Three Replications | | | | 19.8 | 11.1 | 215 | 1773 | 11 | 5.0 | 5.3 | 5.8 | 2198 | 606 | 127 | 41.6 | 36.5 | 192 | 44.1 | 27.5 | 63 | 0.89 | |
| 5 x 356 | 7 | 1 | Mean | 22.4 | 11.6 | 221 | 1772 | 11 | 7.3 | 7.9 | 7.4 | 2190 | 903 | 214 | 63.5 | 67.4 | 342 | 78.4 | 45.5 | 58 | 1.30 | 3 |
| | | | s.d. | 2.2 | 0.6 | 13 | 3 | 0.8 | 0.1 | 0.1 | 1.4 | 13 | 5 | 21 | 3.4 | 3.8 | 25 | 5.4 | 4.0 | 4 | 0.02 | |
| | | 2 | Mean | 20.4 | 11.6 | 214 | 1773 | 11 | 7.4 | 7.9 | 6.5 | 2194 | 904 | 196 | 61.0 | 64.1 | 320 | 73.4 | 42.1 | 57 | 1.31 | 4 |
| | | | s.d. | 2.1 | 0.6 | 14 | 3 | 0.3 | 0.1 | 0.0 | 1.3 | 12 | 5 | 20 | 3.7 | 4.1 | 27 | 5.9 | 3.9 | 4 | 0.02 | |
| | | 3 | Mean | 19.6 | 11.5 | 206 | 1774 | 11 | 7.5 | 7.9 | 5.6 | 2184 | 900 | 188 | 58.9 | 62.8 | 312 | 71.3 | 40.6 | 57 | 1.32 | 8 |
| | | | s.d. | 1.9 | 0.5 | 14 | 2 | 0.0 | 0.1 | 0.0 | 1.4 | 12 | 5 | 18 | 3.4 | 3.8 | 25 | 5.4 | 3.6 | 4 | 0.02 | |
| Mean of Three Replications | | | | 20.8 | 11.6 | 214 | 1773 | 11 | 7.4 | 7.9 | 6.5 | 2189 | 902 | 199 | 61.1 | 64.8 | 324 | 74.3 | 42.7 | 58 | 1.31 | |
| 5 x 356 | 8 | 1 | Mean | 24.8 | 11.6 | 230 | 1771 | 10 | 8.4 | 9.2 | 9.1 | 2073 | 1048 | 292 | 76.6 | 83.4 | 446 | 96.8 | 57.6 | 59 | 1.48 | 2 |
| | | | s.d. | 2.0 | 0.6 | 20 | 2 | 0.2 | 0.3 | 0.2 | 1.4 | 47 | 24 | 23 | 1.9 | 3.1 | 20 | 2.8 | 4.3 | 4 | 0.05 | |
| | | 2 | Mean | 25.4 | 12.4 | 221 | 1769 | 11 | 8.3 | 9.1 | 8.9 | 2060 | 1041 | 298 | 77.0 | 84.0 | 450 | 98.9 | 58.5 | 60 | 1.47 | 6 |
| | | | s.d. | 2.4 | 0.8 | 17 | 4 | 1.5 | 0.4 | 0.3 | 2.0 | 74 | 38 | 28 | 3.0 | 4.3 | 28 | 3.4 | 4.6 | 5 | 0.08 | |
| | | 3 | Mean | 20.6 | 11.7 | 203 | 1771 | 10 | 8.9 | 9.6 | 7.0 | 2162 | 1093 | 242 | 72.0 | 77.1 | 405 | 91.5 | 50.8 | 56 | 1.58 | 7 |
| | | | s.d. | 2.9 | 0.8 | 20 | 3 | 0.3 | 0.4 | 0.3 | 2.1 | 69 | 35 | 34 | 4.0 | 5.3 | 35 | 5.3 | 5.8 | 6 | 0.08 | |
| Mean of Three Replications | | | | 23.6 | 11.9 | 218 | 1770 | 10 | 8.5 | 9.3 | 8.3 | 2098 | 1061 | 277 | 75.2 | 81.5 | 434 | 95.1 | 55.6 | 59 | 1.51 | |
| | | | | $H_A$ | $V_A$ | $D_P$ | $W_P$ | $\varepsilon_l$ | $v_a$ | $v_t$ | S | $\omega_E$ | $\omega_T$ | $T_D$ | $T_{E\%}$ | $T_{F\%}$ | $T_F$ | $P_F$ | $P_D$ | $\eta_T$ | $A_{PT}$ | |

*Table A4.2 – Field results: steady state ploughing, 356mm furrow width (sandy soil)*

| Target Width (mm) | Gear | Rep | | Draught Force (kN) | Vertical Force (kN) | Plough Depth (mm) | Plough Width (mm) | Rock shaft (%) | True Speed (km/h) | Wheel Speed (km/h) | Slip (%) | Engine Speed (rpm) | Trans Speed (rpm) | Draught Torque (Nm) | Engine Torque (%) | Flywheel Torque (%) | Flywheel Torque (Nm) | Flywheel Power (kW) | Drawbar Power (kW) | Tractive Efficiency (%) | Theo. Area (ha/hr) | Run |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 5 x 406 | 5 | 1 | Mean | 20.2 | 11.0 | 215 | 2033 | 11 | 4.9 | 5.3 | 7.1 | 2200 | 607 | 130 | 43.7 | 38.8 | 206 | 47.3 | 27.7 | 59 | 1.01 | 14 |
| | | | s.d. | 2.7 | 0.7 | 16 | 3 | 0.0 | 0.1 | 0.0 | 2.0 | 12 | 3 | 17 | 3.5 | 4.3 | 22 | 5.0 | 3.3 | 5 | 0.02 | |
| | | 2 | Mean | 19.4 | 11.3 | 209 | 2035 | 11 | 5.0 | 5.3 | 6.0 | 2188 | 604 | 125 | 42.0 | 36.7 | 195 | 44.5 | 26.8 | 60 | 1.01 | 17 |
| | | | s.d. | 2.0 | 0.5 | 12 | 2 | 0.3 | 0.1 | 0.0 | 1.3 | 11 | 3 | 13 | 2.8 | 2.6 | 16 | 3.6 | 2.5 | 4 | 0.02 | |
| | | 3 | Mean | 19.3 | 11.2 | 222 | 2031 | 11 | 5.0 | 5.4 | 6.9 | 2216 | 612 | 124 | 43.4 | 38.7 | 204 | 47.3 | 26.8 | 57 | 1.01 | 12 |
| | | | s.d. | 3.0 | 0.5 | 15 | 2 | 0.0 | 0.1 | 0.0 | 1.8 | 11 | 3 | 19 | 3.8 | 4.5 | 25 | 5.6 | 3.7 | 5 | 0.02 | |
| Mean of Three Replications | | | | 19.7 | 11.2 | 215 | 2033 | 11 | 5.0 | 5.3 | 6.7 | 2201 | 608 | 126 | 43.0 | 38.0 | 201 | 46.4 | 27.1 | 59 | 1.01 | |
| 5 x 406 | 7 | 1 | Mean | 21.9 | 11.6 | 209 | 2032 | 12 | 7.3 | 7.9 | 7.3 | 2192 | 904 | 210 | 62.9 | 67.0 | 339 | 77.8 | 44.6 | 57 | 1.49 | 15 |
| | | | s.d. | 2.0 | 0.6 | 15 | 2 | 0.5 | 0.1 | 0.1 | 1.5 | 13 | 5 | 19 | 3.4 | 3.8 | 25 | 5.4 | 3.9 | 5 | 0.03 | |
| | | 2 | Mean | 23.4 | 11.9 | 219 | 2031 | 11 | 7.3 | 8.0 | 8.2 | 2211 | 911 | 224 | 65.6 | 69.5 | 355 | 82.2 | 47.6 | 58 | 1.48 | 16 |
| | | | s.d. | 1.9 | 0.5 | 13 | 2 | 0.2 | 0.1 | 0.1 | 1.3 | 13 | 5 | 19 | 3.3 | 3.7 | 24 | 5.3 | 3.6 | 4 | 0.02 | |
| | | 3 | Mean | 22.8 | 11.8 | 213 | 2023 | 11 | 7.2 | 7.9 | 8.2 | 2182 | 899 | 219 | 64.3 | 68.7 | 350 | 79.9 | 45.8 | 57 | 1.46 | 10 |
| | | | s.d. | 2.0 | 0.5 | 13 | 2 | 0.5 | 0.1 | 0.0 | 1.5 | 11 | 5 | 19 | 3.2 | 3.4 | 22 | 4.8 | 3.8 | 4 | 0.03 | |
| Mean of Three Replications | | | | 22.7 | 11.8 | 214 | 2029 | 11 | 7.3 | 7.9 | 7.9 | 2195 | 905 | 218 | 64.3 | 68.4 | 348 | 80.0 | 46.0 | 58 | 1.48 | |
| 5 x 406 | 8 | 1 | Mean | 24.1 | 11.8 | 225 | 2031 | 11 | 8.1 | 9.0 | 10.2 | 2035 | 1029 | 283 | 77.9 | 85.4 | 459 | 97.6 | 53.9 | 55 | 1.65 | 13 |
| | | | s.d. | 2.6 | 0.6 | 14 | 3 | 0.1 | 0.5 | 0.4 | 2.0 | 88 | 44 | 30 | 3.2 | 4.2 | 27 | 2.6 | 4.7 | 5 | 0.10 | |
| | | 2 | Mean | 26.1 | 12.2 | 227 | 2032 | 11 | 8.1 | 9.0 | 9.7 | 2034 | 1028 | 307 | 78.0 | 84.9 | 456 | 97.0 | 59.1 | 61 | 1.66 | 18 |
| | | | s.d. | 1.6 | 0.6 | 13 | 3 | 0.3 | 0.4 | 0.3 | 1.6 | 63 | 32 | 19 | 2.4 | 3.3 | 22 | 2.4 | 3.9 | 4 | 0.07 | |
| | | 3 | Mean | 24.9 | 12.0 | 221 | 2031 | 11 | 8.3 | 9.1 | 9.4 | 2058 | 1041 | 293 | 77.1 | 84.3 | 452 | 97.3 | 57.1 | 59 | 1.68 | 11 |
| | | | s.d. | 1.7 | 0.5 | 12 | 3 | 0.1 | 0.3 | 0.3 | 1.5 | 60 | 30 | 20 | 2.5 | 3.2 | 21 | 2.4 | 3.5 | 4 | 0.07 | |
| Mean of Three Replications | | | | 25.0 | 12.0 | 224 | 2031 | 11 | 8.2 | 9.0 | 9.8 | 2042 | 1033 | 294 | 77.7 | 84.9 | 456 | 97.3 | 56.7 | 58 | 1.66 | |
| | | | | $H_A$ | $V_A$ | $D_P$ | $W_P$ | $\varepsilon_t$ | $v_a$ | $v_t$ | $S$ | $\omega_E$ | $\omega_T$ | $T_D$ | $T_{ES}$ | $T_{FS}$ | $T_F$ | $P_F$ | $P_D$ | $\eta_T$ | $A_{PT}$ | |

*Table A4.3 – Field results: steady state ploughing, 406mm furrow width (sandy soil)*

| Target Width (mm) | Gear | Rep | | Draught Force (kN) | Vertical Force (kN) | Plough Depth (mm) | Plough Width (mm) | Rock shaft (%) | True Speed (km/h) | Wheel Speed (km/h) | Slip (%) | Engine Speed (rpm) | Trans Speed (rpm) | Draught Torque (Nm) | Engine Torque (%) | Flywheel Torque (%) | Flywheel Torque (Nm) | Flywheel Power (kW) | Drawbar Power (kW) | Tractive Efficiency (%) | Theo. Area (ha/hr) | Run |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 5 x 305 | 4 | 1 | Mean | 31.4 | 11.3 | 215 | 1520 | 14 | 3.6 | 4.1 | 13.0 | 2192 | 471 | 157 | 45.7 | 56.5 | 211 | 48.5 | 31.3 | 65 | 0.55 | 36 |
| | | | s.d. | 4.2 | 0.9 | 14 | 0 | 0.2 | 0.1 | 0.0 | 0.0 | 22 | 5 | 21 | 4.4 | 1.6 | 28 | 6.1 | 3.6 | 7 | 0.02 | |
| | | 2 | Mean | 30.7 | 11.7 | 202 | 1520 | 14 | 3.6 | 4.2 | 12.9 | 2218 | 477 | 154 | 45.7 | 56.0 | 204 | 47.2 | 31.0 | 66 | 0.55 | 33 |
| | | | s.d. | 4.5 | 0.7 | 16 | 0 | 0.3 | 0.1 | 0.0 | 3.0 | 21 | 4 | 23 | 4.7 | 1.6 | 26 | 5.7 | 3.9 | 7 | 0.02 | |
| | | 3 | Mean | 29.9 | 11.3 | 221 | 1520 | 14 | 3.7 | 4.2 | 12.0 | 2218 | 477 | 150 | 45.7 | 56.3 | 207 | 48.0 | 30.5 | 64 | 0.56 | 34 |
| | | | s.d. | 4.0 | 0.8 | 15 | 0 | 0.4 | 0.1 | 0.0 | 2.7 | 19 | 4 | 20 | 4.1 | 1.5 | 25 | 5.5 | 3.4 | 6 | 0.02 | |
| Mean of Three Replications | | | | 30.7 | 11.4 | 213 | 1520 | 14 | 3.6 | 4.2 | 12.7 | 2209 | 475 | 154 | 45.7 | 56.3 | 207 | 47.9 | 30.9 | 65 | 0.55 | |
| 5 x 305 | 5 | 1 | Mean | 30.7 | 11.4 | 222 | 1520 | 14 | 4.6 | 5.3 | 12.5 | 2179 | 601 | 197 | 53.3 | 59.5 | 267 | 60.9 | 39.2 | 65 | 0.70 | 28 |
| | | | s.d. | 3.9 | 0.8 | 17 | 0 | 0.4 | 0.2 | 0.1 | 3.6 | 21 | 6 | 25 | 4.3 | 1.9 | 36 | 7.7 | 4.5 | 8 | 0.03 | |
| | | 2 | Mean | 30.8 | 11.4 | 228 | 1520 | 14 | 4.6 | 5.3 | 12.7 | 2195 | 605 | 197 | 54.0 | 59.8 | 273 | 62.7 | 39.5 | 64 | 0.70 | 31 |
| | | | s.d. | 4.2 | 0.8 | 19 | 0 | 0.4 | 0.2 | 0.1 | 3.0 | 23 | 6 | 27 | 5.0 | 2.3 | 42 | 9.0 | 4.6 | 8 | 0.03 | |
| | | 3 | Mean | 31.0 | 11.2 | 218 | 1520 | 14 | 4.6 | 5.3 | 13.1 | 2194 | 605 | 199 | 54.2 | 60.3 | 283 | 65.0 | 39.6 | 61 | 0.70 | 35 |
| | | | s.d. | 4.3 | 0.8 | 14 | 0 | 0.3 | 0.1 | 0.1 | 2.4 | 22 | 6 | 28 | 4.6 | 2.2 | 39 | 8.4 | 4.9 | 8 | 0.02 | |
| Mean of Three Replications | | | | 30.8 | 11.3 | 223 | 1520 | 14 | 4.6 | 5.3 | 12.7 | 2190 | 603 | 198 | 53.8 | 59.9 | 275 | 62.9 | 39.4 | 63 | 0.70 | |
| 5 x 305 | 6 | 1 | Mean | 35.0 | 11.7 | 230 | 1520 | 15 | 5.4 | 6.4 | 16.4 | 2170 | 734 | 275 | 66.9 | 69.3 | 368 | 83.5 | 52.0 | 63 | 0.82 | 27 |
| | | | s.d. | 4.4 | 1.0 | 17 | 0 | 0.5 | 0.5 | 0.1 | 6.6 | 44 | 15 | 35 | 5.7 | 6.5 | 42 | 7.9 | 6.4 | 9 | 0.08 | |
| | | 2 | Mean | 32.7 | 11.5 | 216 | 1520 | 14 | 5.6 | 6.4 | 13.2 | 2170 | 734 | 257 | 63.2 | 65.7 | 344 | 78.0 | 50.6 | 65 | 0.85 | 32 |
| | | | s.d. | 3.9 | 0.8 | 14 | 0 | 0.3 | 0.2 | 0.1 | 2.7 | 25 | 8 | 30 | 5.1 | 4.9 | 34 | 6.9 | 5.3 | 7 | 0.03 | |
| | | 3 | Mean | 34.1 | 11.9 | 191 | 1520 | 15 | 5.6 | 6.5 | 13.8 | 2190 | 740 | 268 | 65.4 | 68.8 | 365 | 83.5 | 52.9 | 64 | 0.85 | 25 |
| | | | s.d. | 4.4 | 1.0 | 18 | 0 | 0.1 | 0.2 | 0.1 | 3.0 | 32 | 11 | 35 | 5.6 | 6.2 | 41 | 8.4 | 6.2 | 8 | 0.04 | |
| Mean of Three Replications | | | | 33.9 | 11.7 | 212 | 1520 | 15 | 5.5 | 6.5 | 14.5 | 2176 | 736 | 267 | 65.2 | 67.9 | 359 | 81.6 | 51.8 | 64 | 0.84 | |
| 5 x 305 | 7 | 1 | Mean | 31.7 | 11.5 | 205 | 1520 | 14 | 6.7 | 7.6 | 12.7 | 2110 | 869 | 304 | 75.0 | 77.0 | 418 | 92.2 | 58.6 | 64 | 1.01 | 29 |
| | | | s.d. | 3.6 | 0.9 | 14 | 0 | 0.5 | 0.3 | 0.2 | 2.1 | 62 | 25 | 35 | 2.7 | 5.1 | 33 | 5.4 | 6.6 | 7 | 0.04 | |
| | | 2 | Mean | 34.8 | 11.7 | 221 | 1520 | 15 | 6.1 | 7.3 | 15.5 | 2009 | 827 | 334 | 78.8 | 85.4 | 472 | 99.0 | 59.2 | 60 | 0.93 | 26 |
| | | | s.d. | 4.3 | 1.1 | 18 | 0 | 0.1 | 0.5 | 0.0 | 3.4 | 99 | 41 | 41 | 3.3 | 6.4 | 41 | 6.0 | 7.4 | 7 | 0.07 | |
| | | 3 | Mean | 29.9 | 11.3 | 212 | 1520 | 14 | 6.8 | 7.6 | 10.5 | 2104 | 867 | 287 | 70.7 | 72.7 | 390 | 85.9 | 56.5 | 66 | 1.03 | 30 |
| | | | s.d. | 3.7 | 0.9 | 15 | 0 | 0.4 | 0.2 | 0.1 | 2.3 | 37 | 15 | 36 | 5.1 | 6.3 | 41 | 7.6 | 6.6 | 8 | 0.04 | |
| Mean of Three Replications | | | | 32.2 | 11.5 | 213 | 1520 | 14 | 6.5 | 7.5 | 12.9 | 2074 | 855 | 308 | 74.8 | 78.4 | 427 | 92.4 | 58.1 | 63 | 0.99 | |
| | | | | $H_A$ | $V_A$ | $D_P$ | $W_P$ | $\varepsilon_r$ | $v_a$ | $v_t$ | S | $\omega_E$ | $\omega_T$ | $T_D$ | $T_{ES}$ | $T_{FS}$ | $T_F$ | $P_F$ | $P_D$ | $\eta_T$ | $A_{PT}$ | |

*Table A4.4 – Field results: steady state ploughing, 305mm furrow width (clay soil)*

| Target Width (mm) | Gear | Rep | | Draught Force (kN) | Vertical Force (kN) | Plough Depth (mm) | Plough Width (mm) | Rock shaft (%) | True Speed (km/h) | Wheel Speed (km/h) | Slip (%) | Engine Speed (rpm) | Trans Speed (rpm) | Draught Torque (Nm) | Engine Torque (%) | Flywheel Torque (%) | Flywheel Torque (Nm) | Flywheel Power (kW) | Drawbar Power (kW) | Tractive Efficiency (%) | Theo. Area (ha/hr) | Run |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 5 x 356 | 4 | 1 | Mean | 30.8 | 11.5 | 192 | 1744 | 14 | 3.6 | 4.1 | 11.8 | 2192 | 471 | 154 | 47.0 | 56.5 | 212 | 48.6 | 31.1 | 64 | 0.64 | 7 |
| | | | s.d. | 4.6 | 0.8 | 14 | 0 | 0.5 | 0.1 | 0.0 | 2.5 | 21 | 4 | 23 | 4.2 | 1.7 | 30 | 6.5 | 4.0 | 6 | 0.02 | |
| | | 2 | Mean | 35.3 | 11.7 | 222 | 1739 | 15 | 3.5 | 4.1 | 16.3 | 2188 | 471 | 177 | 50.5 | 57.8 | 236 | 54.0 | 33.7 | 63 | 0.60 | 10 |
| | | | s.d. | 5.8 | 1.0 | 15 | 1 | 0.5 | 0.2 | 0.0 | 4.0 | 23 | 5 | 29 | 5.1 | 2.1 | 39 | 8.5 | 4.6 | 7 | 0.03 | |
| | | 3 | Mean | 31.4 | 11.4 | 195 | 1743 | 15 | 3.6 | 4.2 | 14.2 | 2199 | 473 | 157 | 47.5 | 57.2 | 222 | 51.0 | 31.0 | 61 | 0.62 | 9 |
| | | | s.d. | 4.7 | 0.8 | 17 | 1 | 0.5 | 0.1 | 0.0 | 2.4 | 21 | 4 | 24 | 4.4 | 1.7 | 31 | 6.8 | 4.1 | 6 | 0.02 | |
| Mean of Three Replications | | | | 32.5 | 11.6 | 203 | 1742 | 15 | 3.6 | 4.1 | 14.1 | 2193 | 472 | 163 | 48.3 | 57.2 | 223 | 51.2 | 31.9 | 63 | 0.62 | |
| 5 x 356 | 5 | 1 | Mean | 41.2 | 12.1 | 230 | 1741 | 15 | 4.0 | 5.2 | 22.6 | 2134 | 588 | 265 | 66.3 | 70.1 | 371 | 82.7 | 45.2 | 55 | 0.70 | 1 |
| | | | s.d. | 6.9 | 1.1 | 25 | 1 | 0.4 | 0.4 | 0.1 | 7.1 | 35 | 10 | 44 | 7.1 | 7.5 | 52 | 10.5 | 5.8 | 7 | 0.07 | |
| | | 2 | Mean | 38.5 | 11.7 | 225 | 1742 | 15 | 4.3 | 5.3 | 19.3 | 2181 | 601 | 247 | 62.6 | 63.2 | 321 | 73.2 | 45.3 | 62 | 0.74 | 12 |
| | | | s.d. | 4.8 | 1.0 | 19 | 0 | 0.5 | 0.2 | 0.1 | 4.1 | 25 | 7 | 31 | 5.7 | 3.3 | 33 | 6.9 | 4.9 | 7 | 0.04 | |
| | | 3 | Mean | 35.4 | 11.8 | 219 | 1743 | 15 | 4.3 | 5.1 | 16.4 | 2119 | 584 | 227 | 59.0 | 62.1 | 310 | 68.8 | 42.0 | 61 | 0.75 | 3 |
| | | | s.d. | 4.6 | 0.9 | 17 | 1 | 0.5 | 0.2 | 0.1 | 3.5 | 22 | 6 | 30 | 5.0 | 3.0 | 33 | 6.7 | 4.7 | 6 | 0.03 | |
| Mean of Three Replications | | | | 38.4 | 11.9 | 225 | 1742 | 15 | 4.2 | 5.2 | 19.4 | 2145 | 591 | 246 | 62.6 | 65.1 | 334 | 74.9 | 44.2 | 60 | 0.73 | |
| 5 x 356 | 6 | 1 | Mean | 37.9 | 11.7 | 220 | 1743 | 15 | 5.1 | 6.3 | 19.4 | 2125 | 719 | 299 | 72.4 | 74.6 | 403 | 89.4 | 53.4 | 60 | 0.89 | 5 |
| | | | s.d. | 4.2 | 1.0 | 18 | 1 | 0.4 | 0.3 | 0.2 | 4.4 | 49 | 17 | 33 | 4.4 | 6.1 | 39 | 6.9 | 5.9 | 7 | 0.06 | |
| | | 2 | Mean | 41.0 | 11.9 | 231 | 1738 | 15 | 4.7 | 6.2 | 24.3 | 2086 | 705 | 323 | 75.5 | 75.1 | 406 | 88.2 | 53.1 | 60 | 0.82 | 11 |
| | | | s.d. | 4.8 | 1.0 | 21 | 1 | 0.4 | 0.6 | 0.3 | 7.1 | 114 | 38 | 38 | 4.7 | 7.0 | 45 | 5.9 | 6.9 | 9 | 0.11 | |
| | | 3 | Mean | 41.9 | 12.1 | 218 | 1743 | 15 | 4.5 | 6.0 | 24.2 | 2009 | 679 | 330 | 75.6 | 78.9 | 430 | 90.0 | 51.6 | 58 | 0.79 | 2 |
| | | | s.d. | 6.2 | 1.2 | 19 | 1 | 0.3 | 0.8 | 0.3 | 10.0 | 105 | 36 | 49 | 6.8 | 8.5 | 55 | 8.0 | 8.2 | 10 | 0.13 | |
| Mean of Three Replications | | | | 40.3 | 11.9 | 223 | 1741 | 15 | 4.8 | 6.2 | 22.6 | 2073 | 701 | 317 | 74.5 | 76.2 | 413 | 89.2 | 52.7 | 59 | 0.83 | |
| 5 x 356 | 7 | 1 | Mean | 38.5 | 11.9 | 215 | 1741 | 15 | 5.3 | 6.6 | 18.4 | 1794 | 739 | 366 | 83.9 | 88.6 | 492 | 91.8 | 55.9 | 61 | 0.93 | 4 |
| | | | s.d. | 5.2 | 1.1 | 15 | 1 | 0.5 | 0.7 | 0.6 | 4.2 | 191 | 79 | 50 | 3.6 | 6.9 | 44 | 7.1 | 8.9 | 7 | 0.12 | |
| | | 2 | Mean | 33.3 | 12.0 | 194 | 1721 | 16 | 6.5 | 7.5 | 13.2 | 2071 | 853 | 319 | 76.5 | 78.4 | 427 | 92.3 | 59.9 | 65 | 1.12 | 6 |
| | | | s.d. | 4.2 | 1.0 | 12 | 1 | 0.5 | 0.4 | 0.3 | 2.5 | 85 | 35 | 40 | 3.1 | 5.6 | 36 | 5.4 | 7.1 | 7 | 0.07 | |
| | | 3 | Mean | 37.2 | 11.7 | 198 | 1741 | 15 | 5.6 | 6.8 | 17.4 | 1876 | 773 | 357 | 82.4 | 89.0 | 495 | 96.8 | 57.6 | 59 | 0.98 | 8 |
| | | | s.d. | 5.1 | 1.0 | 15 | 1 | 0.4 | 0.6 | 0.5 | 3.6 | 149 | 61 | 49 | 3.9 | 6.5 | 42 | 5.7 | 7.5 | 6 | 0.10 | |
| Mean of Three Replications | | | | 36.3 | 11.9 | 202 | 1734 | 15 | 5.8 | 6.9 | 16.3 | 1914 | 788 | 347 | 81.0 | 85.3 | 471 | 93.6 | 57.8 | 62 | 1.01 | |
| | | | | $H_A$ | $V_A$ | $D_P$ | $W_P$ | $\varepsilon_r$ | $V_a$ | $V_t$ | S | $\omega_E$ | $\omega_T$ | $T_D$ | $T_{E\%}$ | $T_{F\%}$ | $T_F$ | $P_F$ | $P_D$ | $\eta_T$ | $A_{PT}$ | |

