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DESIGN AND DEVELOPMENT OF A DATE HARVESTING MACHINE

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ABSTRACT

Design and Development of a Date Harvesting Machine

Existing date harvesting machines are vehicles equipped with a long arm to lift a man on a platform to harvest the fruits. The arm and the vehicle are heavy (4 to 8 tonnes), expensive (from £16 000) and are not sufficiently manoeuvrable in constricted date groves. Most dates in the main producing countries, including Iran, are therefore harvested manually. The manual method is unsafe, slow, expensive (£0.63 per tree) and the fruit quality is often damaged.

A light, weight 4 wheel drive, remotely controlled tree climbing machine is, therefore, a potential solution to the problems of harvesting and servicing (such as pollinating and pruning). A prototype of such a device was designed, developed and evaluated under laboratory conditions. To determine the operating characteristics and feasibility the machine was designed to climb the tree using pneumatic tyres as traction wheels. The machine can be transformed to ground drive and move between trees under its own power. This approach reduces the machine weight, cost and size because the tree trunk is used as a support for the machine to climb to the fruits. It is operated and controlled from the ground which improves the operator safety.

A vertical traction theory for this type of machine has been developed based on the tree size and surface characteristics and machine size and weight which can be used to design date harvesting and climbing machines with different capacities. The test results showed that the experimental machine could achieve a tractive efficiency of 90% and that the optimum wheel slippage was between 10 - 15%. The machine consumes a maximum of 1.4 kW power which is only 3% of the power requirement of existing systems. The machine weight is 150 kg which is 2 - 4% of the existing systems' weight. It is capable of climbing the tree at a maximum speed of 0.27 m/s although the optimum speed is 0.17 m/s for best control. The prototype can carry a payload of 100 kg of dates and, considering a field efficiency of 75%, it can potentially harvest a tree in 22 minutes which is 18 % faster than the manual system in

Iran and 6 % faster than one of the mechanised systems used in Saudi Arabia. The harvester can work on tree diameter ranges from 300 to 850 mm and can pass over the tree leaf bases of 41 mm high. The machine should not damage the tree because the tree resists the machine stresses with a minimum safety factor of 7.

An economic analysis showed that it can be manufactured in Iran at 20 % of the cost of existing systems. The machine cost per tree is equal to the hand harvesting method (£0.63 per tree) for Iranian farmers if it harvests 978 trees per year.

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CHAPTER 1

INTRODUCTION AND LITERATURE REVIEW

1. Introduction and Literature Review

1.1. Introduction

Date is an important fruit and food in more than 30 countries. It is a concentrated energy food which can be easily stored. Many products are obtained from date including sugar, honey, vinegar, juice and flour together with handicrafts such as baskets, ropes, hats and carpets derived from the leaves (Rohani 1988).

The most important date producing countries in order of crop size (Barreveld 1993) are shown in Table 1.1. The figures show that date production world wide has been steadily increasing over the last 30 years and in 1990 reached 3400 kilotonnes, 85% more than 1960. Six major producers produce 2637 kilotonnes which is 75% of the total world production; these countries plus the four medium producers supply 90% of the total world production. The remaining 10%, a total of 372 kilotonnes, comes from 22 other countries, of which the most important are Morocco, Chad, USA, Israel, United Arab Emirates and the Yemen.

Iran produces 540 kilotonnes of dates which represents 16% of the total world production, and places it in second position among date producing countries. For more than 4000 years people have been cultivating dates along the banks of the Karoon and Karkheh rivers in Iran. Date statistics (1992) showed that there are more than 17 million yielding trees growing in the country.

A recent study (Elhampour 1993) showed that Iran has the potential for further increases in the yield and the quality of dates by using mechanised cultivation methods, so there is a major interest in the mechanisation of harvesting operations, because most of the harvesting is done manually. A skilled worker climbs the tree and cuts the fruit bunches. The worker puts his life at great risk, because some trees grow to a height of 24 meters. It is also an expensive and slow operation and the quality of the fruit can be damaged. There is also a shortage of skilled workers to climb the trees during the harvesting season (Bakhshoodeh & Akbari 1993).