*Table A4.5 – Field results: steady state ploughing, 356mm furrow width (clay soil)*

| Target Width (mm) | Gear | Rep | | Draught Force (kN) | Vertical Force (kN) | Plough Depth (mm) | Plough Width (mm) | Rock shaft (%) | True Speed (km/h) | Wheel Speed (km/h) | Slip (%) | Engine Speed (rpm) | Trans Speed (rpm) | Draught Torque (Nm) | Engine Torque (%) | Flywheel Torque (%) | Flywheel Torque (Nm) | Flywheel Power (kW) | Drawbar Power (kW) | Tractive Efficiency (%) | Theo. Area (ha/hr) | Run |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 5 x 406 | 4 | 1 | Mean | 39.7 | 12.2 | 243 | 2009 | 15 | 3.2 | 4.2 | 24.0 | 2210 | 475 | 199 | 55.5 | 60.6 | 291 | 67.2 | 34.8 | 52 | 0.64 | 16 |
| | | | s.d. | 5.1 | 1.2 | 20 | 1 | 0.1 | 0.2 | 0.0 | 4.4 | 20 | 4 | 26 | 4.3 | 1.8 | 35 | 7.6 | 4.3 | 7 | 0.04 | |
| | | 2 | Mean | 32.8 | 11.5 | 229 | 2013 | 14 | 3.4 | 4.2 | 19.7 | 2223 | 478 | 164 | 49.7 | 57.9 | 238 | 55.2 | 30.5 | 56 | 0.68 | 24 |
| | | | s.d. | 5.1 | 1.0 | 18 | 1 | 0.1 | 0.2 | 0.0 | 5.6 | 20 | 4 | 26 | 4.7 | 2.0 | 39 | 8.7 | 3.7 | 8 | 0.05 | |
| | | 3 | Mean | 35.1 | 11.7 | 240 | 2011 | 14 | 3.3 | 4.2 | 21.5 | 2218 | 477 | 176 | 51.8 | 58.5 | 248 | 57.4 | 31.6 | 56 | 0.66 | 20 |
| | | | s.d. | 6.0 | 1.0 | 16 | 0 | 0.0 | 0.4 | 0.0 | 8.6 | 21 | 5 | 30 | 5.4 | 2.6 | 47 | 10.3 | 3.9 | 10 | 0.08 | |
| Mean of Three Replications | | | | 35.9 | 11.8 | 237 | 2011 | 14 | 3.3 | 4.2 | 21.7 | 2217 | 477 | 180 | 52.3 | 59.0 | 259 | 59.9 | 32.3 | 55 | 0.66 | |
| 5 x 406 | 5 | 1 | Mean | 36.4 | 12.1 | 246 | 2009 | 15 | 4.3 | 5.3 | 19.5 | 2192 | 604 | 233 | 62.0 | 63.9 | 330 | 75.6 | 43.0 | 57 | 0.86 | 15 |
| | | | s.d. | 4.3 | 1.0 | 17 | 0 | 0.0 | 0.2 | 0.1 | 3.5 | 21 | 6 | 28 | 4.7 | 3.7 | 29 | 6.0 | 4.6 | 6 | 0.04 | |
| | | 2 | Mean | 31.3 | 11.8 | 210 | 2014 | 15 | 4.6 | 5.3 | 14.3 | 2205 | 608 | 201 | 57.2 | 60.1 | 280 | 64.7 | 39.6 | 62 | 0.92 | 14 |
| | | | s.d. | 4.0 | 0.9 | 15 | 1 | 0.0 | 0.1 | 0.1 | 2.3 | 21 | 6 | 26 | 4.4 | 1.7 | 34 | 7.3 | 4.6 | 7 | 0.03 | |
| | | 3 | Mean | 35.3 | 11.9 | 243 | 2015 | 15 | 4.1 | 5.3 | 22.9 | 2191 | 604 | 227 | 62.1 | 63.7 | 325 | 74.5 | 39.8 | 54 | 0.82 | 17 |
| | | | s.d. | 5.1 | 1.1 | 18 | 1 | 0.4 | 0.5 | 0.1 | 9.8 | 26 | 7 | 33 | 5.8 | 4.2 | 37 | 7.7 | 6.5 | 10 | 0.11 | |
| Mean of Three Replications | | | | 34.3 | 11.9 | 233 | 2013 | 15 | 4.3 | 5.3 | 18.9 | 2196 | 605 | 220 | 60.5 | 62.6 | 312 | 71.6 | 40.8 | 58 | 0.87 | |
| 5 x 406 | 6 | 1 | Mean | 35.7 | 11.8 | 230 | 2011 | 14 | 4.8 | 6.3 | 25.0 | 2122 | 718 | 281 | 70.6 | 74.4 | 401 | 88.5 | 46.0 | 53 | 0.96 | 23 |
| | | | s.d. | 5.9 | 1.1 | 25 | 1 | 1.1 | 1.2 | 0.3 | 16.8 | 112 | 38 | 47 | 6.8 | 8.4 | 54 | 7.9 | 11.1 | 13 | 0.23 | |
| | | 2 | Mean | 38.0 | 12.1 | 248 | 2011 | 14 | 4.8 | 6.2 | 26.2 | 2098 | 709 | 299 | 73.6 | 76.5 | 415 | 90.6 | 48.0 | 54 | 0.93 | 19 |
| | | | s.d. | 5.3 | 1.0 | 18 | 0 | 0.7 | 1.1 | 0.3 | 15.3 | 100 | 34 | 42 | 5.6 | 8.4 | 54 | 8.2 | 10.6 | 13 | 0.21 | |
| | | 3 | Mean | 38.9 | 12.3 | 235 | 2011 | 14 | 4.6 | 6.2 | 25.7 | 2094 | 708 | 306 | 74.5 | 77.2 | 419 | 91.5 | 49.7 | 55 | 0.93 | 18 |
| | | | s.d. | 5.0 | 1.1 | 20 | 1 | 0.3 | 0.7 | 0.2 | 8.9 | 80 | 27 | 40 | 4.8 | 6.9 | 44 | 6.9 | 7.9 | 9 | 0.14 | |
| Mean of Three Replications | | | | 37.5 | 12.1 | 238 | 2011 | 14 | 4.7 | 6.2 | 25.6 | 2105 | 712 | 295 | 72.9 | 76.0 | 411 | 90.2 | 47.9 | 54 | 0.94 | |
| | | | | $H_A$ | $V_A$ | $D_P$ | $W_P$ | $\varepsilon_I$ | $v_a$ | $v_t$ | $S$ | $\omega_E$ | $\omega_T$ | $T_D$ | $T_{E\%}$ | $T_{F\%}$ | $T_F$ | $P_F$ | $P_D$ | $\eta_T$ | $A_{\phi T}$ | |

*Table A4.6 – Field results: steady state ploughing, 406mm furrow width (clay soil)*

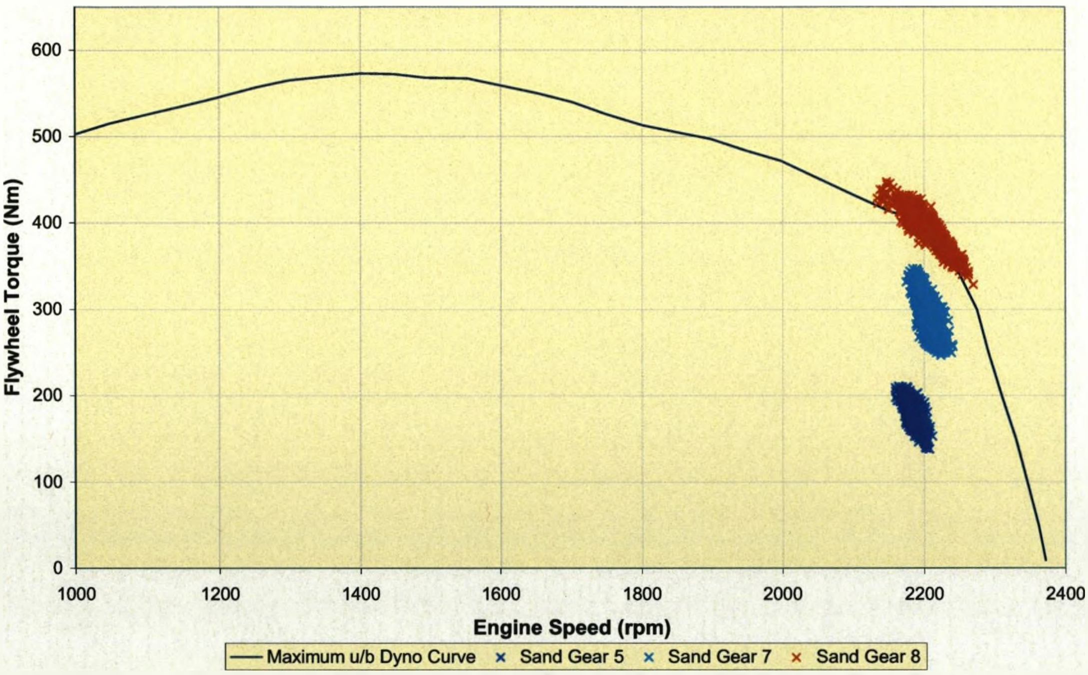


*Figure A4.1 – The effect of gear selection on dynamic loading whilst ploughing sandy soil (305mm furrow width)*

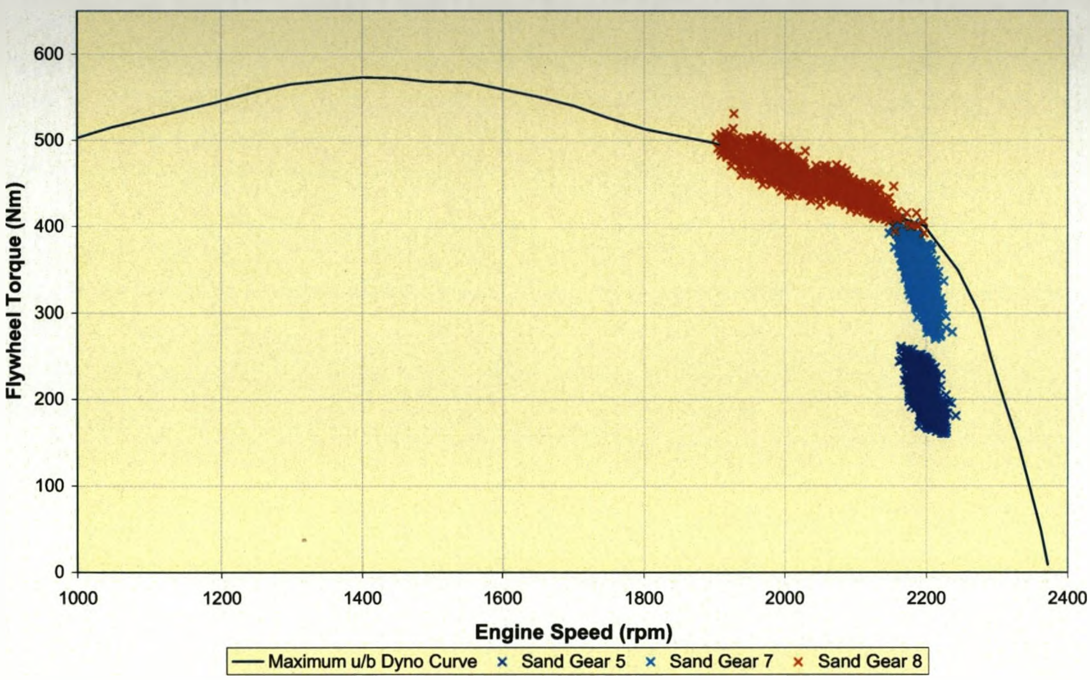


*Figure A4.2 – The effect of gear selection on dynamic loading whilst ploughing sandy soil (406mm furrow width)*

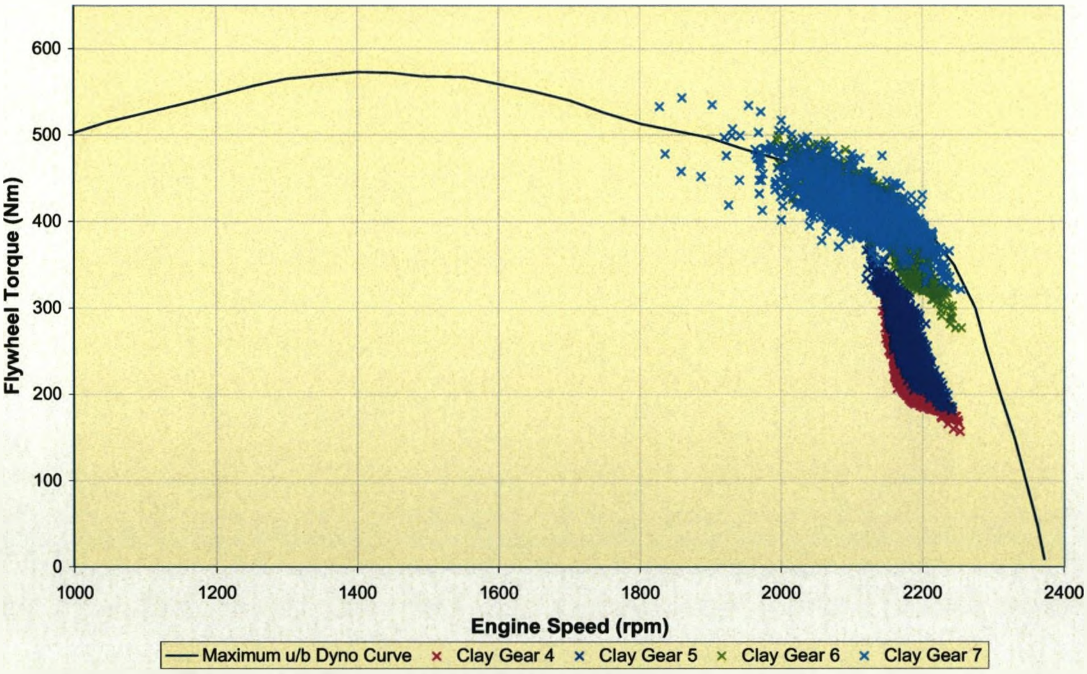


*Figure A4.3 – The effect of gear selection on dynamic loading whilst ploughing clay soil (305mm furrow width)*

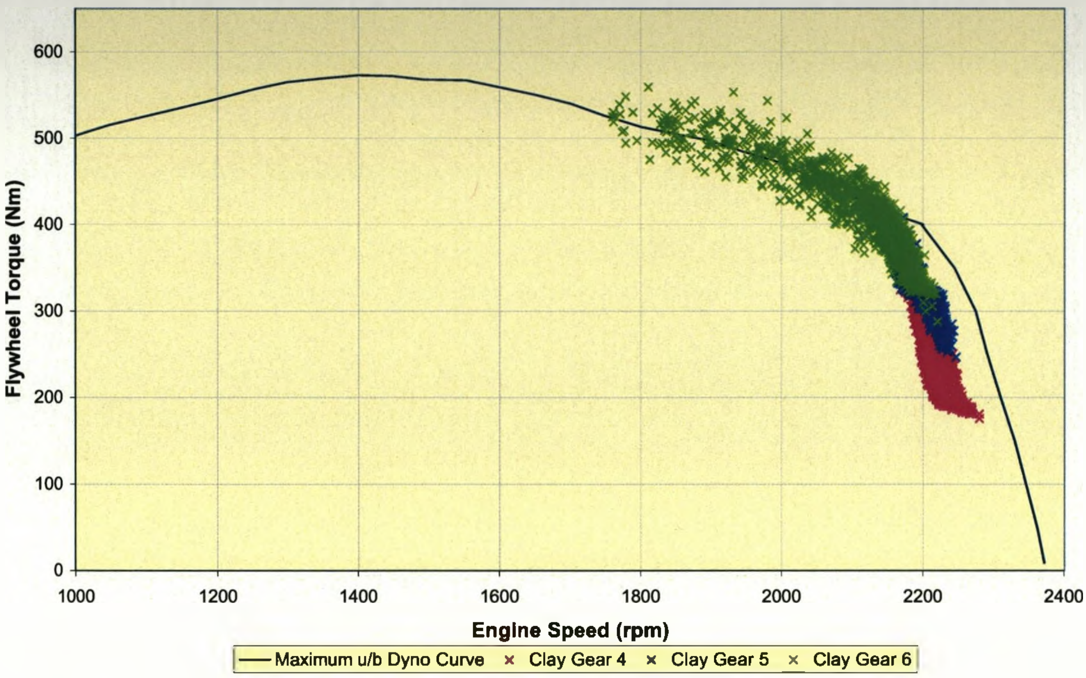


*Figure A4.4 – The effect of gear selection on dynamic loading whilst ploughing clay soil (406mm furrow width)*

## A4.2 Steady State Power Harrowing Data

| Target Depth (mm) | Gear | Rep | | Draught Force (kN) | Vertical Force (kN) | Harrow Depth (mm) | Rock shaft (%) | True Speed (km/h) | Wheel Speed (km/h) | Slip (%) | Engine Speed (rpm) | Trans Speed (rpm) | P.T.O. Speed (rpm) | P.T.O. Torque (Nm) | Engine Torque (%) | Flywheel Torque (%) | Flywheel Torque (Nm) | Flywheel Power (kW) | P.T.O. Power (kW) | Drawbar Power (kW) | Lost Power (kW) | Boost Status | Boost (%) | Torque Demand (%) | Engine Droop Mode | Theo. Area (ha/hr) | Run |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 75 | 4 | 1 | Mean | 12.9 | 0.6 | 80 | 3 | 3.4 | 3.6 | 5.9 | 1915 | 412 | 903 | 507 | 63.0 | 64.3 | 322 | 64.5 | 47.9 | 12.1 | 4.4 | 3.0 | 18.0 | 83.1 | 1 | 1.36 | 22 |
| | | | s.d. | 1.6 | 0.3 | 0 | 1.4 | 0.0 | 0.0 | 1.0 | 9 | 2 | 4 | 50 | 2.0 | 5.1 | 32 | 6.4 | 4.7 | 1.5 | 7.6 | 0.0 | 0.0 | 0.3 | 0 | 0.01 | |
| | | 2 | Mean | 14.2 | 0.7 | 81 | 3 | 3.4 | 3.6 | 7.0 | 1920 | 413 | 906 | 523 | 63.6 | 67.5 | 342 | 68.8 | 49.5 | 13.3 | 5.9 | 3.0 | 18.0 | 83.0 | 1 | 1.35 | 27 |
| | | | s.d. | 1.8 | 0.3 | 0 | 1.7 | 0.0 | 0.0 | 0.9 | 9 | 2 | 4 | 59 | 2.2 | 5.4 | 35 | 7.0 | 5.6 | 1.6 | 8.2 | 0.0 | 0.0 | 0.3 | 0 | 0.01 | |
| | | 3 | Mean | 11.6 | 0.5 | 79 | 2 | 3.5 | 3.6 | 4.5 | 1916 | 412 | 904 | 511 | 62.1 | 63.8 | 319 | 64.0 | 48.3 | 11.1 | 4.5 | 3.0 | 18.0 | 83.0 | 1 | 1.38 | 20 |
| | | | s.d. | 1.6 | 0.3 | 1 | 1.3 | 0.0 | 0.0 | 0.9 | 9 | 2 | 4 | 54 | 2.1 | 5.3 | 33 | 6.5 | 5.1 | 1.5 | 7.7 | 0.0 | 0.0 | 0.3 | 0 | 0.01 | |
| Mean of Three Replications | | | | 12.9 | 0.6 | 80 | 3 | 3.4 | 3.6 | 5.8 | 1917 | 412 | 904 | 513 | 62.9 | 65.2 | 328 | 65.7 | 48.6 | 12.2 | 5.0 | 3.0 | 18.0 | 83.0 | 1 | 1.36 | |
| 75 | 5 | 1 | Mean | 11.5 | 0.9 | 81 | 4 | 4.1 | 4.4 | 7.9 | 1818 | 502 | 858 | 519 | 67.9 | 73.4 | 381 | 72.5 | 46.6 | 13.0 | 12.9 | 4.0 | 18.0 | 103.0 | 1 | 1.62 | 25 |
| | | | s.d. | 1.7 | 0.3 | 0 | 1.5 | 0.0 | 0.0 | 1.0 | 10 | 3 | 5 | 38 | 1.8 | 4.6 | 30 | 5.6 | 3.3 | 1.9 | 6.6 | 0.0 | 0.0 | 0.2 | 0 | 0.02 | |
| | | 2 | Mean | 16.0 | 1.0 | 80 | 5 | 4.2 | 4.6 | 9.1 | 1905 | 526 | 898 | 546 | 71.8 | 76.2 | 399 | 79.5 | 51.4 | 18.6 | 9.5 | 3.0 | 18.0 | 83.3 | 1 | 1.68 | 23 |
| | | | s.d. | 1.7 | 0.3 | 0 | 1.6 | 0.0 | 0.0 | 1.1 | 10 | 3 | 5 | 60 | 2.3 | 5.1 | 34 | 6.6 | 5.5 | 1.9 | 8.3 | 0.0 | 0.0 | 0.5 | 0 | 0.02 | |
| | | 3 | Mean | 13.9 | 1.1 | 80 | 4 | 4.2 | 4.6 | 8.1 | 1908 | 527 | 900 | 511 | 69.3 | 73.0 | 378 | 75.6 | 48.1 | 16.4 | 11.1 | 3.0 | 18.0 | 83.2 | 1 | 1.70 | 21 |
| | | | s.d. | 1.5 | 0.3 | 0 | 1.3 | 0.0 | 0.0 | 0.8 | 9 | 2 | 4 | 56 | 2.0 | 5.3 | 35 | 6.8 | 5.2 | 1.8 | 8.1 | 0.0 | 0.0 | 0.4 | 0 | 0.01 | |
| Mean of Three Replications | | | | 13.8 | 1.0 | 80 | 4 | 4.2 | 4.5 | 8.4 | 1877 | 518 | 885 | 525 | 69.7 | 74.2 | 386 | 75.9 | 48.7 | 16.0 | 11.2 | 3.3 | 18.0 | 89.9 | 1 | 1.67 | |
| 75 | 6 | 1 | Mean | 16.3 | 1.7 | 79 | 5 | 5.0 | 5.4 | 8.5 | 1831 | 620 | 864 | 522 | 75.2 | 80.0 | 424 | 81.3 | 47.2 | 22.5 | 11.6 | 4.0 | 18.0 | 102.9 | 1 | 1.99 | 19 |
| | | | s.d. | 1.7 | 0.3 | 0 | 1.8 | 0.1 | 0.0 | 1.6 | 13 | 4 | 6 | 41 | 3.1 | 5.1 | 33 | 6.1 | 3.6 | 2.4 | 6.8 | 0.0 | 0.0 | 0.3 | 0 | 0.03 | |
| | | 2 | Mean | 16.9 | 2.0 | 81 | 6 | 5.1 | 5.6 | 9.3 | 1894 | 641 | 893 | 545 | 80.3 | 84.6 | 454 | 90.1 | 51.0 | 23.9 | 15.2 | 4.0 | 18.0 | 101.7 | 1 | 2.04 | 24 |
| | | | s.d. | 1.8 | 0.4 | 0 | 1.6 | 0.0 | 0.0 | 1.0 | 12 | 4 | 6 | 55 | 2.8 | 4.4 | 29 | 5.6 | 5.0 | 2.6 | 7.5 | 0.0 | 0.0 | 0.5 | 0 | 0.02 | |
| | | 3 | Mean | 17.8 | 2.0 | 81 | 5 | 5.1 | 5.6 | 9.5 | 1897 | 642 | 895 | 559 | 79.5 | 85.6 | 461 | 91.5 | 52.4 | 25.1 | 14.0 | 3.2 | 18.0 | 86.3 | 1 | 2.04 | 26 |
| | | | s.d. | 1.9 | 0.4 | 0 | 1.7 | 0.1 | 0.1 | 1.1 | 16 | 5 | 7 | 58 | 2.9 | 4.6 | 30 | 5.7 | 5.3 | 2.7 | 7.6 | 0.4 | 0.0 | 6.5 | 0 | 0.02 | |
| Mean of Three Replications | | | | 17.0 | 1.9 | 80 | 5 | 5.1 | 5.5 | 9.1 | 1874 | 634 | 884 | 542 | 78.3 | 83.4 | 446 | 87.6 | 50.2 | 23.9 | 13.6 | 3.7 | 18.0 | 97.0 | 1 | 2.02 | |
| | | | | $H_A$ | $V_A$ | $D_{HA}$ | $\varepsilon_r$ | $v_a$ | $v_t$ | S | $\omega_E$ | $\omega_T$ | $\omega_P$ | $T_{BH}$ | $T_{E\%}$ | $T_{F\%}$ | $T_F$ | $P_F$ | $P_P$ | $P_D$ | $P_L$ | $B_S$ | $B_\%$ | $T_{DV}$ | $\delta_I$ | $A_{HT}$ | |

*Table A4.7 – Field results: steady state power harrowing, 75mm depth (sandy soil)*

| Target Depth (mm) | Gear | Rep | | Draught Force (kN) | Vertical Force (kN) | Harrow Depth (mm) | Rock shaft (%) | True Speed (km/h) | Wheel Speed (km/h) | Slip (%) | Engine Speed (rpm) | Trans Speed (rpm) | P.T.O. Speed (rpm) | P.T.O. Torque (Nm) | Engine Torque (%) | Flywheel Torque (%) | Flywheel Torque (Nm) | Flywheel Power (kW) | P.T.O. Power (kW) | Drawbar Power (kW) | Lost Power (kW) | Boost Status | Boost (%) | Torque Demand (%) | Engine Droop Mode | Theo. Area (ha/hr) | Run |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 100 | 4 | 1 | Mean | 8.4 | 6.5 | 104 | 10 | 3.6 | 3.6 | 1.3 | 1908 | 411 | 900 | 384 | 50.6 | 52.8 | 262 | 52.4 | 36.1 | 8.3 | 7.9 | 3.0 | 18.0 | 83.2 | 1 | 1.42 | 6 |
| | | | s.d. | 1.3 | 0.7 | 0 | 0.0 | 0.0 | 0.0 | 1.0 | 12 | 3 | 6 | 53 | 3.9 | 6.2 | 26 | 5.0 | 4.9 | 1.2 | 5.6 | 0.0 | 0.0 | 0.5 | 0 | 0.02 | |
| | | 2 | Mean | 9.7 | 5.6 | 104 | 10 | 3.5 | 3.6 | 2.4 | 1904 | 410 | 898 | 435 | 56.6 | 62.8 | 313 | 62.4 | 40.9 | 9.5 | 12.1 | 3.0 | 18.0 | 83.3 | 1 | 1.40 | 2 |
| | | | s.d. | 1.3 | 0.6 | 0 | 0.0 | 0.0 | 0.0 | 1.1 | 11 | 2 | 5 | 50 | 3.2 | 6.2 | 37 | 7.3 | 4.6 | 1.3 | 7.0 | 0.0 | 0.0 | 0.5 | 0 | 0.02 | |
| | | 3 | Mean | 8.6 | 6.6 | 105 | 10 | 3.5 | 3.6 | 1.4 | 1899 | 409 | 896 | 372 | 48.9 | 51.9 | 259 | 51.5 | 34.9 | 8.4 | 8.3 | 3.0 | 18.0 | 83.5 | 1 | 1.41 | 7 |
| | | | s.d. | 1.0 | 0.6 | 0 | 0.0 | 0.0 | 0.0 | 1.0 | 12 | 3 | 6 | 50 | 3.1 | 6.2 | 25 | 4.9 | 4.6 | 0.9 | 5.3 | 0.0 | 0.0 | 0.5 | 0 | 0.01 | |
| Mean of Three Replications | | | | 8.9 | 6.2 | 104 | 10 | 3.5 | 3.6 | 1.7 | 1904 | 410 | 898 | 397 | 52.1 | 55.8 | 278 | 55.4 | 37.3 | 8.7 | 9.4 | 3.0 | 18.0 | 83.4 | 1 | 1.41 | |
| 100 | 5 | 1 | Mean | 12.0 | 5.9 | 104 | 10 | 4.5 | 4.6 | 2.8 | 1894 | 523 | 893 | 462 | 63.6 | 68.3 | 348 | 69.0 | 43.2 | 14.8 | 10.9 | 3.0 | 18.0 | 83.7 | 1 | 1.78 | 4 |
| | | | s.d. | 1.8 | 0.9 | 0 | 0.0 | 0.1 | 0.0 | 1.3 | 14 | 4 | 7 | 64 | 4.3 | 6.6 | 43 | 8.3 | 5.9 | 2.2 | 7.3 | 0.0 | 0.0 | 0.5 | 0 | 0.02 | |
| | | 2 | Mean | 11.8 | 5.6 | 104 | 10 | 4.4 | 4.6 | 3.4 | 1887 | 521 | 890 | 471 | 66.2 | 73.0 | 378 | 74.7 | 43.9 | 14.5 | 16.3 | 3.0 | 18.0 | 83.8 | 1 | 1.76 | 1 |
| | | | s.d. | 1.5 | 0.8 | 0 | 0.0 | 0.1 | 0.0 | 1.4 | 12 | 3 | 6 | 52 | 4.0 | 6.5 | 43 | 8.2 | 4.8 | 1.6 | 7.5 | 0.0 | 0.0 | 0.4 | 0 | 0.03 | |
| | | 3 | Mean | 12.3 | 5.8 | 104 | 10 | 4.4 | 4.6 | 3.5 | 1894 | 523 | 893 | 481 | 64.2 | 70.7 | 363 | 72.0 | 44.9 | 15.1 | 12.0 | 3.0 | 18.0 | 83.6 | 1 | 1.77 | 3 |
| | | | s.d. | 1.7 | 0.7 | 0 | 0.0 | 0.1 | 0.0 | 1.3 | 14 | 4 | 7 | 56 | 4.0 | 6.5 | 42 | 8.1 | 5.1 | 2.0 | 7.7 | 0.0 | 0.0 | 0.5 | 0 | 0.02 | |
| Mean of Three Replications | | | | 12.0 | 5.8 | 104 | 10 | 4.4 | 4.6 | 3.2 | 1891 | 522 | 892 | 471 | 64.7 | 70.7 | 363 | 71.9 | 44.0 | 14.8 | 13.1 | 3.0 | 18.0 | 83.7 | 1 | 1.77 | |
| 100 | 6 | 1 | Mean | 15.1 | 5.3 | 104 | 10 | 5.4 | 5.7 | 4.9 | 1912 | 647 | 902 | 527 | 74.9 | 84.8 | 455 | 91.0 | 49.7 | 22.7 | 18.7 | 3.0 | 18.0 | 83.1 | 1 | 2.18 | 8 |
| | | | s.d. | 2.0 | 1.3 | 0 | 0.2 | 0.2 | 0.1 | 2.1 | 36 | 12 | 17 | 68 | 5.8 | 6.5 | 43 | 7.6 | 6.0 | 2.8 | 7.4 | 0.0 | 0.0 | 1.0 | 0 | 0.07 | |
| | | 2 | Mean | 17.4 | 3.9 | 104 | 8 | 5.2 | 5.6 | 8.3 | 1897 | 642 | 895 | 608 | 83.6 | 92.0 | 502 | 99.8 | 56.9 | 24.9 | 18.0 | 4.0 | 18.0 | 101.6 | 1 | 2.07 | 9 |
| | | | s.d. | 2.2 | 1.1 | 0 | 0.5 | 0.1 | 0.0 | 2.4 | 15 | 5 | 7 | 58 | 4.4 | 5.7 | 37 | 7.0 | 5.3 | 2.8 | 7.5 | 0.0 | 0.0 | 0.5 | 0 | 0.06 | |
| | | 3 | Mean | 15.0 | 5.7 | 104 | 10 | 5.4 | 5.7 | 5.6 | 1928 | 653 | 910 | 513 | 75.7 | 83.4 | 446 | 90.0 | 48.9 | 22.5 | 18.7 | 3.0 | 18.0 | 82.6 | 1 | 2.18 | 5 |
| | | | s.d. | 1.9 | 0.9 | 0 | 0.1 | 0.1 | 0.1 | 1.6 | 24 | 8 | 11 | 58 | 4.4 | 6.0 | 39 | 7.3 | 5.3 | 2.7 | 7.5 | 0.0 | 0.0 | 0.7 | 0 | 0.04 | |
| Mean of Three Replications | | | | 15.8 | 5.0 | 104 | 9 | 5.3 | 5.7 | 6.3 | 1913 | 647 | 902 | 549 | 78.1 | 86.7 | 468 | 93.6 | 51.8 | 23.3 | 18.5 | 3.3 | 18.0 | 89.1 | 1 | 2.13 | |
| | | | | $H_A$ | $V_A$ | $D_{HA}$ | $\varepsilon_r$ | $v_a$ | $v_t$ | S | $\omega_E$ | $\omega_T$ | $\omega_P$ | $T_{PH}$ | $T_{E\%}$ | $T_{F\%}$ | $T_F$ | $P_F$ | $P_P$ | $P_D$ | $P_L$ | $B_S$ | $B_\%$ | $T_{DV}$ | $\delta_I$ | $A_{HT}$ | |

*Table A4.8 – Field results: steady state power harrowing, 100mm working depth (sandy soil)*

| Target Depth (mm) | Gear | Rep | | Draught Force (kN) | Vertical Force (kN) | Harrow Depth (mm) | Rock shaft (%) | True Speed (km/h) | Wheel Speed (km/h) | Slip (%) | Engine Speed (rpm) | Trans Speed (rpm) | P.T.O. Speed (rpm) | P.T.O. Torque (Nm) | Engine Torque (%) | Flywheel Torque (%) | Flywheel Torque (Nm) | Flywheel Power (kW) | P.T.O. Power (kW) | Drawbar Power (kW) | Lost Power (kW) | Boost Status | Boost (%) | Torque Demand (%) | Engine Droop Mode | Theo. Area (ha/hr) | Run |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 125 | 4 | 1 | Mean | 14.9 | 3.3 | 126 | 8 | 3.3 | 3.6 | 7.3 | 1896 | 408 | 895 | 687 | 77.5 | 81.4 | 433 | 85.9 | 64.3 | 13.7 | 7.9 | 4.0 | 18.0 | 101.6 | 1 | 1.33 | 17 |
| | | | s.d. | 2.5 | 1.2 | 0 | 0.4 | 0.1 | 0.0 | 2.0 | 15 | 3 | 7 | 69 | 4.9 | 6.3 | 41 | 7.9 | 6.2 | 2.2 | 7.7 | 0.0 | 0.0 | 0.5 | 0 | 0.03 | |
| | | 2 | Mean | 13.2 | 4.9 | 126 | 8 | 3.4 | 3.6 | 4.2 | 1905 | 410 | 899 | 632 | 70.9 | 74.8 | 390 | 77.8 | 59.5 | 12.6 | 5.7 | 4.0 | 18.0 | 101.4 | 1 | 1.38 | 15 |
| | | | s.d. | 2.2 | 1.1 | 0 | 0.1 | 0.1 | 0.0 | 1.5 | 16 | 3 | 7 | 69 | 4.7 | 6.7 | 44 | 8.4 | 6.4 | 2.0 | 7.8 | 0.0 | 0.0 | 0.6 | 0 | 0.02 | |
| | | 3 | Mean | 13.2 | 5.3 | 126 | 8 | 3.5 | 3.6 | 4.4 | 1932 | 416 | 911 | 626 | 70.0 | 74.3 | 387 | 78.2 | 59.7 | 12.7 | 5.7 | 4.0 | 18.0 | 100.6 | 1 | 1.39 | 12 |
| | | | s.d. | 1.9 | 0.6 | 0 | 0.0 | 0.0 | 0.0 | 1.3 | 13 | 3 | 6 | 60 | 3.4 | 5.6 | 37 | 7.2 | 5.7 | 1.7 | 8.0 | 0.0 | 0.0 | 0.5 | 0 | 0.02 | |
| Mean of Three Replications | | | | 13.7 | 4.5 | 126 | 8 | 3.4 | 3.6 | 5.3 | 1911 | 411 | 901 | 648 | 72.8 | 76.8 | 403 | 80.6 | 61.2 | 13.0 | 6.4 | 4.0 | 18.0 | 101.2 | 1 | 1.37 | |
| 125 | 5 | 1 | Mean | 17.5 | 4.1 | 126 | 8 | 4.2 | 4.6 | 6.1 | 1894 | 523 | 893 | 728 | 86.5 | 93.4 | 511 | 101.4 | 68.0 | 20.4 | 12.9 | 4.0 | 18.0 | 101.7 | 1 | 1.68 | 11 |
| | | | s.d. | 2.4 | 1.3 | 0 | 0.7 | 0.1 | 0.0 | 2.3 | 16 | 5 | 8 | 69 | 5.3 | 6.5 | 43 | 8.1 | 6.2 | 2.6 | 8.1 | 0.0 | 0.0 | 0.5 | 0 | 0.04 | |
| | | 2 | Mean | 18.3 | 2.4 | 126 | 8 | 4.1 | 4.6 | 10.8 | 1904 | 525 | 898 | 752 | 91.1 | 94.2 | 517 | 103.0 | 70.7 | 20.9 | 11.3 | 4.0 | 18.0 | 101.4 | 1 | 1.64 | 13 |
| | | | s.d. | 1.9 | 0.8 | 0 | 0.8 | 0.1 | 0.0 | 1.9 | 16 | 4 | 8 | 64 | 3.6 | 5.1 | 33 | 6.4 | 5.9 | 2.2 | 8.4 | 0.0 | 0.0 | 0.5 | 0 | 0.04 | |
| | | 3 | Mean | 16.7 | 4.3 | 126 | 8 | 4.3 | 4.6 | 7.1 | 1910 | 527 | 901 | 701 | 84.3 | 88.4 | 479 | 95.8 | 66.1 | 19.9 | 9.8 | 4.0 | 18.0 | 101.2 | 1 | 1.72 | 18 |
| | | | s.d. | 2.8 | 1.4 | 0 | 0.6 | 0.1 | 0.0 | 2.3 | 18 | 5 | 8 | 83 | 6.3 | 7.2 | 47 | 9.0 | 7.6 | 3.1 | 8.6 | 0.0 | 0.0 | 0.6 | 0 | 0.05 | |
| Mean of Three Replications | | | | 17.5 | 3.6 | 126 | 8 | 4.2 | 4.6 | 8.0 | 1903 | 525 | 897 | 727 | 87.3 | 92.0 | 502 | 100.0 | 68.3 | 20.4 | 11.4 | 4.0 | 18.0 | 101.4 | 1 | 1.68 | |
| 125 | 6 | 1 | Mean | 19.6 | 3.1 | 126 | 8 | 4.7 | 5.2 | 10.7 | 1764 | 597 | 832 | 744 | 97.0 | 104.4 | 584 | 107.7 | 64.7 | 25.4 | 17.6 | 4.0 | 18.0 | 103.9 | 1 | 1.87 | 10 |
| | | | s.d. | 1.9 | 1.0 | 0 | 1.0 | 0.3 | 0.3 | 2.7 | 94 | 32 | 44 | 53 | 3.3 | 4.6 | 30 | 5.4 | 4.6 | 3.0 | 6.5 | 0.0 | 0.0 | 1.4 | 0 | 0.13 | |
| | | 2 | Mean | 19.7 | 3.3 | 126 | 8 | 4.8 | 5.4 | 11.2 | 1823 | 617 | 860 | 731 | 97.0 | 99.8 | 554 | 105.6 | 65.8 | 26.3 | 13.5 | 4.0 | 18.0 | 103.0 | 1 | 1.92 | 16 |
| | | | s.d. | 1.7 | 0.8 | 0 | 0.5 | 0.2 | 0.2 | 1.9 | 64 | 22 | 30 | 53 | 2.4 | 4.5 | 30 | 5.4 | 4.5 | 2.5 | 7.0 | 0.0 | 0.0 | 1.3 | 0 | 0.08 | |
| | | 3 | Mean | 19.6 | 2.6 | 125 | 9 | 4.8 | 5.4 | 11.8 | 1817 | 615 | 857 | 719 | 96.8 | 99.7 | 553 | 105.2 | 64.5 | 25.8 | 14.9 | 4.0 | 18.0 | 103.1 | 1 | 1.90 | 14 |
| | | | s.d. | 2.3 | 0.9 | 0 | 0.9 | 0.2 | 0.2 | 2.1 | 66 | 22 | 31 | 50 | 2.4 | 4.4 | 29 | 5.6 | 4.7 | 3.2 | 7.3 | 0.0 | 0.0 | 1.1 | 0 | 0.08 | |
| Mean of Three Replications | | | | 19.6 | 3.0 | 126 | 8 | 4.7 | 5.3 | 11.2 | 1801 | 610 | 850 | 731 | 96.9 | 101.3 | 564 | 106.1 | 65.0 | 25.8 | 15.3 | 4.0 | 18.0 | 103.3 | 1 | 1.90 | |
| | | | | $H_A$ | $V_A$ | $D_{HA}$ | $\varepsilon$ | $v_a$ | $v_t$ | $s$ | $\omega_E$ | $\omega_T$ | $\omega_P$ | $T_{PH}$ | $T_{ES}$ | $T_{FS}$ | $T_F$ | $P_F$ | $P_P$ | $P_D$ | $P_L$ | $B_S$ | $B_\%$ | $T_{DV}$ | $\delta_I$ | $A_{HT}$ | |

*Table A4.9 – Field results: steady state power harrowing, 125mm working depth (sandy soil)*

| Target Depth (mm) | Gear | Rep | | Draught Force (kN) | Vertical Force (kN) | Harrow Depth (mm) | Rock shaft (%) | True Speed (km/h) | Wheel Speed (km/h) | Slip (%) | Engine Speed (rpm) | Trans Speed (rpm) | P.T.O. Speed (rpm) | P.T.O. Torque (Nm) | P.T.O. Tq (Veh) (Nm) | Engine Torque (%) | Flywheel Torque (%) | Flywheel Torque (Nm) | Flywheel Power (kW) | P.T.O. Power (kW) | Drawbar Power (kW) | Lost Power (kW) | Boost Status | Boost (%) | Torque Demand (%) | Engine Droop | Theo. Area (ha/hr) | Run |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 75 | 4 | 1 | Mean | 7.9 | -1.1 | 78 | 15 | 3.5 | 3.6 | 3.0 | 1917 | 412 | 904 | 219 | 157 | 41.5 | 58.2 | 205 | 41.1 | 20.7 | 7.6 | 12.7 | 4.0 | 15.3 | 98.3 | 1 | 1.40 | 16 |
| | | | s.d. | 1.8 | 0.2 | 1 | 2.3 | 0.0 | 0.0 | 1.0 | 13 | 3 | 6 | 32 | 11 | 2.3 | 0.8 | 13 | 2.3 | 3.0 | 1.7 | 3.3 | 0.0 | 0.5 | 0.7 | 0 | 0.01 | |
| | | 2 | Mean | 8.0 | -1.1 | 74 | 16 | 3.5 | 3.6 | 3.6 | 1908 | 410 | 900 | 219 | 155 | 41.3 | 56.1 | 203 | 40.6 | 20.7 | 7.7 | 12.2 | 4.0 | 15.6 | 98.8 | 1 | 1.39 | 12 |
| | | | s.d. | 1.5 | 0.2 | 0 | 2.6 | 0.0 | 0.0 | 1.2 | 12 | 2 | 6 | 34 | 14 | 2.2 | 0.8 | 13 | 2.3 | 3.2 | 1.5 | 3.1 | 0.0 | 0.7 | 0.8 | 0 | 0.02 | |
| | | 3 | Mean | 7.4 | -1.1 | 78 | 15 | 3.5 | 3.6 | 4.2 | 1914 | 412 | 903 | 214 | 157 | 41.6 | 56.1 | 203 | 40.8 | 20.3 | 7.1 | 13.4 | 4.0 | 15.7 | 98.8 | 1 | 1.38 | 15 |
| | | | s.d. | 1.5 | 0.2 | 0 | 2.2 | 0.0 | 0.0 | 1.0 | 11 | 3 | 5 | 30 | 10 | 1.8 | 0.8 | 12 | 2.2 | 2.8 | 1.4 | 3.0 | 0.0 | 0.5 | 0.6 | 0 | 0.01 | |
| Mean of Three Replications | | | | 7.8 | -1.1 | 76 | 15 | 3.5 | 3.6 | 3.6 | 1913 | 411 | 902 | 218 | 156 | 41.5 | 56.2 | 204 | 40.8 | 20.6 | 7.5 | 12.7 | 4.0 | 15.5 | 98.6 | 1 | 1.39 | |
| 75 | 5 | 1 | Mean | 9.3 | -0.9 | 81 | 15 | 4.4 | 4.6 | 5.2 | 1917 | 528 | 904 | 270 | 181 | 49.6 | 59.6 | 268 | 53.7 | 25.6 | 11.3 | 16.8 | 4.0 | 18.0 | 101.0 | 2 | 1.76 | 11 |
| | | | s.d. | 1.7 | 0.2 | 0 | 2.0 | 0.1 | 0.0 | 1.2 | 13 | 4 | 6 | 41 | 14 | 2.4 | 1.1 | 25 | 4.7 | 3.8 | 2.1 | 4.3 | 0.0 | 0.2 | 0.5 | 0 | 0.02 | |
| | | 2 | Mean | 8.6 | -1.0 | 79 | 16 | 4.4 | 4.6 | 5.2 | 1905 | 525 | 899 | 250 | 178 | 50.3 | 59.3 | 262 | 52.2 | 23.5 | 10.5 | 18.2 | 4.0 | 18.0 | 101.3 | 2 | 1.75 | 18 |
| | | | s.d. | 1.7 | 0.2 | 0 | 2.4 | 0.0 | 0.0 | 1.1 | 12 | 3 | 6 | 35 | 13 | 2.2 | 0.9 | 21 | 3.9 | 3.2 | 2.0 | 3.9 | 0.0 | 0.0 | 0.5 | 0 | 0.02 | |
| | | 3 | Mean | 8.8 | -1.0 | 78 | 17 | 4.4 | 4.6 | 4.9 | 1894 | 522 | 894 | 254 | 176 | 48.5 | 59.2 | 260 | 51.5 | 23.8 | 10.6 | 17.0 | 4.0 | 18.0 | 101.6 | 2 | 1.74 | 13 |
| | | | s.d. | 1.6 | 0.2 | 0 | 2.1 | 0.1 | 0.0 | 1.1 | 13 | 4 | 6 | 34 | 13 | 2.2 | 1.0 | 22 | 4.1 | 3.2 | 2.0 | 4.1 | 0.0 | 0.2 | 0.5 | 0 | 0.02 | |
| Mean of Three Replications | | | | 8.9 | -1.0 | 79 | 16 | 4.4 | 4.6 | 5.1 | 1906 | 525 | 899 | 258 | 178 | 49.4 | 59.4 | 263 | 52.5 | 24.3 | 10.8 | 17.4 | 4.0 | 18.0 | 101.3 | 2 | 1.75 | |
| 75 | 6 | 1 | Mean | 11.2 | -0.9 | 78 | 17 | 5.3 | 5.7 | 6.9 | 1915 | 648 | 903 | 333 | 224 | 60.8 | 66.4 | 350 | 70.1 | 31.5 | 16.4 | 22.2 | 4.0 | 18.0 | 101.1 | 2 | 2.11 | 17 |
| | | | s.d. | 2.1 | 0.2 | 0 | 2.5 | 0.1 | 0.0 | 1.1 | 14 | 5 | 7 | 41 | 14 | 2.6 | 3.0 | 19 | 3.5 | 3.8 | 3.1 | 5.0 | 0.0 | 0.0 | 0.5 | 0 | 0.03 | |
| | | 2 | Mean | 11.0 | -0.9 | 79 | 17 | 5.2 | 5.6 | 7.2 | 1902 | 643 | 897 | 327 | 211 | 59.5 | 65.6 | 344 | 68.5 | 30.7 | 15.9 | 21.9 | 4.0 | 18.0 | 101.5 | 2 | 2.09 | 10 |
| | | | s.d. | 2.1 | 0.2 | 0 | 2.9 | 0.1 | 0.1 | 1.3 | 16 | 5 | 8 | 47 | 17 | 3.2 | 3.1 | 22 | 3.9 | 4.3 | 3.0 | 5.2 | 0.0 | 0.0 | 0.5 | 0 | 0.03 | |
| | | 3 | Mean | 10.8 | -0.9 | 77 | 17 | 5.3 | 5.7 | 6.3 | 1909 | 646 | 901 | 303 | 206 | 58.4 | 64.4 | 336 | 67.2 | 28.6 | 15.8 | 22.7 | 4.0 | 18.0 | 101.2 | 2 | 2.12 | 14 |
| | | | s.d. | 1.9 | 0.2 | 0 | 2.7 | 0.1 | 0.0 | 1.3 | 14 | 5 | 7 | 41 | 16 | 2.7 | 2.5 | 17 | 3.0 | 3.8 | 2.8 | 4.6 | 0.0 | 0.0 | 0.5 | 0 | 0.03 | |
| Mean of Three Replications | | | | 11.0 | -0.9 | 78 | 17 | 5.3 | 5.7 | 6.8 | 1909 | 645 | 900 | 321 | 214 | 59.5 | 65.5 | 343 | 68.6 | 30.3 | 16.1 | 22.3 | 4.0 | 18.0 | 101.3 | 2 | 2.11 | |
| | | | | $H_A$ | $V_A$ | $D_{HA}$ | $\epsilon_i$ | $v_a$ | $v_t$ | $S$ | $\omega_E$ | $\omega_T$ | $\omega_P$ | $T_{SN}$ | $T_{PTO}$ | $T_{E\%}$ | $T_{F\%}$ | $T_F$ | $P_F$ | $P_P$ | $P_D$ | $P_L$ | $B_C$ | $B_\%$ | $T_{D\%}$ | $\delta_I$ | $A_{NT}$ | |