Mechanised harvesting has been successful in the USA. Brown (1983) showed that of the total cultural operations, the harvesting, pollination and pruning are the most labour intensive accounting for more than 80% of the total production costs. Attempts have therefore focused on trying to mechanise these operations first. Mechanisation in the USA date industry has decreased the harvesting cost by 50 % and the number of workers by 75 %.

Table 1.1 Date production, kilotonnes; after Barreveled (1993)

Scale	Country	Average 1961 - 65	Average 1971 - 1975	Average 1981 - 85	1990
Main Producers	Egypt	407	386	457	580
	Iran	305	295	385	540
	Saudi Arabia	170	280	417	525
	Iraq	336	400	346	490
	Pakistan	96	177	234	290
	Algeria	122	159	193	212
	Sub - total				2637
Medium produces	Sudan	58	109	120	130
	Oman	40	49	74	125
	Libya	36	68	84	74
	Tunisia	35	41	61	73
	Sub - total				402
Other 22 producers					372
Total		1 839	2208	2645	3411

1.1.1. Aim

The aim of the project is to investigate the technical and economic feasibility of developing and using a remotely controlled climbing machine to harvest the date palm trees specially to suit the Iranian palms.

1.1.2. Objectives

A remotely controlled tree climbing machine is a potential solution to the problem so it is necessary to develop a vertical traction model which will not damage the tree and has suitable performance both mechanically and economically. The detailed objectives are:

1. To identify and compare the different harvesting and climbing methods and select the most appropriate system.
2. To develop the concept of a tree climbing machine and vertical traction theory.
3. To determine the feasibility of the tree climbing system considering the interaction between the machine and the tree.
4. To evaluate the technical performance of the climbing and transformation to ground travel systems.
5. To assess the economic feasibility of the design under Iranian condition.

1.1.3. Outline methodology

The following steps were followed to achieve the above aim and objectives:

1. To define the problems of date harvesting.
2. To select and use a suitable systematic design method.
3. To use the method to select the machine specifications.
4. To use the method to generate and compare the alternative solutions and to select the suitable concept for a climbing machine.
5. To develop the mathematical equations and principles of vertical traction suitable for the selected climbing machine.
6. To conduct a field study of hand harvesting systems in Iran and measure the physical properties of the date palm tree and fruit and tree sizes concerning the interaction between the tree climbing machine and the date palm.
7. To design and manufacture an experimental machine using the mathematical model and field study data with particular reference to date tree damage.
8. To design a measurement system for the experimental machine to evaluate the mechanical performance of the chosen design in terms of weight, harvesting speed and payload (date weight) capacity.

9. To study the effect of wheel spacing, payload and wheel forces against the tree on the machine performance including the power consumption, wheel slippage and power efficiency using the test rig and measurement system.
10. To conduct a cost/benefit analysis of the chosen design and determine the initial feasibility of the machine for Iranian farmers.

1.2. Literature review

1.2.1. Morphology of date trees

The date palm (Phoenix Dactylifera) is a monocotyledon of the family of the Palmae. It is a feather palm, characterised by compound leaves with a series of leaflets on each side of a common petiole, originating from one growing point at the top of the trunk. The date palm may reach an age of over 100 years and reach up to 24 m in height to the growing point. Normally the age limit is less than this and consequently the height will not be more than 15-20 m maximum before it will be cut down due to declining yield and the increasing difficulty (and danger) of reaching the crown during pollination, bunch management and harvesting (Seelig 1974).

Leaves are formed in buds in a slightly ascending spiral around the growing point, at the rate of 10-30 per year. With an average leaf life span of 3-7 years the number of leaves per palm varies from 30-140. Initially, the young leaf is enclosed in a leaf sheath of tender tissue which will open to give passage to the extruding leaf. The sheath tissue will dry out and eventually only the fibre will remain at the base of the leaf at a length of about 0.2 m. Leaves may reach a length of 6 m, with an average of 4 m (Dowson 1982). The leaves usually grow spines that are very sharp and hard in their lower region near the base and the date grower will quite often remove the spines to prevent injury during cultural operations. Under natural conditions, the leaves, after their useful life is over, will dry and bend down alongside the trunk where they would stay for a significant time before dropping to the ground. The date grower, however, will each year remove the old leaves to give better access to the crown. The leaf bases will remain attached to the trunk and where palms are still climbed, are used as steps for the climber's feet.