*Table A4.10 – Field results: steady state power harrowing, 75mm working depth (clay soil)*

| Target Depth (mm) | Gear | Rep | | Draught Force (kN) | Vertical Force (kN) | Harrow Depth (mm) | Rock shaft (%) | True Speed (km/h) | Wheel Speed (km/h) | Slip (%) | Engine Speed (rpm) | Trans Speed (rpm) | P.T.O. Speed (rpm) | P.T.O. Torque (Nm) | P.T.O. Tq (Veh) (Nm) | Engine Torque (%) | Flywheel Torque (%) | Flywheel Torque (Nm) | Flywheel Power (kW) | P.T.O. Power (kW) | Drawbar Power (kW) | Lost Power (kW) | Boost Status | Boost (%) | Torque Demand (%) | Engine Droop | Theo. Area (ha/h) | Run |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 100 | 4 | 1 | Mean | 9.1 | -1.0 | 106 | 14 | 4.2 | 3.6 | 3.9 | 1875 | 517 | 884 | 392 | 252 | 61.8 | 65.7 | 345 | 67.6 | 36.3 | 13.9 | 17.4 | 4.0 | 17.7 | 101.7 | 2 | 1.38 | 2 |
| | | | s.d. | 1.7 | 0.2 | 0 | 3.0 | 0.1 | 0.0 | 1.1 | 17 | 5 | 8 | 53 | 20 | 3.6 | 3.4 | 24 | 4.2 | 4.8 | 2.6 | 4.9 | 0.0 | 0.6 | 0.6 | 0 | 0.01 | |
| | | 2 | Mean | 9.0 | -1.0 | 103 | 13 | 3.4 | 3.6 | 4.6 | 1908 | 410 | 900 | 340 | 173 | 47.7 | 59.2 | 259 | 51.6 | 32.0 | 8.6 | 11.2 | 4.0 | 17.9 | 101.1 | 1 | 1.37 | 3 |
| | | | s.d. | 1.6 | 0.2 | 0 | 1.9 | 0.0 | 0.0 | 1.1 | 12 | 3 | 6 | 34 | 13 | 2.1 | 0.9 | 21 | 3.9 | 3.2 | 1.5 | 3.8 | 0.0 | 0.4 | 0.6 | 0 | 0.01 | |
| | | 3 | Mean | 8.6 | -1.0 | 106 | 12 | 3.5 | 3.6 | 3.6 | 1914 | 411 | 903 | 358 | 184 | 48.2 | 58.4 | 243 | 48.8 | 33.9 | 8.3 | 6.6 | 4.0 | 15.8 | 98.9 | 1 | 1.39 | 8 |
| | | | s.d. | 1.7 | 0.2 | 0 | 2.6 | 0.0 | 0.0 | 1.4 | 12 | 3 | 6 | 40 | 15 | 2.7 | 1.1 | 23 | 4.5 | 3.7 | 1.6 | 3.6 | 0.0 | 1.0 | 1.1 | 0 | 0.02 | |
| Mean of Three Replications | | | | 8.9 | -1.0 | 105 | 13 | 3.7 | 3.6 | 4.0 | 1899 | 446 | 896 | 363 | 207 | 52.6 | 61.1 | 282 | 56.1 | 34.1 | 10.3 | 11.7 | 4.0 | 17.1 | 100.6 | 1 | 1.38 | |
| 100 | 5 | 1 | Mean | 11.9 | -1.2 | 105 | 26 | 4.1 | 4.5 | 7.3 | 1897 | 508 | 895 | 301 | 209 | 54.2 | 60.5 | 290 | 57.2 | 27.8 | 11.2 | 18.1 | 3.8 | 17.8 | 98.0 | 2 | 1.68 | 4 |
| | | | s.d. | 2.2 | 0.3 | 1 | 23.8 | 0.6 | 0.0 | 1.7 | 46 | 69 | 22 | 171 | 82 | 14.8 | 11.1 | 110 | 21.2 | 15.8 | 4.7 | 7.8 | 0.4 | 0.5 | 7.9 | 0 | 0.01 | |
| | | 2 | Mean | 11.3 | -1.0 | 105 | 14 | 4.3 | 4.6 | 6.4 | 1921 | 529 | 906 | 460 | 238 | 60.9 | 66.7 | 352 | 70.7 | 43.6 | 13.6 | 13.6 | 4.0 | 18.0 | 100.9 | 2 | 1.74 | 6 |
| | | | s.d. | 2.0 | 0.2 | 0 | 1.9 | 0.1 | 0.0 | 1.6 | 15 | 4 | 7 | 43 | 14 | 2.5 | 2.7 | 18 | 3.2 | 3.9 | 2.4 | 4.7 | 0.0 | 0.0 | 0.5 | 0 | 0.03 | |
| | | 3 | Mean | 11.0 | -1.0 | 105 | 14 | 4.3 | 4.6 | 6.7 | 1920 | 529 | 905 | 462 | 247 | 61.8 | 65.5 | 343 | 69.0 | 43.8 | 13.2 | 11.9 | 4.0 | 17.9 | 100.8 | 2 | 1.73 | 7 |
| | | | s.d. | 2.1 | 0.2 | 0 | 2.4 | 0.1 | 0.0 | 1.6 | 16 | 4 | 8 | 51 | 19 | 3.4 | 3.1 | 21 | 3.8 | 4.7 | 2.5 | 4.8 | 0.0 | 0.3 | 0.6 | 0 | 0.03 | |
| Mean of Three Replications | | | | 11.4 | -1.1 | 105 | 18 | 4.3 | 4.6 | 6.8 | 1913 | 522 | 902 | 408 | 232 | 59.0 | 64.2 | 328 | 65.6 | 38.4 | 12.7 | 14.5 | 3.9 | 17.9 | 99.9 | 2 | 1.72 | |
| 100 | 6 | 1 | Mean | 12.9 | -1.0 | 106 | 14 | 5.1 | 5.6 | 9.1 | 1892 | 640 | 893 | 459 | 256 | 66.9 | 72.3 | 388 | 76.8 | 42.9 | 18.3 | 15.6 | 4.0 | 18.0 | 101.7 | 2 | 2.04 | 9 |
| | | | s.d. | 2.1 | 0.2 | 0 | 2.1 | 0.1 | 0.0 | 1.3 | 14 | 5 | 7 | 41 | 14 | 2.5 | 3.2 | 20 | 3.6 | 3.7 | 3.0 | 4.9 | 0.0 | 0.0 | 0.5 | 0 | 0.03 | |
| | | 2 | Mean | 13.1 | -1.1 | 104 | 16 | 5.2 | 5.6 | 8.4 | 1903 | 643 | 898 | 473 | 252 | 67.8 | 74.8 | 404 | 80.4 | 44.5 | 18.8 | 17.2 | 4.0 | 18.0 | 101.4 | 2 | 2.07 | 5 |
| | | | s.d. | 2.1 | 0.2 | 0 | 2.0 | 0.1 | 0.0 | 1.2 | 14 | 4 | 6 | 42 | 13 | 2.4 | 3.3 | 21 | 3.7 | 3.8 | 3.1 | 5.3 | 0.0 | 0.0 | 0.5 | 0 | 0.03 | |
| | | 3 | Mean | 12.2 | -1.0 | 102 | 16 | 5.2 | 5.6 | 7.7 | 1889 | 639 | 891 | 448 | 247 | 68.0 | 74.5 | 402 | 79.4 | 41.8 | 17.5 | 20.1 | 4.0 | 18.0 | 101.8 | 2 | 2.07 | 1 |
| | | | s.d. | 2.1 | 0.2 | 0 | 2.0 | 0.1 | 0.1 | 1.2 | 14 | 5 | 7 | 40 | 15 | 2.6 | 3.2 | 20 | 3.6 | 3.7 | 3.0 | 4.8 | 0.0 | 0.0 | 0.4 | 0 | 0.02 | |
| Mean of Three Replications | | | | 12.7 | -1.0 | 104 | 15 | 5.1 | 5.6 | 8.4 | 1895 | 641 | 894 | 460 | 252 | 67.5 | 73.8 | 398 | 78.9 | 43.0 | 18.2 | 17.6 | 4.0 | 18.0 | 101.7 | 2 | 2.06 | |
| | | | | $H_A$ | $V_A$ | $D_{HA}$ | $\varepsilon_r$ | $v_a$ | $v_t$ | S | $\omega_E$ | $\omega_T$ | $\omega_P$ | $T_{BH}$ | $T_{PTO}$ | $T_{ES}$ | $T_{FS}$ | $T_F$ | $P_F$ | $P_P$ | $P_D$ | $P_L$ | $B_S$ | $B_\%$ | $T_{DV}$ | $\delta_I$ | $A_{HT}$ | |

*Table A4.11 – Field results: steady state power harrowing, 100mm working depth (clay soil)*

| Target Depth (mm) | Gear | Rep | | Draught Force (kN) | Vertical Force (kN) | Harrow Depth (mm) | Rock shaft (%) | True Speed (km/h) | Wheel Speed (km/h) | Slip (%) | Engine Speed (rpm) | Trans Speed (rpm) | P.T.O. Speed (rpm) | P.T.O. Torque (Nm) | P.T.O. Tq (Veh) (Nm) | Engine Torque (%) | Flywheel Torque (%) | Flywheel Torque (Nm) | Flywheel Power (kW) | P.T.O. Power (kW) | Drawbar Power (kW) | Lost Power (kW) | Boost Status | Boost (%) | Torque Demand (%) | Engine Droop | Theo. Area (ha/hr) | Run |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 125 | 4 | 1 | Mean | 11.2 | -1.0 | 126 | 12 | 3.3 | 3.6 | 7.9 | 1921 | 413 | 908 | 440 | 246 | 57.5 | 63.0 | 327 | 65.7 | 41.8 | 10.4 | 13.5 | 4.0 | 16.2 | 99.1 | 1 | 1.33 | 23 |
| | | | s.d. | 2.2 | 0.2 | 0 | 2.0 | 0.1 | 0.0 | 1.5 | 13 | 3 | 6 | 38 | 13 | 2.4 | 1.7 | 14 | 2.5 | 3.5 | 2.0 | 3.9 | 0.0 | 0.9 | 1.1 | 0 | 0.02 | |
| | | 2 | Mean | 10.7 | -1.0 | 126 | 10 | 3.4 | 3.7 | 7.7 | 1935 | 416 | 913 | 409 | 246 | 56.6 | 62.0 | 318 | 64.0 | 39.1 | 10.0 | 15.0 | 4.0 | 15.2 | 97.7 | 1 | 1.35 | 25 |
| | | | s.d. | 2.0 | 0.2 | 0 | 2.1 | 0.1 | 0.0 | 1.8 | 13 | 3 | 8 | 39 | 13 | 2.4 | 1.4 | 17 | 3.2 | 3.8 | 1.9 | 3.9 | 0.0 | 1.3 | 1.4 | 0 | 0.02 | |
| | | 3 | Mean | 11.4 | -1.1 | 127 | 13 | 3.3 | 3.8 | 8.6 | 1915 | 412 | 904 | 430 | 259 | 59.5 | 63.9 | 333 | 66.8 | 40.7 | 10.4 | 15.6 | 4.0 | 15.6 | 98.7 | 1 | 1.32 | 27 |
| | | | s.d. | 2.0 | 0.2 | 0 | 2.3 | 0.0 | 0.0 | 1.4 | 13 | 3 | 6 | 42 | 16 | 2.5 | 2.2 | 16 | 3.0 | 3.9 | 1.8 | 4.0 | 0.0 | 0.7 | 0.8 | 0 | 0.02 | |
| Mean of Three Replications | | | | 11.1 | -1.0 | 126 | 12 | 3.3 | 3.6 | 8.1 | 1924 | 414 | 908 | 427 | 250 | 57.9 | 63.0 | 325 | 65.5 | 40.5 | 10.3 | 14.7 | 4.0 | 15.7 | 98.5 | 1 | 1.33 | |
| 125 | 5 | 1 | Mean | 13.2 | -0.9 | 128 | 11 | 4.2 | 4.6 | 7.5 | 1894 | 522 | 893 | 516 | 277 | 65.8 | 71.1 | 380 | 75.3 | 48.3 | 15.5 | 11.6 | 4.0 | 18.0 | 101.7 | 2 | 1.69 | 19 |
| | | | s.d. | 2.2 | 0.2 | 0 | 2.1 | 0.1 | 0.0 | 2.1 | 13 | 3 | 6 | 41 | 13 | 2.2 | 3.0 | 19 | 3.5 | 3.7 | 2.7 | 4.7 | 0.0 | 0.0 | 0.5 | 0 | 0.04 | |
| | | 2 | Mean | 13.6 | -1.2 | 126 | 14 | 4.2 | 4.6 | 9.8 | 1918 | 528 | 905 | 495 | 289 | 67.9 | 73.3 | 394 | 79.1 | 46.9 | 15.8 | 16.4 | 4.0 | 17.9 | 100.9 | 2 | 1.67 | 24 |
| | | | s.d. | 2.4 | 0.3 | 0 | 2.4 | 0.1 | 0.0 | 1.6 | 15 | 4 | 7 | 51 | 18 | 2.9 | 3.9 | 25 | 4.5 | 4.6 | 2.8 | 5.1 | 0.0 | 0.2 | 0.5 | 0 | 0.03 | |
| | | 3 | Mean | 13.7 | -1.1 | 127 | 14 | 4.1 | 4.5 | 11.2 | 1920 | 529 | 905 | 478 | 287 | 68.1 | 74.0 | 398 | 80.0 | 45.3 | 15.7 | 19.1 | 4.0 | 18.0 | 100.9 | 2 | 1.65 | 26 |
| | | | s.d. | 2.4 | 0.2 | 0 | 2.2 | 0.1 | 0.0 | 1.5 | 14 | 4 | 7 | 42 | 15 | 2.5 | 3.5 | 23 | 4.1 | 3.9 | 2.7 | 4.9 | 0.0 | 0.0 | 0.5 | 0 | 0.03 | |
| Mean of Three Replications | | | | 13.5 | -1.1 | 126 | 13 | 4.2 | 4.6 | 9.5 | 1910 | 526 | 901 | 496 | 284 | 67.3 | 72.8 | 391 | 78.2 | 46.8 | 15.7 | 15.7 | 4.0 | 18.0 | 101.2 | 2 | 1.67 | |
| 125 | 6 | 1 | Mean | 15.0 | -1.3 | 126 | 16 | 4.9 | 5.7 | 14.5 | 1915 | 647 | 903 | 538 | 312 | 78.5 | 85.4 | 472 | 94.6 | 50.9 | 20.2 | 23.5 | 4.0 | 18.0 | 101.1 | 2 | 1.94 | 22 |
| | | | s.d. | 2.7 | 0.3 | 1 | 2.5 | 0.1 | 0.1 | 1.8 | 17 | 6 | 8 | 48 | 16 | 3.1 | 4.0 | 25 | 4.5 | 4.4 | 3.8 | 6.3 | 0.0 | 0.0 | 0.6 | 0 | 0.04 | |
| | | 2 | Mean | 14.6 | -1.0 | 126 | 14 | 4.9 | 5.6 | 13.0 | 1896 | 641 | 894 | 560 | 300 | 76.7 | 83.0 | 457 | 90.6 | 52.4 | 19.8 | 18.4 | 4.0 | 18.0 | 101.6 | 2 | 1.96 | 20 |
| | | | s.d. | 2.4 | 0.2 | 0 | 2.0 | 0.1 | 0.0 | 1.7 | 15 | 5 | 7 | 42 | 14 | 2.7 | 3.4 | 22 | 3.8 | 3.8 | 3.3 | 5.3 | 0.0 | 0.0 | 0.5 | 0 | 0.04 | |
| | | 3 | Mean | 14.8 | -1.0 | 126 | 15 | 4.9 | 5.6 | 12.8 | 1895 | 641 | 894 | 552 | 303 | 77.3 | 83.7 | 461 | 91.4 | 51.6 | 20.2 | 19.6 | 4.0 | 18.0 | 101.6 | 2 | 1.96 | 21 |
| | | | s.d. | 2.4 | 0.3 | 0 | 2.2 | 0.1 | 0.0 | 2.4 | 14 | 5 | 7 | 44 | 14 | 2.8 | 3.6 | 23 | 4.0 | 3.9 | 3.3 | 5.4 | 0.0 | 0.0 | 0.5 | 0 | 0.06 | |
| Mean of Three Replications | | | | 14.8 | -1.1 | 126 | 15 | 4.9 | 5.6 | 13.4 | 1902 | 643 | 897 | 550 | 305 | 77.5 | 84.1 | 463 | 92.2 | 51.6 | 20.1 | 20.6 | 4.0 | 18.0 | 101.5 | 2 | 1.95 | |
| | | | | $H_A$ | $V_A$ | $D_{HA}$ | $\varepsilon_r$ | $v_a$ | $v_t$ | $S$ | $\omega_E$ | $\omega_T$ | $\omega_P$ | $T_{SH}$ | $T_{PTO}$ | $T_{E\%}$ | $T_{F\%}$ | $T_F$ | $P_F$ | $P_P$ | $P_D$ | $P_L$ | $B_S$ | $B_\%$ | $T_{DV}$ | $\delta_I$ | $A_{HT}$ | |

*Table A4.12 – Field results: steady state power harrowing, 125mm working depth (clay soil)*

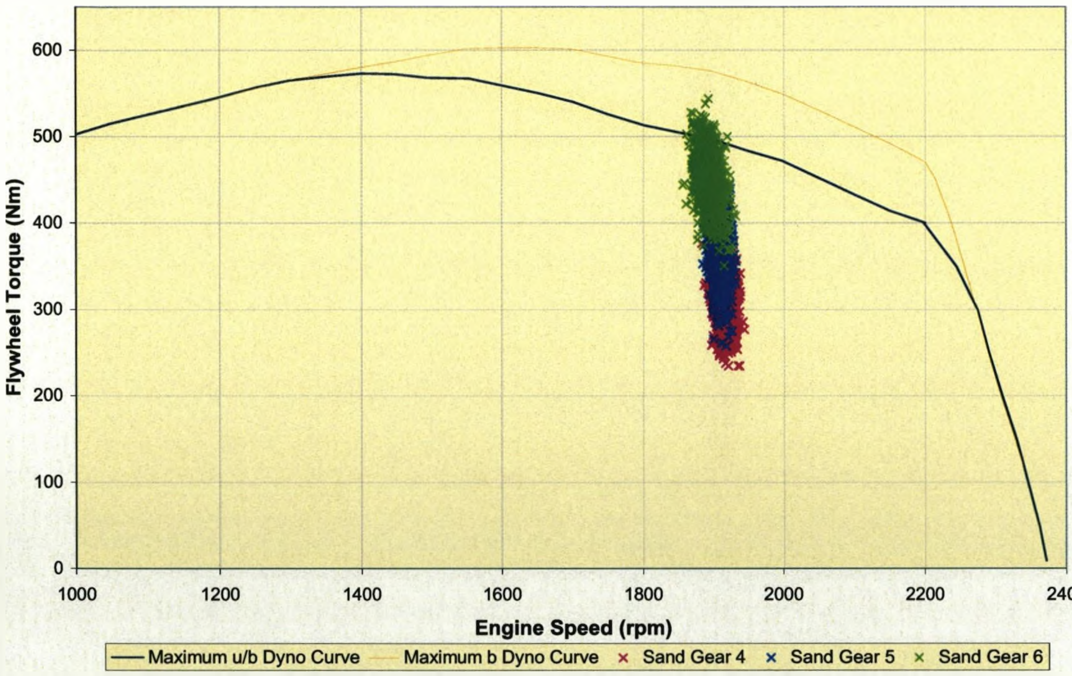


*Figure A4.5 – The effect of gear selection on dynamic loading whilst power harrowing sandy soil (75mm tine depth)*

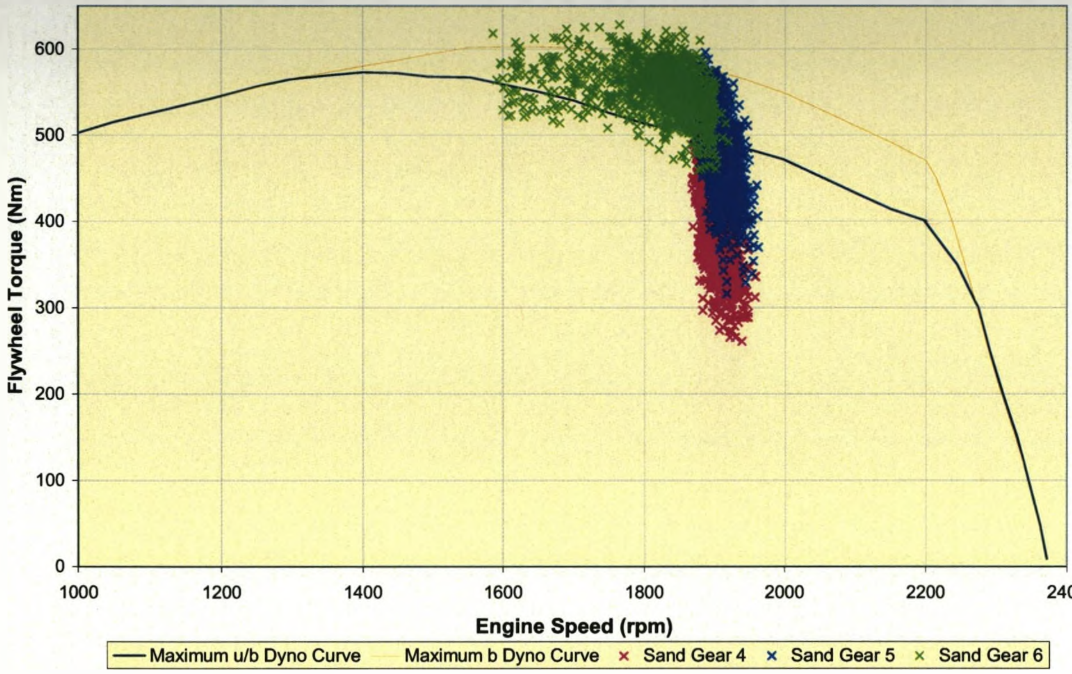


*Figure A4.6 – The effect of gear selection on dynamic loading whilst power harrowing sandy soil (125mm tine depth)*