As the date palm is dioecious fertilisation of the female flowers is required for fruit setting, which in fruit palm cultivation is not left to the wind or insects but is traditionally done manually by inserting a piece of the spikelet of a male flower when the female flower opens. Modern methods collect the pollen from the male flower which is diluted with a carrier powder (such as flour) and is pneumatically dusted onto the female flowers by a controlled flow of pressurised air accumulated in a knapsack sprayer tank (Loghavi 1993).

Some 12 (can range from 0 - 25) flower buds develop during the winter in the axils of the leaves just below the growing point. The inflorescence enveloped in a spathe, pushes through the fibre on the leaf base. It will split open upon maturity of the internal flower cluster which consists of a main stem (fruit stalk) which rapidly lengthens outside the spathe. Annual crop volume and individual fruit size can be influenced by the grower by bunch removal and fruit thinning. An average of 8 bunches are left on the tree with an overall leaf to bunch ratio of 8 or 9. Figure 1-1 shows a young yielding tree with a bunch of fruits. The bunch stalk usually bends down and the fruit bunches will stay lower than the tree crown around the tree.

It takes around 200 days from pollination to reach full maturation (tamr stage). The mature fruits are typically harvested by either hand or mechanised harvesting.

Figure 1-1 A young yielding date tree with a bunch of fruits

1.2.2. Hand harvesting methods

In the main and medium producer countries in the world, which have been planting date for centuries, date is mainly harvested by hand. Skilled and trained labourers climb the tree either "free hand" or using a belt around themselves and the tree. They carry a knife and possibly other tools such as a rope and a pulley equipped with a hook. At the tree crown the pulley is suspended and the rope used to send the fruit down. If they are not equipped with these tools they drop the fruit onto the ground.

To help climb the tree, some growers dig holes or hammer nails into the tree trunk. When the trees are close to each other, some workers walk on the leaves and move from one tree to another.

Depending on the variety and climate, dates are hand harvested by one of three methods:

a) Bunch cutting is used where all fruits on a bunch ripen at the same time. The worker cuts the bunch, ties the rope to it and lowers it to the ground. If the fruits are dry and they do not have the rope they drop the bunch from the tree.

b) Bunch shaking is used when all fruits do not ripen at the same time and where shaking the bunch does not seriously affect the quality of the remaining fruit. In this case the tree is harvested several times during the season. The worker shakes the bunch with his hand or a stick, with other labourers holding a circular cloth around the tree on the ground to collect the falling fruits. This damages many of the fruits because of the impact shock.

c) Hand picking is used for those varieties which have fruits that are sensitive to vibration. Workers pick mature fruits one by one and collect them in a small basket. When the basket is full it is lowered to the ground (Kashani 1992).

1.2.3. Mechanised methods

Several types of harvesting machine are used in different countries, most of which are similar in principle. They are typically self propelled or trailed machines equipped with a boom and platform to lift a worker up to the tree crown. Examples of these systems are as follow:

1. Tractor lifting power platforms which are modified in Iran to harvest dates (Lagvar 1994; Shiraz Nardebam 1994). These are in use in a few places around the country.

2. Several systems for elevating workers into the trees, removing the fruit from the bunch and handling the harvested fruit have been developed in the USA. The harvesting system most generally adopted uses a truck-mounted telescopic boom to lift the worker on a platform up to the tree near the bunches. The mature bunches are cut and placed in a basket. The platform is then lowered to a shaker trailer where the fruit is mechanically shaken from the bunches into a bulk bin. Three men operate the system, a truck driver-boom operator, a bunch cutter, and a shaker operator (Brown and Perkins 1964, 1965, 1967; Perkins and Brown

1964, 1966). Brown (1983) explains that by 1966 in the main date growing area of the USA, 80 % of dates were being harvested mechanically.