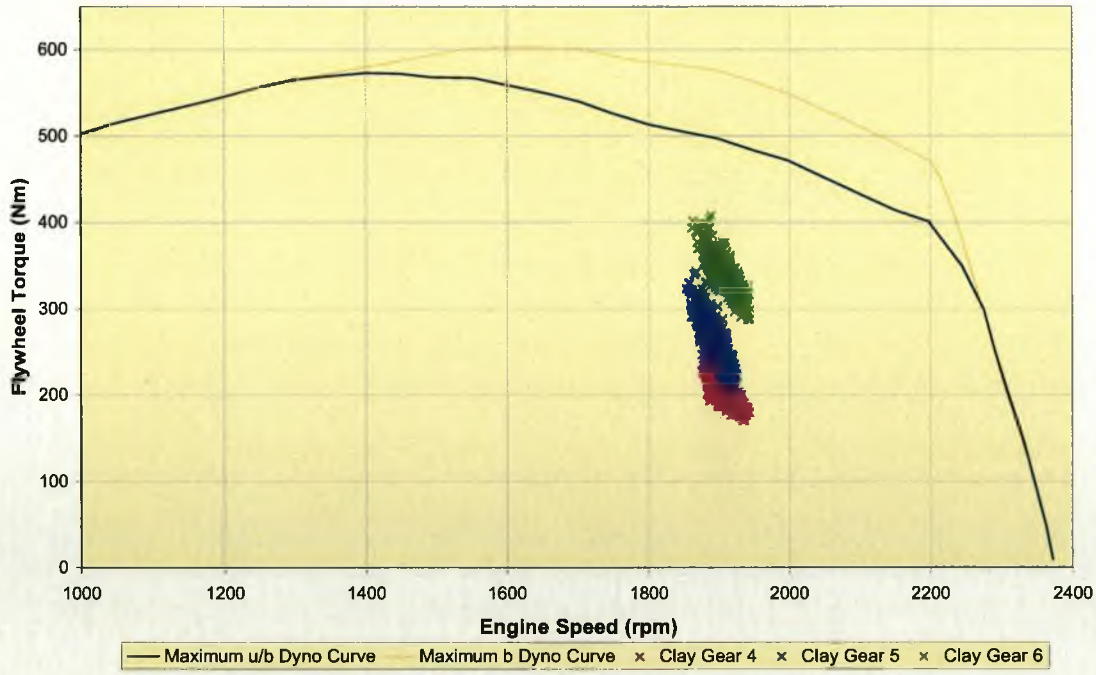


*Figure A4.7 – The effect of gear selection on dynamic loading whilst power harrowing clay soil (75mm tine depth)*

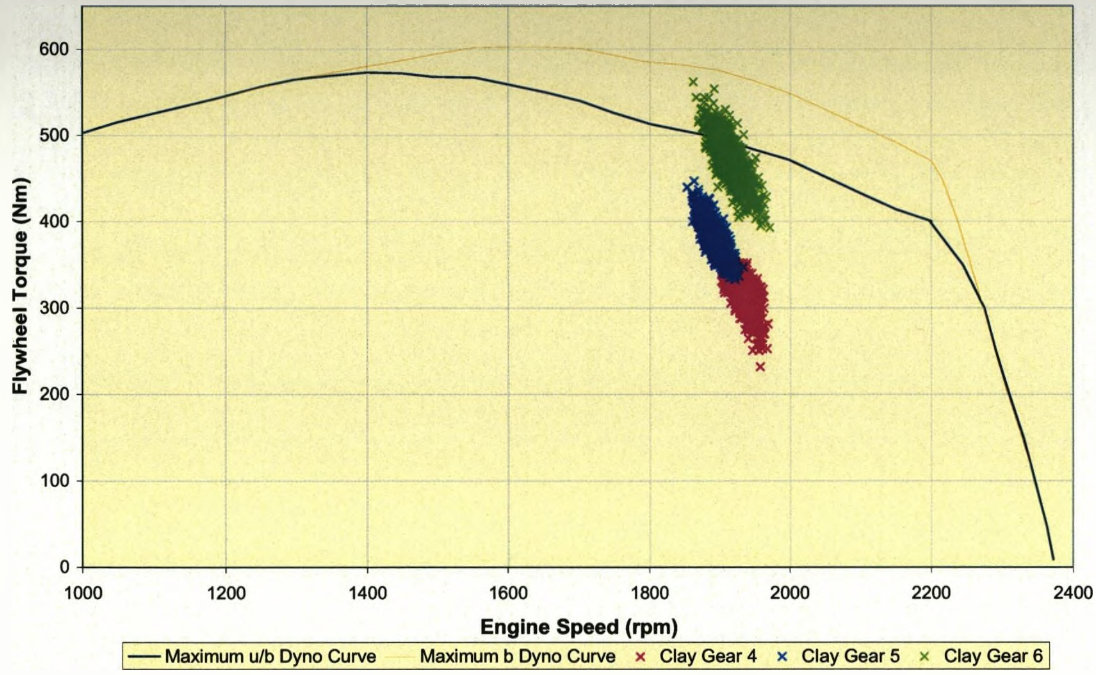


*Figure A4.8 – The effect of gear selection on dynamic loading whilst power harrowing clay soil (125mm tine depth)*

# A5 Additional Axle Dynamometer Data

## A5.1 Load Cell Calibration Results

*Table A5.1 – Axle dynamometer load cell calibration data*

| Wheel 3 (Rear Left) | | | | Wheel 4 (Rear Right) | | | |
|---|---|---|---|---|---|---|---|
| Test Torque | Indicated Values | | Error | Test Torque | Indicated Values | | Error |
| kNm | Bits | kNm | kNm | kNm | Bits | kNm | kNm |
| 0 | 305 | 0.06 | 0.06 | 0 | 306 | 0.04 | 0.04 |
| 10 | 1065 | 10.04 | 0.04 | 10 | 1066 | 10.00 | 0.00 |
| 20 | 1826 | 20.05 | 0.05 | 20 | 1827 | 20.00 | 0.00 |
| 30 | 2586 | 30.05 | 0.05 | 30 | 2588 | 30.03 | 0.03 |
| 40 | 3364 | 40.23 | 0.23 | 40 | 3364 | 40.20 | 0.20 |
| 0 | 304 | 0.05 | 0.05 | 0 | 305 | 0.04 | 0.04 |
| 10 | 1064 | 10.03 | 0.03 | 10 | 1067 | 10.03 | 0.03 |
| 20 | 1826 | 20.04 | 0.04 | 20 | 1831 | 20.05 | 0.05 |
| 30 | 2585 | 30.04 | 0.04 | 30 | 2591 | 30.04 | 0.04 |
| 40 | 3364 | 40.25 | 0.25 | 40 | 3367 | 40.25 | 0.25 |
| 0 | 300 | 0.01 | 0.01 | 0 | 312 | 0.12 | 0.12 |
| 10 | 1062 | 10.00 | 0.00 | 10 | 1073 | 10.09 | 0.09 |
| 20 | 1823 | 20.00 | 0.00 | 20 | 1837 | 20.15 | 0.15 |
| 30 | 2584 | 30.02 | 0.02 | 30 | 2599 | 30.15 | 0.15 |
| 40 | 3362 | 40.24 | 0.24 | 40 | 3372 | 40.33 | 0.33 |

## A5.2 Driveline Loss Mean Data

*Table A5.2 – Loss and efficiency data – mean of three replications (No applied brake torque)*

| G | $\omega_E$ (rpm) | $\omega_T$ (rpm) | $v_t$ (km/h) | $T_{EF}$ (Nm) | $T_F$ (Nm) | $T_{LOSS}$ (Nm) | S.E.M. (of $T_{LOSS}$) | $P_{LOSS}$ (kW) | S.E.M. (of $P_{LOSS}$) | $\eta$ (%) | S.E.M. (of $\eta$) |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 1400 | 164 | 1.44 | 0.0 | 6.4 | 6.4 | 0.35 | 0.9 | 0.05 | 0.0 | 0.00 |
| 1 | 1802 | 212 | 1.86 | 0.0 | 8.7 | 8.7 | 0.17 | 1.6 | 0.03 | 0.0 | 0.00 |
| 1 | 2202 | 259 | 2.27 | 0.0 | 9.3 | 9.3 | 0.47 | 2.1 | 0.11 | 0.0 | 0.00 |
| 2 | 1403 | 202 | 1.77 | 0.0 | 6.3 | 6.3 | 0.38 | 0.9 | 0.06 | 0.0 | 0.00 |
| 2 | 1803 | 260 | 2.28 | 0.0 | 8.9 | 8.9 | 0.54 | 1.7 | 0.10 | 0.0 | 0.00 |
| 2 | 2200 | 317 | 2.78 | 0.0 | 9.3 | 9.3 | 0.51 | 2.1 | 0.12 | 0.0 | 0.00 |
| 3 | 1401 | 246 | 2.16 | 0.0 | 10.2 | 10.2 | 0.42 | 1.5 | 0.06 | 0.0 | 0.00 |
| 3 | 1800 | 316 | 2.77 | 0.0 | 12.2 | 12.2 | 0.27 | 2.3 | 0.05 | 0.0 | 0.00 |
| 3 | 2198 | 386 | 3.38 | 0.0 | 12.5 | 12.5 | 0.57 | 2.9 | 0.13 | 0.0 | 0.00 |
| 4 | 1401 | 301 | 2.64 | 0.0 | 10.4 | 10.4 | 0.24 | 1.5 | 0.03 | 0.0 | 0.00 |
| 4 | 1799 | 387 | 3.39 | 0.0 | 12.7 | 12.7 | 0.70 | 2.4 | 0.13 | 0.0 | 0.00 |
| 4 | 2201 | 474 | 4.15 | 0.0 | 13.3 | 13.3 | 0.62 | 3.1 | 0.14 | 0.0 | 0.00 |
| 5 | 1400 | 386 | 3.39 | 0.0 | 10.7 | 10.7 | 0.47 | 1.6 | 0.07 | 0.0 | 0.00 |
| 5 | 1802 | 497 | 4.36 | 0.0 | 13.1 | 13.1 | 0.66 | 2.5 | 0.12 | 0.0 | 0.00 |
| 5 | 2198 | 607 | 5.32 | 0.0 | 13.9 | 13.9 | 0.70 | 3.2 | 0.16 | 0.0 | 0.00 |
| 6 | 1400 | 474 | 4.15 | 0.0 | 11.9 | 11.9 | 0.62 | 1.7 | 0.09 | 0.0 | 0.00 |
| 6 | 1798 | 609 | 5.34 | 0.0 | 13.8 | 13.8 | 0.19 | 2.6 | 0.04 | 0.0 | 0.00 |
| 6 | 2203 | 746 | 6.54 | 0.0 | 14.1 | 14.1 | 0.93 | 3.3 | 0.22 | 0.0 | 0.00 |
| 7 | 1397 | 576 | 5.05 | 0.0 | 19.1 | 19.1 | 0.50 | 2.8 | 0.07 | 0.0 | 0.00 |
| 7 | 1800 | 742 | 6.51 | 0.0 | 23.0 | 23.0 | 1.93 | 4.3 | 0.37 | 0.0 | 0.00 |
| 7 | 2202 | 907 | 7.96 | 0.0 | 24.0 | 24.0 | 2.43 | 5.5 | 0.56 | 0.0 | 0.00 |
| 8 | 1402 | 709 | 6.22 | 0.0 | 20.0 | 20.0 | 0.90 | 2.9 | 0.13 | 0.0 | 0.00 |
| 8 | 1799 | 909 | 7.98 | 0.0 | 24.8 | 24.8 | 3.05 | 4.7 | 0.58 | 0.0 | 0.00 |
| 8 | 2201 | 1113 | 9.76 | 0.0 | 30.6 | 30.6 | 2.70 | 7.1 | 0.62 | 0.0 | 0.00 |
| 9 | 1401 | 643 | 5.64 | 0.0 | 8.9 | 8.9 | 0.16 | 1.3 | 0.02 | 0.0 | 0.00 |
| 9 | 1801 | 827 | 7.26 | 0.0 | 11.1 | 11.1 | 0.09 | 2.1 | 0.02 | 0.0 | 0.00 |
| 9 | 2199 | 1010 | 8.86 | 0.0 | 12.0 | 12.0 | 0.29 | 2.8 | 0.06 | 0.0 | 0.00 |
| 10 | 1400 | 789 | 6.92 | 0.0 | 10.0 | 10.0 | 0.40 | 1.5 | 0.06 | 0.0 | 0.00 |
| 10 | 1801 | 1014 | 8.90 | 0.0 | 12.2 | 12.2 | 0.52 | 2.3 | 0.10 | 0.0 | 0.00 |
| 10 | 2200 | 1239 | 10.87 | 0.0 | 13.9 | 13.9 | 0.31 | 3.2 | 0.08 | 0.0 | 0.00 |
| 11 | 1398 | 959 | 8.41 | 0.0 | 16.0 | 16.0 | 0.26 | 2.3 | 0.04 | 0.0 | 0.00 |
| 11 | 1800 | 1235 | 10.83 | 0.0 | 21.2 | 21.2 | 0.97 | 4.0 | 0.18 | 0.0 | 0.00 |
| 11 | 2201 | 1510 | 13.24 | 0.0 | 19.3 | 19.3 | 2.62 | 4.5 | 0.60 | 0.0 | 0.00 |
| 12 | 1403 | 1181 | 10.36 | 0.0 | 19.4 | 19.4 | 0.20 | 2.9 | 0.03 | 0.0 | 0.00 |
| 12 | 1803 | 1518 | 13.31 | 0.0 | 24.1 | 24.1 | 2.98 | 4.5 | 0.56 | 0.0 | 0.00 |
| 12 | 2202 | 1853 | 16.25 | 0.0 | 21.4 | 21.4 | 0.68 | 4.9 | 0.16 | 0.0 | 0.00 |
| 13 | 1397 | 1508 | 13.22 | 0.0 | 23.1 | 23.1 | 1.20 | 3.4 | 0.18 | 0.0 | 0.00 |
| 13 | 1800 | 1943 | 17.04 | 0.0 | 27.3 | 27.3 | 2.65 | 5.2 | 0.50 | 0.0 | 0.00 |
| 13 | 2201 | 2375 | 20.83 | 0.0 | 36.3 | 36.3 | 0.84 | 8.4 | 0.19 | 0.0 | 0.00 |
| 14 | 1402 | 1856 | 16.27 | 0.0 | 29.8 | 29.8 | 2.30 | 4.4 | 0.34 | 0.0 | 0.00 |
| 14 | 1801 | 2383 | 20.90 | 0.0 | 33.1 | 33.1 | 6.28 | 6.2 | 1.18 | 0.0 | 0.00 |
| 14 | 2202 | 2914 | 25.56 | 0.0 | 42.8 | 42.8 | 4.07 | 9.9 | 0.94 | 0.0 | 0.00 |
| 15 | 1400 | 2256 | 19.79 | 0.0 | 44.7 | 44.7 | 4.58 | 6.5 | 0.67 | 0.0 | 0.00 |
| 15 | 1802 | 2904 | 25.46 | 0.0 | 56.7 | 56.7 | 4.73 | 10.7 | 0.88 | 0.0 | 0.00 |
| 15 | 2203 | 3551 | 31.14 | 0.0 | 67.9 | 67.9 | 1.20 | 15.7 | 0.28 | 0.0 | 0.00 |
| 16 | 1403 | 2774 | 24.33 | 0.0 | 54.4 | 54.4 | 1.90 | 8.0 | 0.24 | 0.0 | 0.00 |
| 16 | 1802 | 3562 | 31.24 | 0.0 | 72.8 | 72.8 | 1.24 | 13.7 | 0.22 | 0.0 | 0.00 |
| 16 | 2201 | 4351 | 38.16 | 0.0 | 87.9 | 87.9 | 2.55 | 20.3 | 0.61 | 0.0 | 0.00 |

*Table A5.3 – Loss and efficiency data – mean of three replications (≈ 100Nm flywheel torque)*

| G | $\omega_E$ (rpm) | $\omega_T$ (rpm) | $v_t$ (km/h) | $T_{EF}$ (Nm) | $T_F$ (Nm) | $T_{LOSS}$ (Nm) | S.E.M. (of $T_{LOSS}$) | $P_{LOSS}$ (kW) | S.E.M. (of $P_{LOSS}$) | $\eta$ (%) | S.E.M. (of $\eta$) |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 1403 | 165 | 1.45 | 91.7 | 104.8 | 13.1 | 1.91 | 1.9 | 0.29 | 87.5 | 1.53 |
| 1 | 1803 | 212 | 1.86 | 87.8 | 104.8 | 17.0 | 1.07 | 3.2 | 0.21 | 83.8 | 0.71 |
| 1 | 2198 | 258 | 2.26 | 88.7 | 104.6 | 15.9 | 2.44 | 3.7 | 0.55 | 84.8 | 1.58 |
| 2 | 1399 | 202 | 1.77 | 92.0 | 103.4 | 11.4 | 1.14 | 1.7 | 0.17 | 89.0 | 1.20 |
| 2 | 1799 | 259 | 2.27 | 88.4 | 102.1 | 13.7 | 0.73 | 2.6 | 0.15 | 86.5 | 0.84 |
| 2 | 2202 | 317 | 2.78 | 88.8 | 104.8 | 16.1 | 1.20 | 3.7 | 0.27 | 84.7 | 1.16 |
| 3 | 1403 | 246 | 2.16 | 86.4 | 102.9 | 16.5 | 1.43 | 2.4 | 0.21 | 84.0 | 0.99 |
| 3 | 1803 | 316 | 2.77 | 85.5 | 106.7 | 21.2 | 1.22 | 4.0 | 0.24 | 80.2 | 0.22 |
| 3 | 2199 | 386 | 3.38 | 91.0 | 114.6 | 23.6 | 2.00 | 5.4 | 0.46 | 79.4 | 1.12 |
| 4 | 1400 | 301 | 2.64 | 84.0 | 98.5 | 14.6 | 0.40 | 2.1 | 0.05 | 85.2 | 0.44 |
| 4 | 1800 | 387 | 3.40 | 83.3 | 103.3 | 20.1 | 0.73 | 3.8 | 0.13 | 80.6 | 0.39 |
| 4 | 2200 | 473 | 4.15 | 85.9 | 107.3 | 21.4 | 2.51 | 4.9 | 0.58 | 80.1 | 1.20 |
| 5 | 1404 | 387 | 3.40 | 84.9 | 100.3 | 15.4 | 0.92 | 2.3 | 0.14 | 84.6 | 0.75 |
| 5 | 1802 | 497 | 4.36 | 80.3 | 100.0 | 19.7 | 1.98 | 3.7 | 0.36 | 80.3 | 1.01 |
| 5 | 2198 | 607 | 5.32 | 88.9 | 110.4 | 21.5 | 3.19 | 5.0 | 0.73 | 80.5 | 1.48 |
| 6 | 1403 | 475 | 4.16 | 91.8 | 108.7 | 16.9 | 2.74 | 2.5 | 0.39 | 84.5 | 1.46 |
| 6 | 1796 | 608 | 5.33 | 85.2 | 105.7 | 20.5 | 1.27 | 3.8 | 0.24 | 80.6 | 0.62 |
| 6 | 2201 | 745 | 6.53 | 81.2 | 101.5 | 20.3 | 0.56 | 4.7 | 0.12 | 80.0 | 0.74 |
| 7 | 1395 | 575 | 5.04 | 74.8 | 97.6 | 22.8 | 1.31 | 3.3 | 0.18 | 76.7 | 0.66 |
| 7 | 1799 | 741 | 6.50 | 73.9 | 105.3 | 31.4 | 3.64 | 5.9 | 0.69 | 70.2 | 0.60 |
| 7 | 2195 | 905 | 7.93 | 68.1 | 101.0 | 33.0 | 1.02 | 7.6 | 0.24 | 67.4 | 0.82 |
| 8 | 1402 | 709 | 6.21 | 78.2 | 102.2 | 24.1 | 2.27 | 3.5 | 0.35 | 76.5 | 0.60 |
| 8 | 1796 | 908 | 7.96 | 69.2 | 97.7 | 28.5 | 1.02 | 5.4 | 0.22 | 70.8 | 0.77 |
| 8 | 2202 | 1113 | 9.76 | 74.1 | 107.0 | 32.9 | 1.61 | 7.6 | 0.37 | 69.3 | 2.08 |
| 9 | 1403 | 644 | 5.65 | 86.3 | 99.8 | 13.6 | 2.73 | 2.0 | 0.40 | 86.4 | 1.71 |
| 9 | 1796 | 825 | 7.24 | 87.9 | 106.9 | 19.0 | 0.35 | 3.6 | 0.06 | 82.2 | 0.25 |
| 9 | 2200 | 1011 | 8.86 | 84.6 | 105.2 | 20.5 | 1.54 | 4.7 | 0.35 | 80.5 | 1.01 |
| 10 | 1406 | 792 | 6.95 | 90.3 | 107.3 | 17.0 | 2.83 | 2.5 | 0.41 | 84.1 | 1.02 |
| 10 | 1797 | 1012 | 8.88 | 86.8 | 107.8 | 21.0 | 2.15 | 3.9 | 0.40 | 80.5 | 0.85 |
| 10 | 2199 | 1239 | 10.86 | 76.4 | 97.8 | 21.4 | 1.64 | 4.9 | 0.36 | 78.1 | 1.16 |
| 11 | 1402 | 962 | 8.43 | 82.6 | 104.6 | 22.0 | 1.30 | 3.2 | 0.19 | 79.0 | 0.73 |
| 11 | 1796 | 1232 | 10.81 | 76.9 | 103.7 | 26.7 | 0.70 | 5.0 | 0.11 | 74.2 | 0.93 |
| 11 | 2195 | 1506 | 13.20 | 78.3 | 107.7 | 29.4 | 1.51 | 6.8 | 0.35 | 72.7 | 0.97 |
| 12 | 1401 | 1179 | 10.34 | 79.5 | 104.0 | 24.5 | 1.20 | 3.6 | 0.17 | 76.4 | 1.13 |
| 12 | 1800 | 1515 | 13.29 | 75.4 | 106.3 | 30.9 | 1.03 | 5.8 | 0.18 | 71.0 | 0.89 |
| 12 | 2200 | 1851 | 16.24 | 72.6 | 105.8 | 33.2 | 1.35 | 7.6 | 0.31 | 68.6 | 0.79 |
| 13 | 1398 | 1508 | 13.22 | 77.5 | 106.6 | 29.0 | 0.78 | 4.2 | 0.11 | 72.8 | 1.03 |
| 13 | 1803 | 1945 | 17.06 | 64.5 | 98.9 | 34.5 | 1.82 | 6.5 | 0.33 | 65.2 | 2.78 |
| 13 | 2200 | 2373 | 20.81 | 62.3 | 103.4 | 41.1 | 2.08 | 9.5 | 0.45 | 60.3 | 0.35 |
| 14 | 1403 | 1857 | 16.28 | 65.2 | 99.5 | 34.3 | 0.42 | 5.0 | 0.08 | 65.5 | 1.90 |
| 14 | 1804 | 2387 | 20.94 | 59.9 | 104.0 | 44.1 | 1.86 | 8.3 | 0.34 | 57.6 | 1.36 |
| 14 | 2202 | 2914 | 25.56 | 50.8 | 100.9 | 50.1 | 1.47 | 11.6 | 0.35 | 50.3 | 2.68 |
| 15 | 1397 | 2251 | 19.74 | 50.7 | 101.3 | 50.6 | 2.09 | 7.4 | 0.30 | 50.1 | 0.80 |
| 15 | 1799 | 2900 | 25.43 | 39.0 | 103.0 | 64.0 | 0.77 | 12.1 | 0.17 | 37.9 | 2.30 |
| 15 | 2196 | 3540 | 31.04 | 33.2 | 103.7 | 70.5 | 1.06 | 16.2 | 0.30 | 32.0 | 2.34 |
| 16 | 1403 | 2773 | 24.32 | 35.1 | 97.3 | 62.2 | 2.69 | 9.1 | 0.39 | 36.1 | 0.52 |
| 16 | 1795 | 3549 | 31.13 | 28.8 | 103.6 | 74.8 | 2.25 | 14.1 | 0.40 | 27.8 | 2.15 |
| 16 | 2195 | 4340 | 38.06 | 19.1 | 106.1 | 87.0 | 1.61 | 20.0 | 0.39 | 18.0 | 3.54 |