3. Al - Suhaibani et al (1988) reported using a date service machine in Saudi Arabia which was designed at Silsoe College (KSU date harvesting machine). A "U" shape platform allows the worker to reach all of the bunches of dates without any additional movement of the platform. The machine can be fully controlled from the platform. A 50 kW diesel engine provides the power for the 4 wheel drive truck and hydraulic system which lifts the platform (Figure 1-2). The worker cuts the bunches and puts them in a basket. The basket can carry 200 kg of dates. Fruit Bunches can be lowered to the ground by a hydraulic powered winch.

4.

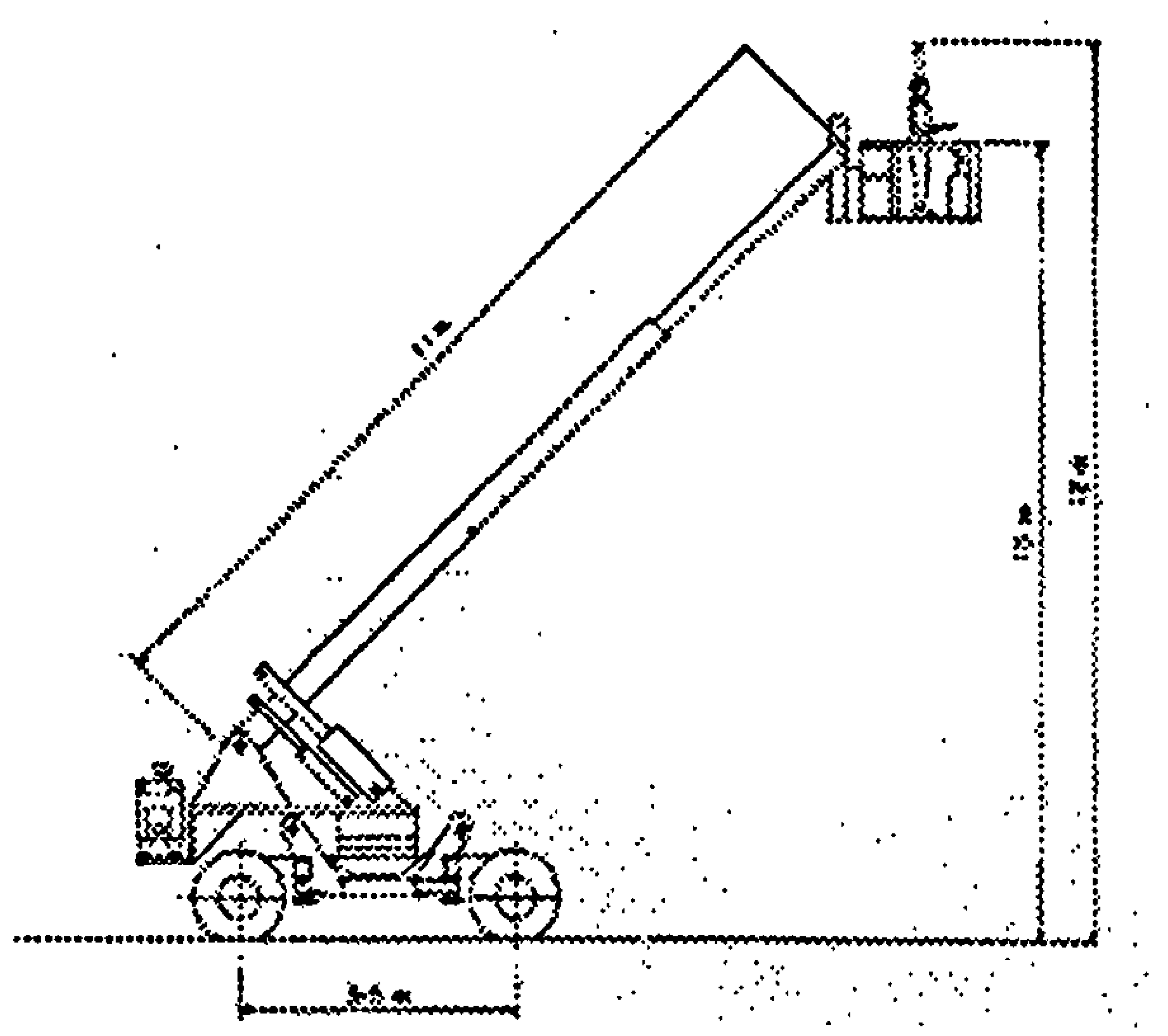


Figure 1-2 Field position of KSU date harvesting machine

The field test of the machine in 1990 by Al-Suhaibani et al (1992) showed that it can harvest a tree in 21 minutes which is faster, safer and easier than hand harvesting. Changing the planting techniques to have trees in rows was suggested to decrease the machine complexity and preparation time. Reducing the number of irrigation channels was also suggested to improve the machine harvesting time.

In Israel, Yoav Saring et al (1989) developed an integrated mechanical system that can harvest the fruits by shaking the tree trunk. A roll-out canvas catching frame is commonly used for collecting the fruit. This method is only suitable for those varieties of dates whose fruits on a bunch do not ripen at the same time.

1.2.4. Tree climbing machines

Climbing machines have not been used commercially to harvest dates, but investigators have designed and made prototypes to climb date palm and coconut trees.

Shamsi (1990) designed a climbing machine to harvest dates. The operator sits and holds on to the handle then he drives the climber by pushing the pedals with his feet (Figure 1-3). The traction element is a track made from a flat belt with aluminium angles bolted on it to grip on the leaf bases which produces a suitable traction force.

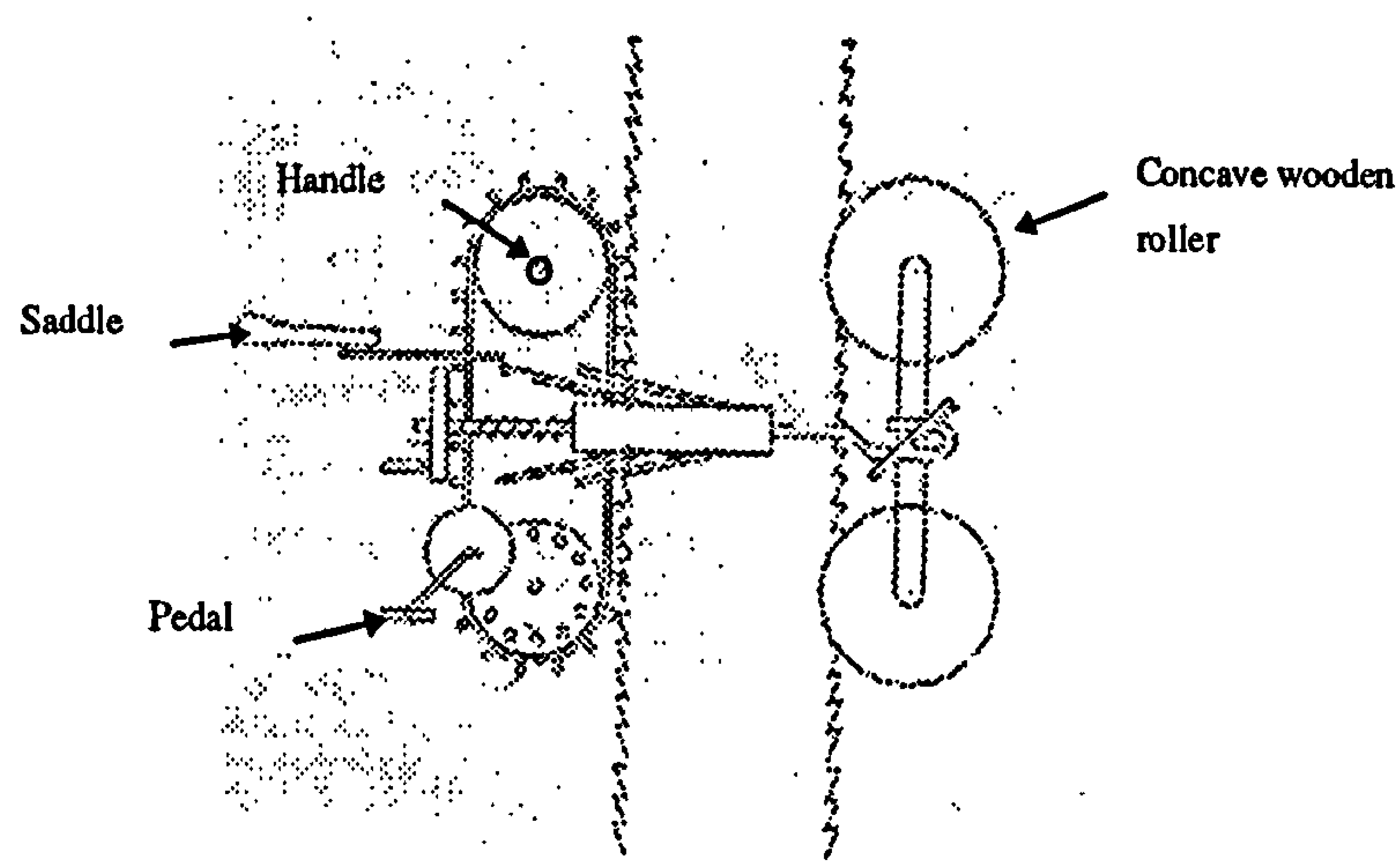


Figure 1-3 Man powered date harvester

A slider mechanism and a pair of springs provide the griping force on the tree. Two wooden concave rollers provide the lateral stability and prevent the machine from sliding from side to side on the tree. The study showed that the operator can climb with the machine at 0.2 m/sec.

Shamsi (1985) also proposed a design for a walking machine to carry one person to the crown of the palm tree, this was powered by a petrol engine and used hydraulic rams to provide the articulation.

Nicklin (1993) designed a tree climbing test rig to lift a man up to the date palm trees. The test rig (as manufactured) utilised a pair of concave steel rollers (coated with rubber) mounted in a fulcrum frame which transferred the operator weight to the rollers in order to grip the tree (Figure 1-4). The test rig was powered by electric motors and both of the rollers are driven by a sprocket and chain transmission system. The test rig could climb the tree at speeds up to 0.21 m/sec.