*Table A5.4 – Loss and efficiency data – mean of three replications (≈ 250Nm flywheel torque)*

| G | $\omega_E$ (rpm) | $\omega_T$ (rpm) | $v_t$ (km/h) | $T_{EF}$ (Nm) | $T_F$ (Nm) | $T_{LOSS}$ (Nm) | S.E.M. (of $T_{LOSS}$) | $P_{LOSS}$ (kW) | S.E.M. (of $P_{LOSS}$) | $\eta$ (%) | S.E.M. (of $\eta$) |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 4 | 1402 | 302 | 2.65 | 243.9 | 263.0 | 19.0 | 1.33 | 2.8 | 0.20 | 92.8 | 0.53 |
| 4 | 1794 | 386 | 3.39 | 253.8 | 278.2 | 24.4 | 4.53 | 4.6 | 0.85 | 91.2 | 0.93 |
| 4 | 2203 | 474 | 4.16 | 239.6 | 261.1 | 21.5 | 4.38 | 5.0 | 1.03 | 91.8 | 1.32 |
| 5 | 1402 | 387 | 3.39 | 247.7 | 262.4 | 14.6 | 0.73 | 2.1 | 0.12 | 94.4 | 0.48 |
| 5 | 1799 | 496 | 4.35 | 240.3 | 256.1 | 15.8 | 1.93 | 3.0 | 0.36 | 93.8 | 0.53 |
| 5 | 2196 | 606 | 5.31 | 254.2 | 279.1 | 24.9 | 0.83 | 5.7 | 0.19 | 91.1 | 0.31 |
| 6 | 1407 | 476 | 4.18 | 242.9 | 253.2 | 10.2 | 0.92 | 1.5 | 0.13 | 96.0 | 0.29 |
| 6 | 1797 | 608 | 5.33 | 247.5 | 258.6 | 11.2 | 1.28 | 2.1 | 0.24 | 95.7 | 0.45 |
| 6 | 2197 | 744 | 6.52 | 242.0 | 254.9 | 13.0 | 4.12 | 3.0 | 0.95 | 94.9 | 1.32 |
| 7 | 1408 | 580 | 5.09 | 228.8 | 250.4 | 21.7 | 0.56 | 3.2 | 0.08 | 91.4 | 0.65 |
| 7 | 1799 | 742 | 6.50 | 232.3 | 259.6 | 27.3 | 2.56 | 5.1 | 0.48 | 89.5 | 0.50 |
| 7 | 2200 | 907 | 7.95 | 236.5 | 267.9 | 31.4 | 0.69 | 7.2 | 0.18 | 88.3 | 0.22 |
| 8 | 1406 | 711 | 6.23 | 243.5 | 264.7 | 21.2 | 3.32 | 3.1 | 0.49 | 92.0 | 1.25 |
| 8 | 1800 | 910 | 7.98 | 244.9 | 274.5 | 29.6 | 1.47 | 5.6 | 0.29 | 89.2 | 0.40 |
| 8 | 2202 | 1113 | 9.76 | 230.6 | 258.1 | 27.5 | 1.06 | 6.3 | 0.24 | 89.4 | 0.28 |
| 9 | 1406 | 646 | 5.66 | 239.3 | 255.8 | 16.5 | 2.74 | 2.4 | 0.40 | 93.6 | 0.62 |
| 9 | 1799 | 826 | 7.25 | 247.6 | 267.7 | 20.1 | 4.78 | 3.8 | 0.89 | 92.5 | 1.46 |
| 9 | 2198 | 1009 | 8.85 | 242.6 | 259.7 | 17.1 | 5.75 | 3.9 | 1.32 | 93.4 | 1.93 |
| 10 | 1407 | 793 | 6.95 | 236.0 | 250.7 | 14.7 | 1.43 | 2.2 | 0.21 | 94.1 | 0.64 |
| 10 | 1808 | 1018 | 8.93 | 244.5 | 261.3 | 16.8 | 6.36 | 3.2 | 1.19 | 93.6 | 2.12 |
| 10 | 2195 | 1237 | 10.85 | 248.5 | 269.3 | 20.8 | 2.51 | 4.8 | 0.57 | 92.3 | 0.64 |
| 11 | 1398 | 959 | 8.41 | 236.9 | 258.4 | 21.5 | 1.68 | 3.1 | 0.26 | 91.7 | 0.12 |
| 11 | 1804 | 1238 | 10.85 | 228.0 | 251.1 | 23.1 | 1.76 | 4.4 | 0.33 | 90.8 | 0.44 |
| 11 | 2203 | 1511 | 13.25 | 237.1 | 261.6 | 24.5 | 1.25 | 5.7 | 0.27 | 90.6 | 0.34 |
| 12 | 1405 | 1183 | 10.37 | 235.4 | 257.6 | 22.2 | 1.16 | 3.3 | 0.17 | 91.4 | 0.06 |
| 12 | 1798 | 1513 | 13.27 | 248.8 | 276.5 | 27.7 | 1.77 | 5.2 | 0.33 | 90.0 | 0.72 |
| 12 | 2200 | 1851 | 16.24 | 221.7 | 246.0 | 24.3 | 3.31 | 5.6 | 0.76 | 90.1 | 0.88 |
| 13 | 1399 | 1509 | 13.23 | 235.1 | 256.9 | 21.9 | 1.54 | 3.2 | 0.23 | 91.5 | 0.33 |
| 13 | 1798 | 1940 | 17.01 | 241.4 | 270.7 | 29.3 | 1.17 | 5.5 | 0.22 | 89.2 | 0.48 |
| 13 | 2195 | 2368 | 20.79 | 237.8 | 271.4 | 33.5 | 1.37 | 7.7 | 0.30 | 87.6 | 0.35 |
| 14 | 1401 | 1855 | 16.27 | 224.8 | 251.1 | 26.3 | 2.68 | 3.9 | 0.37 | 89.5 | 0.98 |
| 14 | 1808 | 2393 | 20.98 | 207.3 | 240.1 | 32.8 | 0.82 | 5.4 | 0.19 | 87.4 | 0.30 |
| 14 | 2199 | 2910 | 25.49 | 207.6 | 241.3 | 33.7 | 2.55 | 7.8 | 0.57 | 86.0 | 1.13 |
| 15 | 1402 | 2259 | 19.81 | 205.6 | 250.8 | 45.2 | 2.97 | 6.6 | 0.44 | 82.0 | 1.62 |
| 15 | 1804 | 2908 | 25.50 | 203.8 | 257.8 | 54.0 | 2.49 | 10.2 | 0.49 | 79.0 | 0.50 |
| 15 | 2202 | 3549 | 31.13 | 192.3 | 252.8 | 60.4 | 2.95 | 13.9 | 0.67 | 76.1 | 0.85 |
| 16 | 1402 | 2771 | 24.30 | 201.5 | 259.0 | 57.5 | 2.87 | 8.4 | 0.44 | 77.8 | 0.91 |
| 16 | 1804 | 3566 | 31.27 | 191.1 | 257.8 | 66.8 | 2.45 | 12.6 | 0.52 | 74.1 | 0.10 |
| 16 | 2190 | 4330 | 37.93 | 183.4 | 257.3 | 73.9 | 0.14 | 16.9 | 0.04 | 71.3 | 0.13 |

*Table A5.5 – Loss and efficiency data – mean of three replications (≈ 400Nm flywheel torque)*

| G | $\omega_E$ (rpm) | $\omega_T$ (rpm) | $v_t$ (km/h) | $T_{EF}$ (Nm) | $T_F$ (Nm) | $T_{LOSS}$ (Nm) | S.E.M. (of $T_{LOSS}$) | $P_{LOSS}$ (kW) | S.E.M. (of $P_{LOSS}$) | $\eta$ (%) | S.E.M. (of $\eta$) |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 6 | 1408 | 477 | 4.18 | 390.6 | 403.3 | 12.6 | 5.72 | 1.9 | 0.84 | 96.9 | 1.37 |
| 6 | 1802 | 610 | 5.35 | 392.4 | 408.8 | 16.4 | 3.26 | 3.1 | 0.62 | 96.0 | 0.74 |
| 6 | 2172 | 735 | 6.45 | 370.2 | 394.7 | 24.6 | 4.66 | 5.6 | 1.07 | 93.8 | 1.12 |
| 7 | 1406 | 579 | 5.08 | 383.6 | 414.8 | 31.2 | 2.51 | 4.6 | 0.36 | 92.5 | 0.85 |
| 7 | 1802 | 743 | 6.51 | 374.1 | 408.7 | 34.6 | 5.48 | 6.5 | 1.04 | 91.5 | 1.27 |
| 7 | 2173 | 896 | 7.85 | 355.2 | 393.0 | 37.9 | 5.00 | 8.6 | 1.23 | 90.4 | 1.35 |
| 8 | 1400 | 708 | 6.21 | 383.2 | 407.3 | 24.1 | 3.82 | 3.5 | 0.55 | 94.1 | 1.05 |
| 8 | 1802 | 911 | 7.99 | 379.5 | 410.7 | 31.2 | 8.84 | 5.9 | 1.67 | 92.4 | 1.98 |
| 8 | 2147 | 1086 | 9.52 | 366.2 | 402.2 | 36.0 | 4.04 | 8.1 | 1.00 | 91.1 | 1.16 |
| 9 | 1406 | 646 | 5.66 | 379.1 | 406.3 | 27.2 | 6.05 | 4.0 | 0.88 | 93.3 | 1.50 |
| 9 | 1798 | 826 | 7.24 | 380.4 | 402.6 | 22.2 | 2.59 | 4.2 | 0.49 | 94.5 | 0.61 |
| 9 | 2178 | 1000 | 8.77 | 366.9 | 398.2 | 31.4 | 4.57 | 7.1 | 1.01 | 92.1 | 1.10 |
| 10 | 1405 | 792 | 6.94 | 387.5 | 405.7 | 18.2 | 2.24 | 2.7 | 0.33 | 95.5 | 0.60 |
| 10 | 1798 | 1013 | 8.88 | 383.8 | 404.7 | 20.9 | 2.25 | 3.9 | 0.44 | 94.8 | 0.56 |
| 10 | 2177 | 1227 | 10.76 | 368.7 | 394.4 | 25.7 | 5.19 | 5.9 | 1.15 | 93.5 | 1.24 |
| 11 | 1409 | 967 | 8.48 | 383.1 | 409.7 | 26.6 | 0.96 | 3.9 | 0.12 | 93.5 | 0.31 |
| 11 | 1801 | 1236 | 10.84 | 373.2 | 403.6 | 30.4 | 2.64 | 5.7 | 0.48 | 92.5 | 0.54 |
| 11 | 2163 | 1484 | 13.01 | 366.7 | 399.0 | 32.3 | 3.10 | 7.3 | 0.74 | 91.9 | 0.77 |
| 12 | 1409 | 1186 | 10.40 | 367.8 | 398.5 | 30.6 | 4.05 | 4.5 | 0.61 | 92.3 | 1.06 |
| 12 | 1797 | 1512 | 13.26 | 368.0 | 401.4 | 33.4 | 6.46 | 6.3 | 1.19 | 91.7 | 1.27 |
| 12 | 2157 | 1815 | 15.92 | 369.6 | 412.3 | 42.8 | 5.71 | 9.7 | 1.25 | 89.6 | 1.27 |
| 13 | 1408 | 1520 | 13.33 | 371.0 | 386.7 | 15.7 | 2.48 | 2.3 | 0.36 | 95.9 | 0.63 |
| 13 | 1789 | 1930 | 16.92 | 382.6 | 409.3 | 26.7 | 1.17 | 5.0 | 0.21 | 93.5 | 0.37 |
| 13 | 2156 | 2326 | 20.40 | 365.8 | 406.7 | 40.9 | 6.20 | 9.2 | 1.47 | 89.9 | 1.61 |
| 14 | 1400 | 1853 | 16.29 | 387.3 | 408.9 | 21.6 | 1.99 | 3.2 | 0.30 | 94.7 | 0.41 |
| 14 | 1803 | 2387 | 20.93 | 377.6 | 412.2 | 34.6 | 1.04 | 6.5 | 0.19 | 91.6 | 0.20 |
| 14 | 2157 | 2855 | 25.04 | 353.6 | 402.0 | 48.4 | 2.98 | 10.9 | 0.81 | 88.0 | 0.90 |
| 15 | 1405 | 2264 | 19.88 | 355.9 | 395.9 | 40.0 | 4.74 | 5.9 | 0.69 | 89.9 | 1.36 |
| 15 | 1795 | 2893 | 25.38 | 346.2 | 398.8 | 52.6 | 3.24 | 9.9 | 0.56 | 86.8 | 0.65 |
| 15 | 2146 | 3459 | 30.34 | 339.2 | 409.3 | 70.1 | 7.54 | 15.8 | 1.98 | 82.9 | 2.18 |
| 16 | 1401 | 2771 | 24.30 | 357.6 | 412.2 | 54.5 | 3.22 | 8.0 | 0.46 | 86.8 | 0.55 |
| 16 | 1799 | 3556 | 31.18 | 330.1 | 405.9 | 75.7 | 9.53 | 14.3 | 1.25 | 81.7 | 2.23 |
| 16 | 2136 | 4222 | 37.03 | 327.2 | 409.6 | 82.4 | 1.47 | 18.4 | 0.32 | 79.9 | 0.45 |

## *A5.3 Additional Torque Loss Charts (100, 250, 400Nm)*

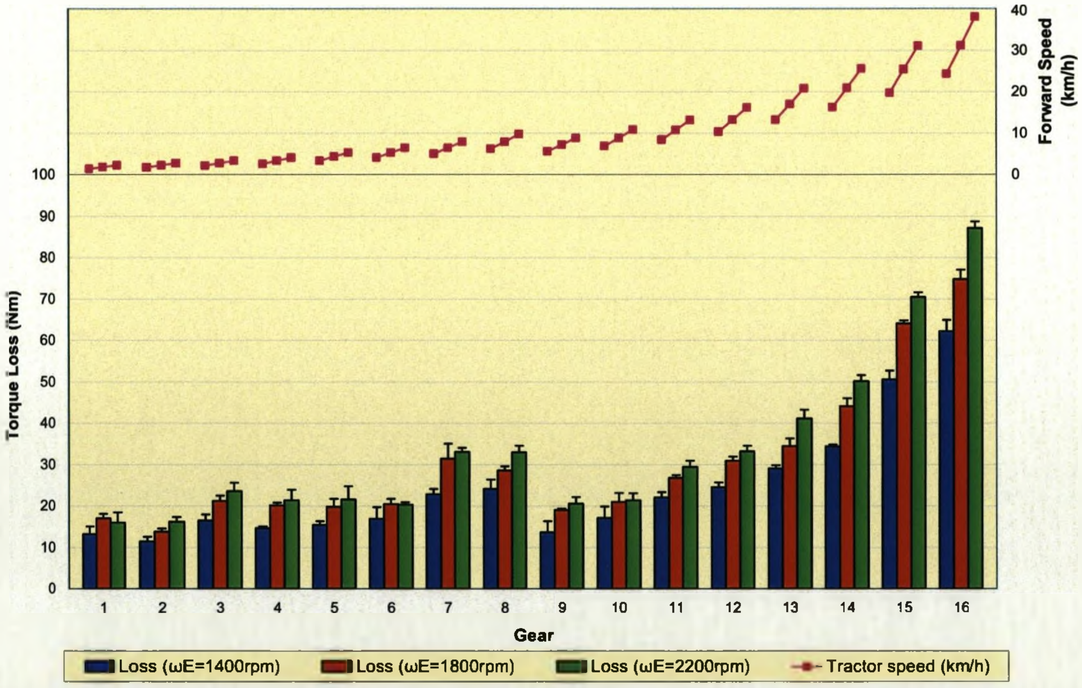


*Figure A5.1 – Torque loss at varying transmission input speeds ($\omega_E$) in each gear with 100Nm flywheel torque*

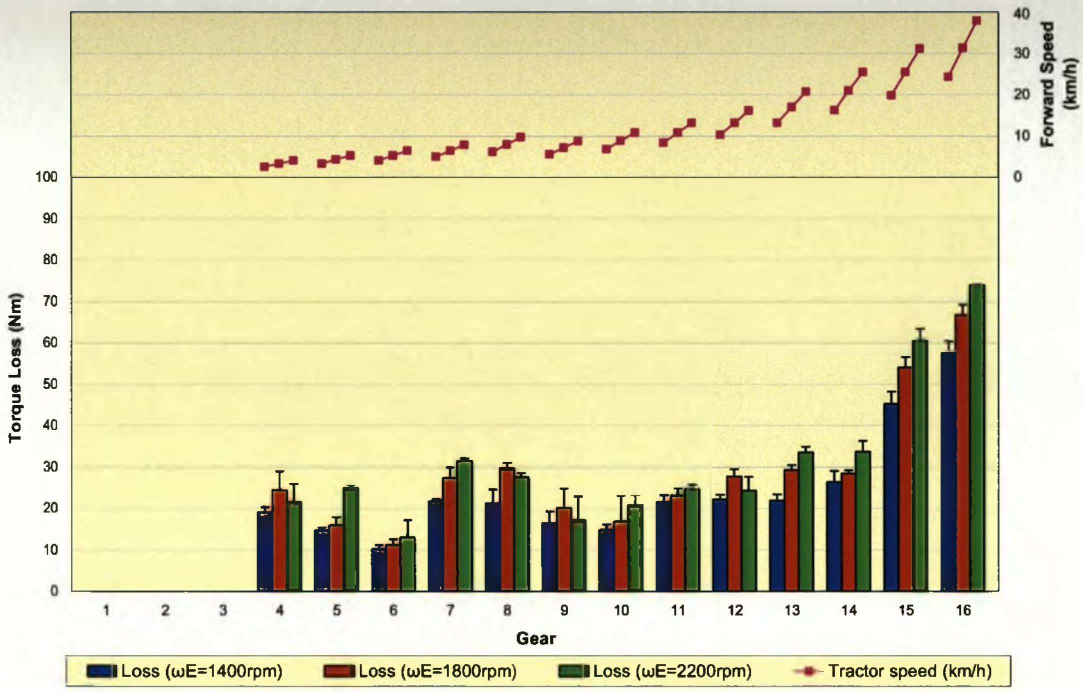


*Figure A5.2 – Torque loss at varying transmission input speeds ($\omega_E$) in each gear with 250Nm flywheel torque*

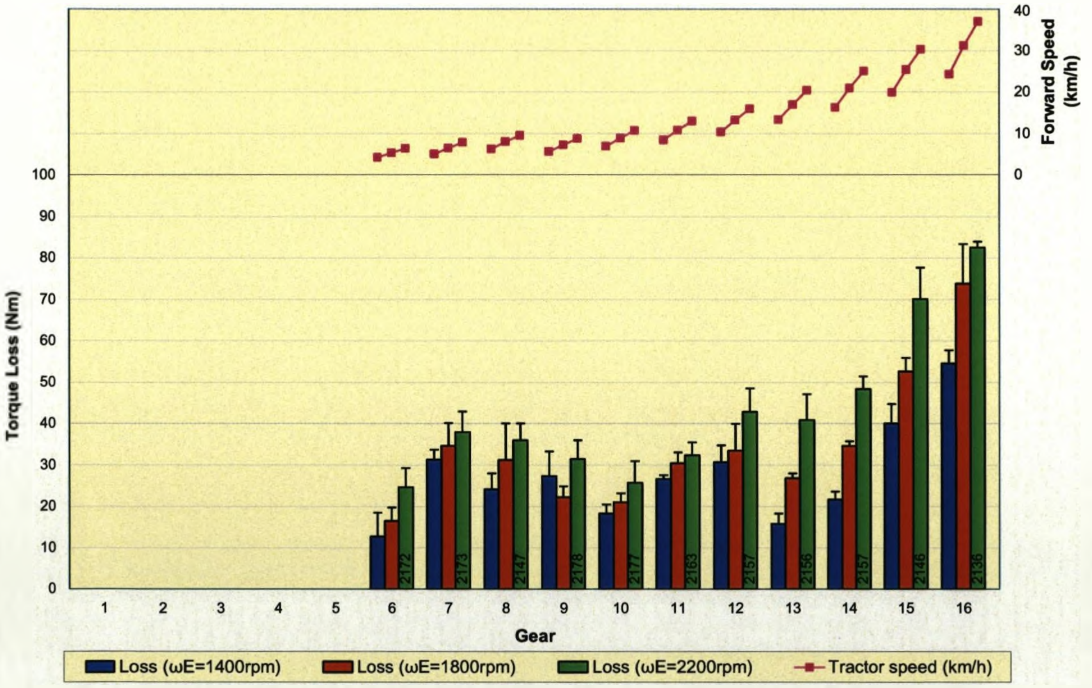


*Figure A5.3 – Torque loss at varying transmission input speeds ($\omega_E$) in each gear with 400Nm flywheel torque*

## A5.4 No-load Differences

*Table A5.6 – Statistical Differences between no-load experiments on the axle dynamometer and on the tractor wheels*

| G | Tractor on Axle Dynamometer | | | Tractor on (raised) wheels | | | combined | $t_{CALC}$ | difference at 5% |
|---|---|---|---|---|---|---|---|---|---|
| | $\omega_E$ | $T_{LOSS}$ | S.E.M ($T_{LOSS}$) | $\omega_E$ | $T_{LOSS}$ | S.E.M ($T_{LOSS}$) | S.E.M. | | level (>2.776)? |
| 1 | 1400 | 5.94 | 0.29 | 1400 | 6.4 | 0.35 | 0.46 | 0.98 | NO |
| 1 | 1801 | 8.4 | 0.09 | 1802 | 8.7 | 0.17 | 0.19 | 1.95 | NO |
| 1 | 2197 | 8.9 | 0.09 | 2202 | 9.3 | 0.47 | 0.48 | 0.77 | NO |
| 2 | 1402 | 5.9 | 0.22 | 1403 | 6.3 | 0.38 | 0.44 | 0.94 | NO |
| 2 | 1802 | 8.3 | 0.10 | 1803 | 8.9 | 0.54 | 0.55 | 1.01 | NO |
| 2 | 2203 | 9.0 | 0.87 | 2200 | 9.3 | 0.51 | 1.01 | 0.21 | NO |
| 3 | 1398 | 8.4 | 0.09 | 1401 | 9.8 | 0.49 | 0.50 | 2.71 | NO |
| 3 | 1804 | 10.7 | 0.07 | 1800 | 11.8 | 0.50 | 0.51 | 2.20 | NO |
| 3 | 2194 | 10.5 | 0.50 | 2198 | 12.5 | 0.57 | 0.76 | 2.66 | NO |
| 4 | 1397 | 8.3 | 0.13 | 1401 | 9.8 | 0.55 | 0.56 | 2.65 | NO |
| 4 | 1798 | 10.5 | 0.42 | 1799 | 12.7 | 0.70 | 0.82 | 2.64 | NO |
| 4 | 2197 | 11.1 | 0.66 | 2201 | 13.3 | 0.62 | 0.91 | 2.37 | NO |
| 5 | 1398 | 9.5 | 0.45 | 1400 | 10.7 | 0.47 | 0.65 | 1.84 | NO |
| 5 | 1800 | 11.8 | 0.40 | 1802 | 13.1 | 0.66 | 0.77 | 1.69 | NO |
| 5 | 2196 | 12.3 | 0.28 | 2198 | 13.9 | 0.70 | 0.76 | 2.13 | NO |
| 6 | 1401 | 10.6 | 0.11 | 1400 | 11.9 | 0.62 | 0.63 | 2.08 | NO |
| 6 | 1802 | 12.3 | 0.52 | 1798 | 13.6 | 0.22 | 0.57 | 2.45 | NO |
| 6 | 2197 | 12.4 | 0.70 | 2203 | 14.1 | 0.93 | 1.16 | 1.49 | NO |
| 7 | 1397 | 17.0 | 0.35 | 1397 | 18.6 | 0.57 | 0.67 | 2.51 | NO |
| 7 | 1799 | 18.5 | 1.72 | 1800 | 23.7 | 0.85 | 1.92 | 2.70 | NO |
| 7 | 2199 | 18.4 | 0.32 | 2202 | 21.2 | 0.98 | 1.03 | 2.68 | NO |
| 8 | 1396 | 17.0 | 0.39 | 1402 | 19.4 | 0.78 | 0.87 | 2.76 | NO |
| 8 | 1798 | 20.8 | 1.72 | 1799 | 24.8 | 3.05 | 3.50 | 1.14 | NO |
| 8 | 2199 | 20.2 | 0.85 | 2201 | 23.3 | 1.34 | 1.59 | 1.93 | NO |
| 9 | 1397 | 8.5 | 0.22 | 1401 | 8.9 | 0.16 | 0.27 | 1.62 | NO |
| 9 | 1797 | 10.9 | 0.19 | 1801 | 11.1 | 0.09 | 0.21 | 1.21 | NO |
| 9 | 2200 | 11.0 | 0.30 | 2199 | 12.0 | 0.29 | 0.42 | 2.23 | NO |
| 10 | 1397 | 9.3 | 0.23 | 1400 | 10.0 | 0.40 | 0.46 | 1.39 | NO |
| 10 | 1797 | 11.7 | 0.36 | 1801 | 12.2 | 0.52 | 0.63 | 0.68 | NO |
| 10 | 2197 | 12.2 | 0.47 | 2200 | 13.8 | 0.36 | 0.59 | 2.73 | NO |
| 11 | 1398 | 13.7 | 0.43 | 1398 | 16.0 | 0.26 | 0.50 | 4.54 | YES |
| 11 | 1800 | 15.4 | 0.37 | 1800 | 19.9 | 1.66 | 1.71 | 2.64 | NO |
| 11 | 2198 | 15.9 | 0.39 | 2201 | 19.3 | 2.62 | 2.65 | 1.28 | NO |
| 12 | 1399 | 17.0 | 0.34 | 1403 | 19.4 | 0.20 | 0.39 | 6.02 | YES |
| 12 | 1799 | 18.6 | 1.79 | 1803 | 21.4 | 0.74 | 1.93 | 1.46 | NO |
| 12 | 2194 | 19.3 | 0.84 | 2202 | 22.0 | 1.25 | 1.50 | 1.84 | NO |
| 13 | 1404 | 21.6 | 0.64 | 1397 | 23.1 | 1.20 | 1.36 | 1.11 | NO |
| 13 | 1797 | 25.8 | 1.56 | 1800 | 27.3 | 2.65 | 3.08 | 0.51 | NO |
| 13 | 2202 | 32.2 | 0.36 | 2201 | 34.5 | 0.76 | 0.84 | 2.69 | NO |
| 14 | 1394 | 27.7 | 1.76 | 1402 | 29.8 | 2.30 | 2.89 | 0.72 | NO |
| 14 | 1796 | 36.8 | 0.73 | 1801 | 36.8 | 3.15 | 3.23 | 0.01 | NO |
| 14 | 2198 | 42.1 | 0.86 | 2202 | 42.8 | 4.07 | 4.16 | 0.16 | NO |
| 15 | 1402 | 41.7 | 2.10 | 1400 | 44.7 | 4.58 | 5.04 | 0.59 | NO |
| 15 | 1797 | 52.5 | 0.63 | 1802 | 54.0 | 0.64 | 0.90 | 1.63 | NO |
| 15 | 2193 | 62.6 | 1.85 | 2203 | 65.2 | 1.34 | 2.28 | 1.13 | NO |
| 16 | 1401 | 53.1 | 0.83 | 1403 | 54.3 | 1.36 | 1.59 | 0.73 | NO |
| 16 | 1796 | 64.4 | 3.02 | 1802 | 68.1 | 2.81 | 4.12 | 0.92 | NO |
| 16 | 2199 | 80.1 | 1.24 | 2201 | 82.6 | 2.54 | 2.82 | 0.90 | NO |

## A5.5 Driveline Loss Model Statistics

Regression statistics for the driveline models are presented in Tables A5.7 to A5.16.

*Table A5.7 – Model 1 regression statistics*

***** **Regression Analysis Model 1** *****
Response variate: $T_F$
Fitted terms: $T_{EF}$, $\omega_E$, $r_d$

*** Summary of analysis ***

| | d.f. | s.s. | m.s. | v.r. | F pr. |
|---|---|---|---|---|---|
| Regression | 3 | 25682433. | 8560810.9 | 42179.26 | <.001 |
| Residual | 501 | 101684. | 203.0 | | |
| Total | 504 | 25784117. | 51159.0 | | |

Percentage variance accounted for 99.0
Standard error of observations is estimated to be 14.2

*** Estimates of parameters ***

| | estimate | s.e. | t(501) | t pr. |
|---|---|---|---|---|
| $T_{EF}$ | 1.00525 | 0.00455 | 220.94 | <.001 |
| $\omega_E$ | 0.021422 | 0.000692 | 30.98 | <.001 |
| $r_d$ | -0.13501 | 0.00911 | -14.82 | <.001 |

*Table A5.8 – Model 2 regression statistics*

***** **Regression Analysis Model 2** *****
Response variate: $T_F$
Fitted terms: $T_{EF}$, $\omega_E$, $r_d$, G

*** Summary of analysis ***

| | d.f. | s.s. | m.s. | v.r. | F pr. |
|---|---|---|---|---|---|
| Regression | 4 | 25714237. | 6428559.3 | 45997.35 | <.001 |
| Residual | 500 | 69880. | 139.8 | | |
| Total | 504 | 25784117. | 51159.0 | | |

Percentage variance accounted for 99.3
Standard error of observations is estimated to be 11.8

*** Estimates of parameters ***

| | estimate | s.e. | t(500) | t pr. |
|---|---|---|---|---|
| $T_{EF}$ | 0.99716 | 0.00381 | 261.49 | <.001 |
| $\omega_E$ | 0.00281 | 0.00136 | 2.07 | 0.039 |
| $r_d$ | -0.0070 | 0.0114 | **-0.62** | 0.539 |
| G | 2.668 | 0.177 | 15.09 | <.001 |

*Table A5.9 – Model 3 regression statistics*

******* Regression Analysis Model 3 *******

Response variate: $T_F$
Fitted terms: $T_{EF}$, $\omega_E$, $\omega_T$

*** Summary of analysis ***

| | d.f. | s.s. | m.s. | v.r. | F pr. |
|---|---|---|---|---|---|
| Regression | 3 | 25756737. | 8585578.97 | 157098.67 | <.001 |
| Residual | 501 | 27380. | 54.65 | | |
| Total | 504 | 25784117. | 51158.96 | | |

Percentage variance accounted for 99.7
Standard error of observations is estimated to be 7.39

*** Estimates of parameters ***

| | estimate | s.e. | t(501) | t pr. |
|---|---|---|---|---|
| $T_{EF}$ | 1.00926 | 0.00233 | 433.81 | <.001 |
| $\omega_E$ | 0.003390 | 0.000347 | 9.76 | <.001 |
| $\omega_T$ | 0.015616 | 0.000335 | 46.64 | <.001 |

*Table A5.10 – Model 4 regression statistics*

******* Regression Analysis Model 4 *******

Response variate: $T_F$
Fitted terms: $T_{EF}$, $\omega_E$, $\omega_T$, $G_m$

*** Summary of analysis ***

| | d.f. | s.s. | m.s. | v.r. | F pr. |
|---|---|---|---|---|---|
| Regression | 4 | 25761357. | 6440339.20 | 141482.37 | <.001 |
| Residual | 500 | 22760. | 45.52 | | |
| Total | 504 | 25784117. | 51158.96 | | |

Percentage variance accounted for 99.8
Standard error of observations is estimated to be 6.75

*** Estimates of parameters ***

| | estimate | s.e. | t(500) | t pr. |
|---|---|---|---|---|
| $T_{EF}$ | 1.00948 | 0.00212 | 475.41 | <.001 |
| $\omega_E$ | -0.000750 | 0.000519 | **-1.44** | 0.149 |
| $\omega_T$ | 0.016636 | 0.000322 | 51.68 | <.001 |
| $G_m$ | 1.881 | 0.187 | 10.07 | <.001 |

*Table A5.11 – Model 5 regression statistics*

******* Regression Analysis Model 5 *******

Response variate: $T_F$
Fitted terms: $T_{EF}$, $\omega_T$

*** Summary of analysis ***

| | d.f. | s.s. | m.s. | v.r. | F pr. |
|---|---|---|---|---|---|
| Regression | 2 | 25751530. | 12875764.95 | 198349.50 | <.001 |
| Residual | 502 | 32587. | 64.91 | | |
| Total | 504 | 25784117. | 51158.96 | | |

Percentage variance accounted for 99.7
Standard error of observations is estimated to be 8.06

*** Estimates of parameters ***

| | estimate | s.e. | t(502) | t pr. |
|---|---|---|---|---|
| $T_{EF}$ | 1.02000 | 0.00223 | 456.60 | <.001 |
| $\omega_T$ | 0.017812 | 0.000270 | 65.91 | <.001 |

*Table A5.12 – Model 6 regression statistics*

******* Regression Analysis Model 6 *******

Response variate: $T_F$
Fitted terms: $T_{EF}$, $\omega_T$, $r_d$

*** Summary of analysis ***

| | d.f. | s.s. | m.s. | v.r. | F pr. |
|---|---|---|---|---|---|
| Regression | 3 | 25758646. | 8586215.40 | 168887.27 | <.001 |
| Residual | 501 | 25471. | 50.84 | | |
| Total | 504 | 25784117. | 51158.96 | | |

Percentage variance accounted for 99.7
Standard error of observations is estimated to be 7.13

*** Estimates of parameters ***

| | estimate | s.e. | t(501) | t pr. |
|---|---|---|---|---|
| $T_{EF}$ | 1.01178 | 0.00210 | 482.86 | <.001 |
| $\omega_T$ | 0.017540 | 0.000240 | 73.00 | <.001 |
| $r_d$ | 0.03883 | 0.00328 | 11.83 | <.001 |

*Table A5.13 – Model 7 regression statistics*

******* Regression Analysis Model 7 *******
Response variate: $T_F$
Fitted terms: $T_{EF}$, $\omega_T$, G

*** Summary of analysis ***

| | d.f. | s.s. | m.s | v.r. | F pr. |
|---|---|---|---|---|---|
| Regression | 3 | 25751681. | 8583893.75 | 132586.08 | <.001 |
| Residual | 501 | 32436. | 64.74 | | |
| Total | 504 | 25784117. | 51158.96 | | |

Percentage variance accounted for 99.7
Standard error of observations is estimated to be 8.05

*** Estimates of parameters ***

| | estimate | s.e. | t(501) | t pr. |
|---|---|---|---|---|
| $T_{EF}$ | 1.01791 | 0.00262 | 389.00 | <.001 |
| $\omega_T$ | 0.016846 | 0.000688 | 24.50 | <.001 |
| G | 0.193 | 0.126 | **1.53** | 0.127 |

*Table A5.14 – Model 8 regression statistics*

******* Regression Analysis Model 8 *******
Response variate: $T_F$
Fitted terms: $T_{EF}$, $\omega_T$, $G_m$

*** Summary of analysis ***

| | d.f. | s.s. | m.s. | v.r. | F pr. |
|---|---|---|---|---|---|
| Regression | 3 | 25761501. | 8587167.11 | 188430.57 | <.001 |
| Residual | 501 | 22832. | 45.57 | | |
| Total | 504 | 25784333. | 51159.39 | | |

Percentage variance accounted for 99.8
Standard error of observations is estimated to be 6.75

*** Estimates of parameters ***

| | estimate | s.e. | t(501) | t pr. |
|---|---|---|---|---|
| $T_{EF}$ | 1.00860 | 0.00203 | 497.17 | <.001 |
| $\omega_T$ | 0.016359 | 0.000248 | 66.00 | <.001 |
| $G_m$ | 1.660 | 0.114 | 14.55 | <.001 |

*Table A5.15 – Model 9 regression statistics*

******* Regression Analysis Model 9 *******

Response variate: $T_F$
Fitted terms: $T_{EF}$, $\omega_T$, $r_d$, $G_m$

*** Summary of analysis ***

| | d.f. | s.s. | m.s. | v.r. | F pr. |
|---|---|---|---|---|---|
| Regression | 4 | 25761417. | 6440354.15 | 141855.46 | <.001 |
| Residual | 500 | 22700. | 45.40 | | |
| Total | 504 | 25784117. | 51158.96 | | |

Percentage variance accounted for 99.8
Standard error of observations is estimated to be 6.74

*** Estimates of parameters ***

| | estimate | s.e. | t(500) | t pr. |
|---|---|---|---|---|
| $T_{EF}$ | 1.00842 | 0.00203 | 497.64 | <.001 |
| $\omega_T$ | 0.016504 | 0.000263 | 62.76 | <.001 |
| $r_d$ | 0.00907 | 0.00491 | **1.85** | 0.065 |
| $G_m$ | 1.409 | 0.180 | 7.81 | <.001 |

*Table A5.16 – Model 10 regression statistics*

******* Regression Analysis Model 10 *******

Response variate: $T_F$
Fitted terms: $T_{EF}$, $\omega_T$, G, $G_m$

*** Summary of analysis ***

| | d.f. | s.s. | m.s. | v.r. | F pr. |
|---|---|---|---|---|---|
| Regression | 4 | 25763524. | 6440881.03 | 156386.18 | <.001 |
| Residual | 500 | 20593. | 41.19 | | |
| Total | 504 | 25784117. | 51158.96 | | |

Percentage variance accounted for 99.8
Standard error of observations is estimated to be 6.42

*** Estimates of parameters ***

| | estimate | s.e. | t(500) | t pr. |
|---|---|---|---|---|
| $T_{EF}$ | 1.01465 | 0.00210 | 484.11 | <.001 |
| $\omega_T$ | 0.020309 | 0.000585 | 34.71 | <.001 |
| G | -0.882 | 0.119 | -7.41 | <.001 |
| $G_m$ | 2.173 | 0.128 | 16.96 | <.001 |

## *A5.6 Driveline Inertia – Wheel Modelling Data*

The dish, rim and tyre model from AutoCAD is shown in Figure A5.4:

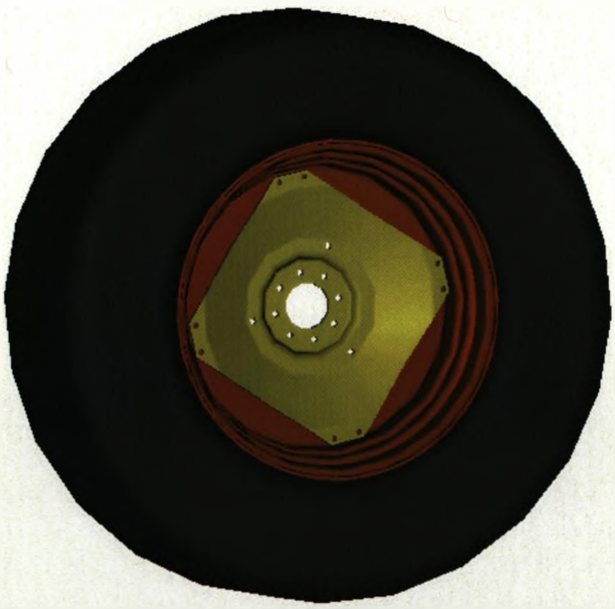

*Figure A5.4 – AutoCAD 3-dimensional wheel and tyre model*

The three AutoCAD screen grabs (see Figures A5.5 to A5.7) show the part mass properties for the wheel dish and rim; and for the tyre, the units are kg and mm.

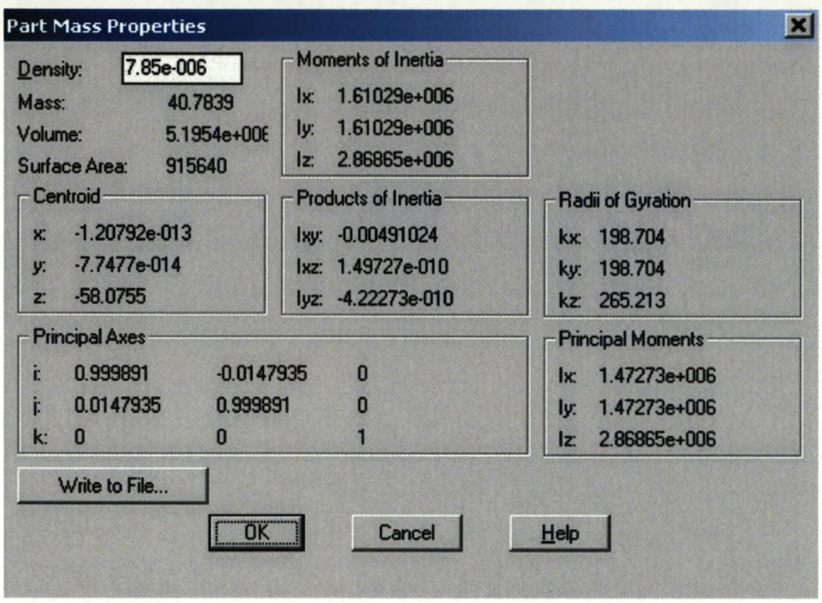


*Figure A5.5 – Mass properties for the rear wheel dish*

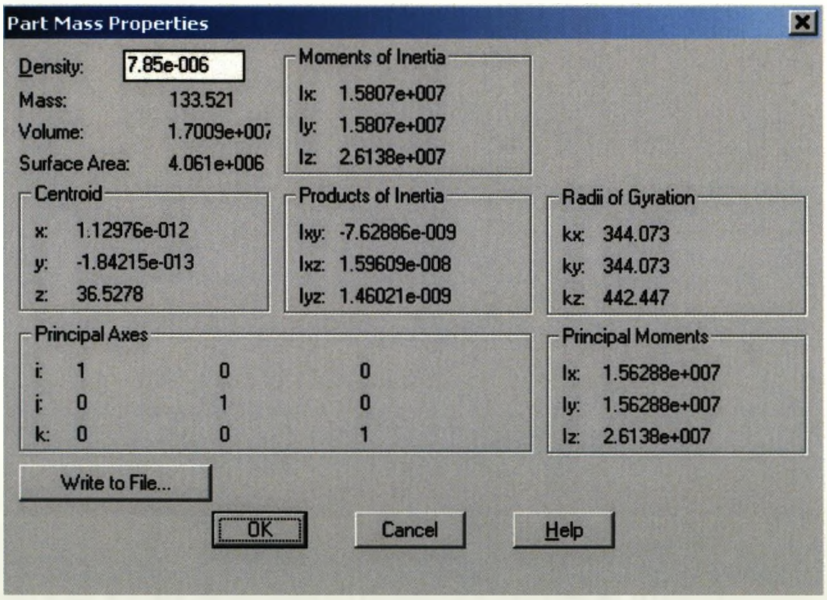


*Figure A5.6 – Mass properties for the rear wheel rim*

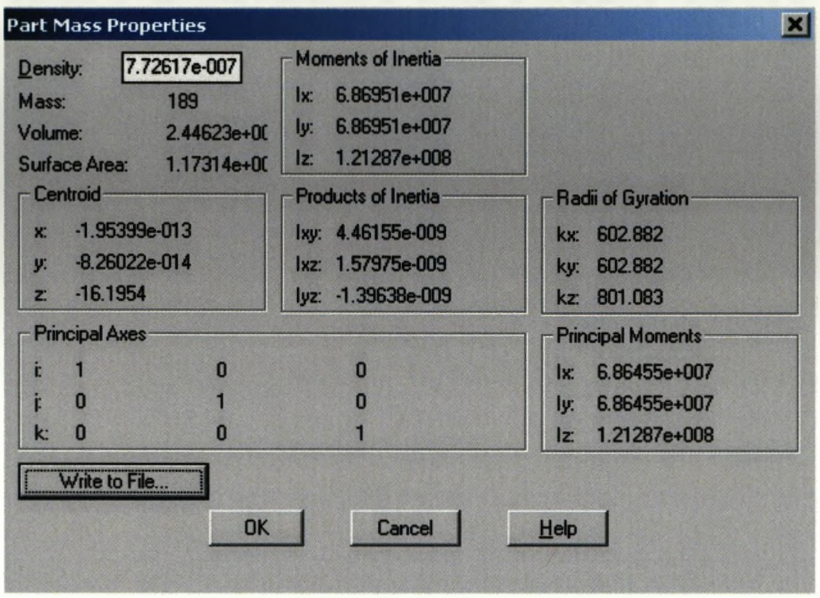


*Figure A5.7 – Mass properties for the rear tyre (600/65 R38)*

## A5.7 Driveline Inertia Equations

The five gears (9 to 13) each allow the generation of one equation of motion:

$$0.12 = I_{TA}\left(\frac{1}{1}\right)^2 + I_{TB}\left(\frac{1}{2.3492}\right)^2 + I_{TC}\left(\frac{1}{2.1773}\right)^2 + I_D\left(\frac{1}{11.3655}\right)^2 + 2I_W\left(\frac{1}{76.7171}\right)^2$$

*(gear no.9) Equation A5-1*

$$0.19 = I_{TA}\left(\frac{1}{1}\right)^2 + I_{TB}\left(\frac{1}{2.3492}\right)^2 + I_{TC}\left(\frac{1}{1.7749}\right)^2 + I_D\left(\frac{1}{9.2652}\right)^2 + 2I_W\left(\frac{1}{62.5401}\right)^2$$

*(gear no.10) Equation A5-2*

$$0.25 = I_{TA}\left(\frac{1}{0.6695}\right)^2 + I_{TB}\left(\frac{1}{1.5727}\right)^2 + I_{TC}\left(\frac{1}{1.4577}\right)^2 + I_D\left(\frac{1}{7.6090}\right)^2 + 2I_W\left(\frac{1}{51.3607}\right)^2$$

*(gear no.11) Equation A5-3*

$$0.36 = I_{TA}\left(\frac{1}{0.6695}\right)^2 + I_{TB}\left(\frac{1}{1.5727}\right)^2 + I_{TC}\left(\frac{1}{1.1883}\right)^2 + I_D\left(\frac{1}{6.2029}\right)^2 + 2I_W\left(\frac{1}{41.8695}\right)^2$$

*(gear no.12) Equation A5-4*

$$0.55 = I_{TA}\left(\frac{1}{1}\right)^2 + I_{TB}\left(\frac{1}{1}\right)^2 + I_{TC}\left(\frac{1}{0.9268}\right)^2 + I_D\left(\frac{1}{4.8380}\right)^2 + 2I_W\left(\frac{1}{32.6568}\right)^2$$

*(gear no.13) Equation A5-5*

By adding in the known value of $I_W$ (150.3 kg.m$^2$) these equations become:

$$0.0690 = I_{TA}\left(\frac{1}{1}\right)^2 + I_{TB}\left(\frac{1}{2.3492}\right)^2 + I_{TC}\left(\frac{1}{2.1773}\right)^2 + I_D\left(\frac{1}{11.3655}\right)^2$$

*(gear no.9) Equation A5-6*

$$0.1133 = I_{TA}\left(\frac{1}{1}\right)^2 + I_{TB}\left(\frac{1}{2.3492}\right)^2 + I_{TC}\left(\frac{1}{1.7749}\right)^2 + I_D\left(\frac{1}{9.2652}\right)^2$$

*(gear no.10) Equation A5-7*

$$0.1362 = I_{TA}\left(\frac{1}{0.6695}\right)^2 + I_{TB}\left(\frac{1}{1.5727}\right)^2 + I_{TC}\left(\frac{1}{1.4577}\right)^2 + I_D\left(\frac{1}{7.6090}\right)^2$$

*(gear no.11) Equation A5-8*

$$0.1889 = I_{TA}\left(\frac{1}{0.6695}\right)^2 + I_{TB}\left(\frac{1}{1.5727}\right)^2 + I_{TC}\left(\frac{1}{1.1883}\right)^2 + I_D\left(\frac{1}{6.2029}\right)^2$$

*(gear no.12) Equation A5-9*

$$0.2686 = I_{TA}\left(\frac{1}{1}\right)^2 + I_{TB}\left(\frac{1}{1}\right)^2 + I_{TC}\left(\frac{1}{0.9268}\right)^2 + I_D\left(\frac{1}{4.8380}\right)^2$$

*(gear no.13) Equation A5-10*

# A6 Customer Satisfaction Survey Information

## A6.1 Interviewee A Transcript

Name

Brian & Mick Chappell

Address

Quarry Farm, Tickhill Road, Loversall, Doncaster. DN11 9DH

Interview date

Friday 3rd September 2004

Business details

Farmed as a partnership by two brothers, in addition to the 26ha (65 acres) they own and farm themselves, the major business interest is agricultural contracting, specialising in potato work where they prepare ground for (plough, power harrow and destone), and plant 325ha a year and harvest over 400ha a year. In addition to potato work, they undertake 1200ha of cereal cultivation and 7300ha of spraying a year.

Non-farming activity

None

Tractor fleet profile[1]

- Case MX120 (new 1999). 90kW with 16/12 semi-powershift transmission (4 powershift, 4/3 mechanical ranges). 800 hours p.a.
- Case CS150 (new 2002). 110kW with 24/24 semi-powershift transmission (4 powershift, 6 mechanical ranges + creeper gears). 800 hours p.a.
- Case CVX130 (new 2000). 96kW with CVT. 2000 hours p.a.
- Case CVX130 (2nd hand 2003 - 18 months old). 96kW with CVT. 2000 hours p.a.
- Case CVX1155 (new 2004). 115kW with CVT. 1500 hours p.a. (est).
- Case CVX1190 (new 2004). 140kW with CVT. 1500 hours p.a. (est).

[1] Engine power from OECD tests, or brochures if no OECD data exists, brochures may quote a bare engine rather than P.T.O power.

- Case MX285 (new 2003). 210kW with 18/4 full powershift transmission. 1000 hours p.a.

Driver policy

Five employees each have their own tractor which they drive, those being the four CVX tractors and the Magnum. The MX and CS tractors are driven as required by two other employees, who mainly drive the self propelled sprayers the business also operates.

Tractor replacement policy

Generally the business tries to operate new machines which are replaced every 3 years. However the last two years has seen a large growth in the potato work which has meant additional tractors, more hours per year and an increase in tractor size, therefore earlier replacement than normal has been necessary.

Reasons for fleet choices

The business runs mainly CVX tractors for a number of reasons, the most obvious being they are ideal for potato work with the variable forward speed ability. They have found a reduction in fuel consumption with the CVX tractors. They are also very easy to drive and not as tiring after a long day. The MX285 would have been a variable transmission tractor, but the CVX range does not extend to that tractor size. They demonstrated and considered a Fendt but wanted to remain with a Case tractor due to dealer support.

Next tractor

The next tractor will be a CVX tractor, the MX120 and the CS150 will both be replaced with CVX tractors. The MX285 will be replaced with a similar machine if no Case variable transmission model is available at the time of replacement.

Specific CVT tasks

The main work of the CVX (and all the other tractors) is undertaking potato ground preparation, planting and harvesting. They have also been used with a cereal drill and trailers.

User benefits of a CVT transmission

The ability to perfectly match forward speed to the conditions, independently of engine speed is very useful, this showed up during the 2003 potato harvest where very dry conditions meant very little soil was passed up into the harvester. By getting the forward speed of the harvester correct, the operators were able to keep the beds full of potatoes, thus preventing excessive damage. During potato ground preparation they had a CVX1155 working alongside a Case MXM155, both running destoners. During the day the MXM used 170 litres of diesel whereas the CVX used 140 litres. The tractors have proved better with a cereal drill, headlands turns and are far easier on trailer work. Mr Chappell stated that they have not yet found a job where a CVX is not better than a conventional transmission.

Potential to replicate these benefits with a conventional transmission

Not really, a variable speed P.T.O. would help with the potato issue, but the point about having the flexibility to adjust both P.T.O and forward speeds to maximise output whilst minimising damage would be difficult to replicate.

Detrimental aspects of the CVT transmission

There have been a number of occurrences during a wet and late potato harvest during 2002 when CVX tractors with laden trailers got stuck. As the tractor got bogged down, the controller seemed to slow the transmission down and speed the engine up, which of made the problem worse and the tractor was stuck before the operator could react.

Acceleration of the tractor

The CVX is marginally quicker at accelerating than the other tractors.

Ease of use of tractor transmissions (all types)

For their operators (all of whom are experienced) the CVX tractors are by far the easiest to operate, followed by the full powershift MX285. The semi-powershift MX120 and CS150 are the most demanding to operate. Mr Chappell stated that even a beginner tractor driver would find the CVX easier to drive, provided someone explained how the basic controls work. It was discussed that there is a big difference between driving a CVX tractor, and driving it to its optimum.

Views on the control systems for their tractors

The multi function joystick was a major plus point for the CVX tractor. Having the controls to hand makes the tractor far easier and less tiring to drive.

The brothers had a Fendt vario on demonstration and although it looked more complicated they were able to operate it due to the experience with the Case tractors. Mr Chappell doubted whether it would have been so easy if they had not had experience with a CVX.

Potential improvements to any tractor control systems

He felt there were too many buttons spread around CVX tractor. More of the less used functions could be programmed into the pillar display.

Additional desirable features on CVT tractor

An easy way of automatically reducing forward speed during a headland turn, without the need to fully programme complicated headland management systems.

Any unimportant or unused features

None were identified

Do they know who manufactures the transmission

Identified ZF as the transmission manufacturer.

Price premium paid for the CVT tractor

For the CVX1155 a premium of around £6,000 was paid over a semi-powershift tractor.

Does the CVT tractor provide value for money

Definitely value in terms of extra land area covered. In addition, the operators were happier and willing to work longer hours on these tractors. The experience of buying the 2nd hand CVX showed Mr Chappell that they hold their value as the premium over a comparable semi-powershift tractor was still £4,000.

Maximum price premium prepared to pay for a CVT

They would be willing to pay up to £8,000 over a comparable powershift transmission, however a £5,000 premium would be more acceptable.

Desirable future tractor powertrain developments

None identified

Dealer service

Excellent dealer backup and service, they are beginning to get a few minor issues with the older tractors for example seat switch failures around 5,000 hours which have all be sorted very quickly. Mr Chappell made the point they have a lot of tractors from them as a result of good service, not the other way round.

Precision farming techniques used or considered

None used at present. They have investigated GPS controlled steering for spraying, but were put off by the cost for the required level of accuracy of at least £2,500. Fuel maps would be of interest, but would require an improvement in the accuracy of the fuel metering device which it was felt was at best 90% accurate at present.

## A6.2 Interviewee B Transcript

Name

Peter Saville

Address

Home Farm, Perlethorpe, Newark. NG22 9EQ

Interview date

Tuesday 16th November 2004

Business details

2065ha (5100 acre) estate privately owned, run by a farm manager. Split into three blocks of land, the main farmstead being situated on 900ha of sandy soil, ideal for growing salad & crisping potatoes, and a variety of vegetables, including carrots and onions. Barley and sugar beet are also grown in rotation. The two other units (810ha and 355ha) are heavy clay land growing wheat, rape and beans with minimum tillage techniques.

Non-farming activity

None

Tractor fleet profile[1]

- Case CVX1170 (new 2003). 125kW with CVT. 1500 hours p.a.
- Case CVX1155 (new 2004). 115kW with CVT. 2000 hours p.a. (est).
- Case CVX1145 (new 2003). 108kW with CVT. 2000 hours p.a.
- Case CVX1135 (new 2004). 101kW with CVT. 2000 hours p.a. (est).
- Case CVX1135 (new 2004). 101kW with CVT. 2000 hours p.a. (est).
- Case CVX1135 (new 2004). 101kW with CVT. 2000 hours per p.a. (est).
- Case MX270 (new 2002). 200kW with 18/4 full powershift transmission. 800 hours p.a.
- Case MX135 (new 2001). 96kW with 16/12 semi-powershift transmission (4 powershift, 4/3 mechanical ranges). 2000 hours p.a.
- Case MX135 (new 2001). 96kW with 16/12 semi-powershift transmission (4 powershift, 4/3 mechanical ranges). 2000 hours p.a.

[1] Engine power from OECD tests, or brochures if no OECD data exists, brochures may quote a bare engine rather than P.T.O power.

- Case MX135 (new 2001). 96kW with 16/12 semi-powershift transmission (4 powershift, 4/3 mechanical ranges). 1500 hours p.a.
- Case MX135 (new 2002). 96kW with 16/12 semi-powershift transmission (4 powershift, 4/3 mechanical ranges). 1500 hours p.a.
- John Deere 8110 (new 1998). 145kW with 16/5 full powershift transmission. 800 hours p.a.

Driver policy

The general policy is each employee is responsible for one of the tractors. This works generally quite well, apart from when drivers move onto combines or the MX270 (only used for heavy land cultivation). At these times their tractors are driven by summer migrant workers.

Tractor replacement policy

Tractors have been historically replaced every three to four years, but the business is moving towards two year replacement, partly due to the number of hours some tractors do (mainline tractors tend to do 1500 to 2000 hours a year). In addition, he has been advised that it is better to replace the CVX tractor after two years to retain its value.

Reasons for fleet choices

Four years ago a switch to Case tractors was made after continual reliability problems with the tractor make at the time. At that point, the business purchased a number of MX tractors which were well liked by the operators. When these came to be replaced, the MXM range was not well liked, especially the gearbox (6 powershift gears in 3 electronically selected ranges). The business had been considering CVX tractors anyway, as it was thought they would be useful for destoning work, so the decision was made to move to these instead. The first ones were well received by the operators which is important to the manager. The John Deere is a remnant from when one of the units was managed separately, but has been retained for cheap power during the cultivations period.

Next tractor

The next tractor will definitely by a Case CVX. It is likely all four MX135 tractors will be replaced with CVX machines. The MX270 would also be replaced with a CVX if a (Case) model existed in that power range. At present that will be replaced with a Case full powershift tractor.

Specific CVT tasks

The CVX tractors are used for nearly all operations on the busy farms, but a CVX is always used for destoning and harvesting vegetables and potatoes.

User benefits of a CVT transmission

The ability to perfectly match the tractors' forward speeds with the conditions is a major benefit of these tractors, particularly for the volume of vegetable and potato operations undertaken. Further benefits can be seen when harvesting with a 2nd CVX alongside (with a trailer) as a perfect matching of tractor speeds is possible. The operators like the driveability of the tractors, especially the ability to slow down with loaded trailers in a controlled manner rather than the forced speed reduction caused by downshift with a conventional gearbox. It has also been noticed that the tractors have improved traction abilities, this could be partly due to higher tractor weight, but also the absence of a lurching motion, common during gearshifts, helps to keep momentum and avoid getting stuck during late autumn harvesting.

Possible to replication these benefits with a conventional transmission

It was thought that a smoother gearshift to avoid getting stuck should be possible with a conventional tractor, and would be a big improvement, but ultimately the need for variable forward speed can not be easily replicated with a conventional transmission even with a high number of ratios and creeper gears. Volume of harvested material such as carrots can, in a good year, mean a very low forward speed is necessary to avoid blockages and waste.

Detrimental aspects of the CVT transmission

There have been no major detrimental aspects of the transmission so far. Even fuel consumption was thought to be better but no accurate records were kept. The tractor was thought to be a bit hesitant at starting from a standstill, this had been attributed to the parking brake but was not a major concern.

Acceleration of the tractor

The acceleration of the tractors was possibly slower than others, but this could possibly be attributed to a decision to de-rate them all to 40km/h, against the advice of Case. This was done as he felt the trailers were not capable of 50km/h operation and doubted the tractor brakes with laden trailers.

Ease of use of tractor transmissions (all types)

After initial problems identifying how to operate the CVX tractors properly, they have quickly become the easiest to use for the experienced full time employees. This can cause issues for casual users who tend to be restricted to the semi-powershift MX135 tractors where possible.

Views on the control systems for their tractors

Difficult to answer this question as the amount of direct contact with the tractors has been minimal, but perhaps there are too many different ways of achieving a forward speed, hence the issues for the casual user.

Potential improvements to any tractor control systems

Again, a lack of experience with the tractor meant this was not answered.

Additional Desirable Features on CVT tractor

None were identified

Any Unimportant or Unused Features

None were identified

Do they know who manufactures the transmission

Mr Saville identified Steyr as the transmission manufacturer, ZF (now the manufacturer based on Steyr original design) was identified as being a quality engineering company.

Price premium paid for the CVT tractor

The huge purchasing influence of a large business such as this meant the price premium paid for the CVX tractors was minimal.

Does the CVT tractor provide value for money

Yes, partly as little extra money was paid and for all the reasons identified earlier. In addition Mr Saville wanted to stick with a Case product rather than a common platform model.

Maximum price premium prepared to pay for a CVT

With a like for like tractor Mr Saville would be willing to pay a £3,000 price premium for the variable transmission. The difficulty comes as some of the other features of this tractor are not on comparable tractors.

Desirable future tractor powertrain developments

None stated.

Dealer Service

Very happy with the dealer service, but then would expect to be given the number of machines being operated. Dealers undertake all major servicing of the tractors. The service interval (250 hours) is too short and the associated cost and downtime is an issue with the CVX tractors.

Precision farming techniques used or considered

Not really, a ±10% button is used with the seed drill to reduce the seed rates in valleys, but done by the drill operator. Yield and input records are undertaken purely on a field by field basis. They would consider using more techniques if there was a sound business case.

## A6.3 Interviewee C Transcript

Name

Phillip Benson

Address

Barley Cottage, Thrupp, Faringdon, Oxford. SN7 8JX

Interview date

Tuesday 23rd November 2004

Business details

A large contracting business centred around, but not limited to, grassland contracting. Undertake work on around 120 farms, of which a quarter are small equestrian units which require hay or haylage making together with pasture maintenance. The larger units mainly require baling and solid manure spreading. An agreement with another contractor also creates around 1000ha of mowing and raking for silage each year. The business runs five large square and three round balers, three bale wrappers together with three mowers, two tedders and a rake. Four rear discharge manure spreaders are used or hired and the business also undertakes hedge trimming for customers.

Non farming activity

In addition to manure spreader hire, landscaping work is undertaken during quiet periods. A tracked excavator is hired in as required for this work.

Tractor fleet profile[1]

- Case 5150 (2nd hand 2000 – 4 years old). 93kW with 16/12 semi-powershift transmission (4 powershift, 4/3 mechanical ranges). 500 hours p.a.
- Case Magnum 7210 (new 1996). 134kW with 18/4 full powershift transmission. 500 hours p.a.
- Case Magnum 7220 (2nd 1999 – 3 years old). 149kW with 18/4 full powershift transmission. 500 hours p.a.
- Case MX135 (new 2000). 104kW with 16/12 semi-powershift transmission (4 powershift, 4/3 mechanical ranges). 1000 hours p.a.

[1] Engine power from OECD tests, or brochures if no OECD data exists, brochures may quote a bare engine rather than P.T.O power.

- Case MX135 (new 2001). 104kW with 16/12 semi-powershift transmission (4 powershift, 4/3 mechanical ranges). 1000 hours p.a.
- Case MX170 (new 2001). 130kW with 16/12 semi-powershift transmission (4 powershift, 4/3 mechanical ranges). 700 hours p.a.
- Case MX200 (new 2000). 144kW with 18/4 full powershift transmission. 1000 hours p.a.
- Case CVX170 (new 2002). 125kW with CVT. 1300 hours p.a.
- Case MX135 (new 2003). 104kW with 16/12 semi-powershift transmission (4 powershift, 4/3 mechanical ranges). 1000 hours p.a.

Driver policy

Although the business employs seasonal workers, the policy is to try and keep one driver on each tractor throughout the season. More emphasis is placed on the CVX tractor, which is used by one of the full time employees, not a casual member of staff.

Tractor replacement policy

Although once the business was established all equipment was purchased new and replaced frequently, the current policy is to make tractors last longer, provided they are reliable, and add 2nd hand tractors as required for some of the less used vehicles in the fleet. There are no strict hours or age of replacement, this is partly due to the fact that some of the tractors (e.g. the 7210 and 7220) are only used with a large baler in the summer months, therefore typically only working 500 hours per year. Whilst they have depreciated to a level which costs the business little each year, replacing them with new high horsepower tractors to drive the balers would require a large capital outlay.

Reasons for fleet choices

Tractors are purchased based on the size required for the likely tasks to be undertaken with that machine. At the time of purchasing the CVX tractor, the business was looking to buy a 125kW tractor and the CVX was demonstrated. The tractor was very appealing, especially for baling and for operating a sophisticated baler & wrapper combination. Other reasons for the purchase of a CVX included the 50km/h transmission (most tractors were only 40km/h at the time), low engine speeds for road work and the comfort of the tractor for the operator (importance of looking after the full time employees was mentioned). As discussed, other tractors are only used for the summer months with balers, here the emphasis is on cheap power (particularly for operating the square balers).

Next tractor

At present, until the full effects of the mid term review is known on the business, there is a hold on non-essential machinery purchases. There is every intention to replace the CVX tractor with another, probably the 1190 model as an increase in engine power would be useful. Ideally, the desire is to run two CVX tractors to allow two main drivers to have one each. At present there would not be any business justification for more than two variable transmission tractors.

Specific CVT tasks

The CVX tractor has been used to great effect with both round and square balers. The tractor has done some ploughing work together with a substantial amount of power harrowing and drilling. Muck spreading and general haulage tasks have also been undertaken.

User benefits of a CVT transmission

Both Mr Benson and the operator stated that the main benefit of the CVX tractor was the ability to match forward speeds with conditions, independently of engine speed. The tractor is ideal for baling where faster progress can be made. Similarly, with spreading manure, faster cycle times were possible. Interestingly, this is one of the few situations where it is possible to directly compare two tractors when working in a team. For road work the lower engine speed together with 50km/h capabilities meant they felt fuel was saved and faster progress made (no accurate fuel records have been kept). The lack of potentially harsh gear changes was seen as having a positive impact on implement life. Smoother changes in direction had been noticed by the operator, and the lack of gear changing was appreciated. The operator felt much less tired at the end of a day than he used to when operating conventional tractors.

Possible to replication these benefits with a conventional transmission

Both felt that the big advantage of the CVX transmission was the possibility of infinitely variable speed, which is not possible with a standard transmission. The smoothness of gear changes and the ease of these could be improved to help make the tractor easier to drive. They both felt there was no reason why the smoother change in direction of the CVX cannot be replicated with a conventional transmission.

Detrimental aspects of the CVT transmission

When the tractor first arrived, fuel consumption was considerably higher than other tractors. This was resolved with an updated fuel pump and software updates, the tractor now probably uses slightly lower fuel, but no accurate records are kept.

Acceleration of the tractor

The tractor is noticeably quicker at accelerating than other tractors.

Ease of use of tractor transmissions (all types)

Both stated that, for an experienced operator, the CVX tractor was the easiest to operate with the full powershift being easier than the semi-powershift transmissions. Mr Benson was keen to point out that they would never consider allowing casual staff to operate the CVX tractor.

Views on the control systems for their tractors

As their CVX is one of the older models, it features two preset speeds (0-14km/h and 0-50km/h) rather than the three on the newer models. Although these preset speeds can be selected by a push button, there is still the ability to manually adjust the preset speed to the desired level. The operator tended to only use the preset road speed (i.e. 0-50km/h) when undertaking roadwork. The rest of the time he set his own speed via the controller. The control system for this tractor was much preferred to the other tractors in the fleet.

Potential improvements to any tractor control systems

Their only problem with the control system was with reference to the autopark feature. The tractor has a power neutral hold (to prevent it from rolling when stationary), but after a period of time the mechanical park lock is engaged, this can be an issue when waiting at a road junction, as it takes too long to disengage again and set off once the travel pedal is pressed.

Additional Desirable Features on CVT tractor

None were identified with respect to the powertrain, their only real issue was the small size of the CVX cab.

Any Unimportant or Unused Features

The 0-14km/h speed setting was not used, the engine speed reduction when lifting an implement was also not used.

Do they know who manufactures the transmission

Both identified the transmission as being designed and built by Steyr (is actually ZF, formally Steyr). Mr Benson stated he would not have bought a Steyr tractor as he prefers the support Case give him, but identified them as a good technologically advanced company.

Price premium paid for the CVT tractor

Between £5,000 and £6,000 more was paid for the CVX tractor.

Does the CVT tractor provide value for money

A lot of advantages are assumed, but not really known as it is difficult to compare like for like (spreading muck being the exception). With the increasing cost of fuel Mr Benson would have liked to have seen some practical information to justify his purchasing decision. He was hoping for a higher resale value when he replaces the tractor in the next year, which would help to justify the higher outlay.

Maximum price premium prepared to pay for a CVT

He felt he paid about the maximum he would be prepared to pay extra for the variable transmission. But it is a difficult point, as at the time of purchase 50km/h standard tractors were rare, as was front suspension, it is therefore difficult to attribute a known amount to the transmission.

Desirable future tractor powertrain developments

Nothing was raised.

Dealer Service

Very happy with their dealer and Case themselves, although he thought this was probably due to the number of machines operated.

Precision farming techniques used or considered

Nothing used or likely to be considered.

## A6.4 Interviewee D Transcript

Name

Jim & James Fairley

Address

Wolves Hall Farm, Wolves Hall Lane, Tendring, Essex. CO16 0DG.

Interview date

Friday 10th December 2004

Business details

365ha (900 acre) all arable farm. The farm is 2/3 owned, the remainder is rented, and is farmed as a father and son partnership. Soil is variable, including grade 1 sandy loam and heavy Essex clay. Principle cropping is wheat, barley, peas & oil seed rape, together with sugar beet (34ha) and potatoes (16ha). The farm provides work for Jim and James Fairley together with a full time employee, plus seasonal help as required.

Non farming activity

None

Tractor fleet profile[1]

- Case CVX1190 (new 2003). 140kW with CVT. 600 hours p.a.
- Case MXM155 (new February 2004). 114kW with 18/6 semi-powershift transmission (6 powershift, 3 electronic ranges). 500 hours p.a. (est).
- Case MX135 (new 2002). 96KW with 16/12 semi-powershift transmission (4 powershift, 4/3 mechanical ranges). 500 hours p.a.

Driver policy

There is no strict policy for driving the tractors, but generally either of the business owners tends to drive the CVX tractor, this is partly due to the need to drive the tractor properly in order to maximise its effectiveness.

Tractor replacement policy

The business is committed to running new tractors, and as such replaces the tractors on a three to four year basis, unless business requirements dictate, e.g.

[1] Engine power from OECD tests, or brochures if no OECD data exists, brochures may quote a bare engine rather than P.T.O power.

an increase in farmed area or larger equipment dictates an increase in tractor size. This is how the CVX1190 replaced an earlier CVX170 after 2½ years.

Reasons for fleet choices

The CVX170 arrived on the farm as a demonstration tractor to be compared against a Case MX170 and a New Holland TM170. The three tractors were all used for ploughing where the CVX outperformed the other two tractors, hence the reason for the purchase. This tractor was then replaced with a larger CVX as the business needs dictated. The MXM155 was purchased rather than another CVX due to the additional CVX cost and its weight.

Next tractor

The next tractor will probably be a Case MXU135, as the MX135 is due to be replaced. It was stated that the farm will probably only run one CVX tractor due to the cost and weight issues. In addition, there is no smaller range of CVX tractors.

Specific CVT tasks

The main role for the CVX tractor is ploughing and drilling cereals. It is also used for carting crops from the combine to the store.

User benefits of a CVT transmission

If operated correctly (i.e. engine throttled back, transmission ratio higher) it was felt the CVX would use approximately 50% less diesel than the MXM whilst ploughing. This was working at 8-10 km/h with a 6 furrow plough. It should be noted however, that accurate records of fuel consumption were not kept. In addition to the fuel savings, the ability to maintain a high forward speed was an advantage for maximising the area covered. During carting the advantage of being able to easily match the combine forward speed was appreciated, especially as on at least one occasion grain had been spilt when the MXM stalled when trying to reduce forward speed whilst in too higher gear. The 50km/h road speed and quicker road acceleration were also positive features.

Possible to replication these benefits with a conventional transmission

Both felt that the major advantage of the CVX was the ability to choose any forward speed, there is no conceivable way this can be replicated.

Detrimental aspects of the CVT transmission

They have experienced a situation where the tractor was loaded to an extent where it would not drive at all. This was with a three-legged subsoiler, when the tractor was put into drive with the implement in the ground. The only solution was to remove a leg from the subsoiler. No operational aspects of the CVX transmission were seen as detrimental.

Acceleration of the tractor

The tractor was noticeably quicker to accelerate when used with a trailer (typically 14 or 15 tonnes) compared with other tractors. This was even taking into account the higher engine power.

Ease of use of tractor transmissions (all types)

The easiest tractor for a novice to drive was their MXM155 as all gear changes are on a button and are relatively easy to use. For someone who uses the tractors everyday the CVX is very straightforward to use, but does require some initial learning to get the most out of it. The MX135 is relatively straightforward to use, but not as easy as the MXM155 due to the need for mechanical gear changes.

Views on the control systems for their tractors

They both felt some of the CVX tractor controls were poor, especially the hydraulics. The ergonomics of the tractor were not well thought out. Gear selections, particularly direction changes, were easy to do on the MX135. They were both fairly neutral about the controls of the MXM155.

Potential improvements to any tractor control systems

Their views were different on how the transmission controls could be improved; both agreed that they would like a method of speed selection on a lever close to hand, in addition to the somewhat isolated cab pillar unit.

Additional Desirable Features on CVT tractor

None were identified

Any Unimportant or Unused Features

None were identified

Do they know who manufactures the transmission

Both knew that the transmission was a ZF unit, and both identified ZF as a company who specialises in gearbox manufacturing and as a good company. It was stated at this point that a Fendt tractor had been tried in the past and the transmission control was found to be a lot more complicated to use.

Price premium paid for the CVT tractor

They paid approximately £4,000 more for the CVX170 compared to the best price for the MX170 they also demonstrated. At the time of replacement with the CVX1190, comparisons with standard transmission tractors were not made as they wanted another variable transmission tractor.

Does the CVT tractor provide value for money

They both felt that improved fuel consumption alone should go a long way to making up the price premium. Although it is difficult to tell if the tractor covers much more ground during a working day, the ease of operation and the improved functionality were benefits they were willing to pay for. They also felt that their policy of replacing tractors after 3 to 4 years would mean they would gain from the expected higher trade-in value of the CVX tractor.

Maximum price premium prepared to pay for a CVT

Unable to say as it would depend on the deal available at the time, but they suggested that, based on the replacement of the CVX170, provided a satisfactory deal could be made they would not be concerned with the price of a tractor with a fixed ratio transmission.

Desirable future tractor powertrain developments

They liked the intuitive nature of the CVX tractor on the road and felt all modern powershift tractors could be made to drive like this, i.e. the foot throttle to be used to control forward speed rather than engine speed.

Dealer Service

They felt the service engineers in general, and particularly with the CVX, are very knowledgeable. The issue is their local Case dealer has closed, meaning their dealer is now 45 minutes away. New Holland and John Deere dealers are much closer so this may mean they change in the future.

Precision farming techniques used or considered

None are presently used as they are not deemed to be cost effective, but they would not rule out some use in the future.

## A6.5 Customer Survey Questions

The 30 questions which survey participants were asked to score and rank are detailed below.

1. How important is it if you have a cab interior with maximum durability?
2. How important is it if you have a cab with maximum quality of finish (e.g. parts fitting well together, etc.)?
3. How important is it if you have maximum # of specialised cab storage options (e.g. cold, hot, tools, etc.)?
4. How important is it if you have maximum smoothness when changing transmission ratios?
5. How important is it if you have maximum smoothness of ride?
6. How important is it if you can enter and exit your cab with minimum effort (e.g. steps, steering wheel, door)?
7. How important is it if you have all round visibility with minimum discomfort?
8. How important is it if your tractor is highly visible to others?
9. How important is it if you have maximum visibility 24 hours a day in all operating conditions (e.g. fog, dust, snow, mist, etc.)?
10. How important is it if you have maximum range of your vehicle speeds (forward or reverse)?
11. How important is it if you have maximum control over your implement/attachment/trailer functions in all your applications?
12. How important is it if you can use your tractor functions with minimum training or experience?
13. How important is it if you can manage the control of your cab environment with maximum flexibility?
14. How important is it if you can control functions with minimal effort?
15. How important is it if you have maximum flexibility to capture data related to your work?

16. How important is it if you can install any information or control components (such as GPS/Auto-guidance or monitors) in a maximum number of your vehicles, including different brands?
17. How important is it if you can use a maximum number of implements/attachments/trailer with your tractor (front/rear)?
18. How important is it if you can reconfigure tractor weight distribution with minimum effort?
19. How important is it if you have a maximum choice of expanded tread setting on single wheel/tracks?
20. How important is it if you can attach an implement with minimum effort?
21. How important is it if you can maximize manoeuvrability?
22. How important is it if you clean your tractor with minimum effort?
23. How important is it if you have to do minimum routine maintenance?
24. How important is it if you are alerted to failures about to happen or vehicle service needs, in a maximum number of conditions?
25. How important is it if you can quickly troubleshoot tractor problems?
26. How important is it if you have minimal tractor downtime?
27. How important is it if you have maximum fuel efficiency with your tractor?
28. How important is it if you can use renewable fuel with minimum effort?
29. How important is it if you can obtain traction with minimum soil or crop disturbance?
30. How important is it if you can obtain maximum traction in all applications?

# A7 Additional Control Strategies Data

*Table A7.1 – Additional allowable torque in boosted mode due to transmission power*

| Transmission Power (hp) | Boost ($B_{\%}$) |
|---|---|
| 0 | 0 |
| 1 | 1 |
| 6 | 3 |
| 8 | 4 |
| 10 | 5 |
| 12 | 6 |
| 14 | 8 |
| 16 | 9 |
| 18 | 12 |
| 20 | 14 |
| 22 | 16 |
| 24 | 17 |
| 26 | 18 |
| 28 | 19 |