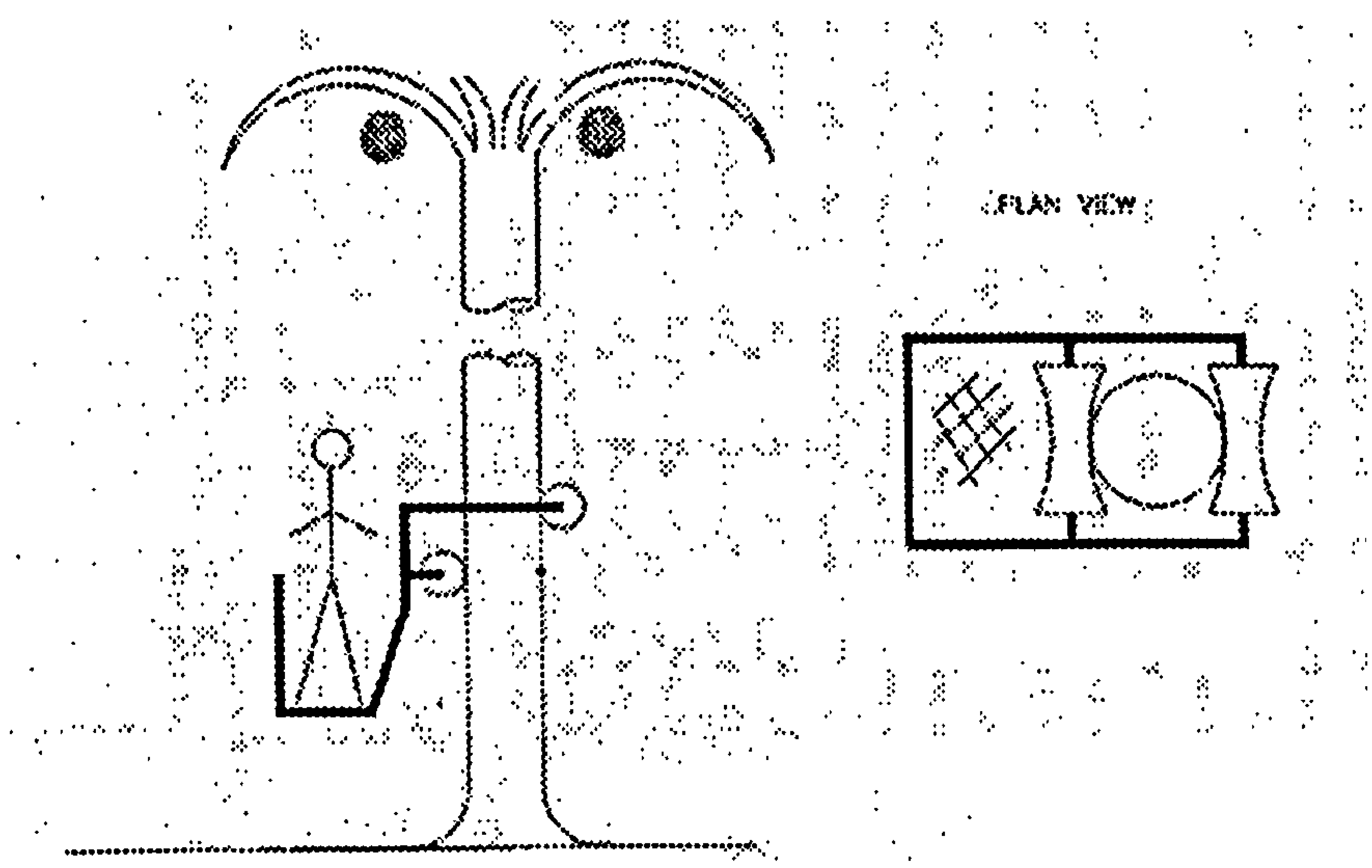


Figure 1-4 Concave steel roller tree climbing machine

It was concluded that for a real date tree the rollers design should be improved to cope with the surface characteristics of date palm trees. Recommendations for future work included finding methods of reaching the fruits on the far side of the tree because the man on the platform cannot reach to the far side. Moving the operator to the far side makes the machine unstable (because the operator weight provides the gripping force to the tree by producing an anticlockwise moment, as shown in Figure 1-4).

Davis (1977) developed a coconut tree climbing bicycle, it consisted of an angle-iron framework with wooden platform on which the operator rides. Friction rollers are pressed against the trunk of the palm by the operator's weight and he is carried to the top as he turns the handle by hand.

Climbing machines are being used in many other fields. Todd (1986) introduced machines that climb smooth vertical surfaces, using sucker feet or permanent magnet wheels that stick to the surface. A pipe climbing machine was made in Germany (Lawitzky et al 1997).

Hughes (1997) developed a portable security surveillance system that is equipped with a CCTV (Closed Circuit Television) camera to provide aerial images of the surrounding locality. To obtain height advantage, it has been designed to climb vertical posts of varying diameters and, together with the telemetry , is totally remotely controlled. The operator can control the unit from a remote position of up to 200 metres while the video images can be received by a module 600 metres away.

1.2.5. Tree climbing pruning machines

These machines are used to prune the extra branches from timber and are not used in date industry but they were studied to use the analogies in design. Pruning is an important cultivation operation which consists of cutting all the branches up to a defined height on live standing timber. The aim is to obtain the largest amount of wood without knots after 40 to 60 years.

Seirei Kogyo (1996) is currently producing a pruning machine. The machine has eight rubber wheels and moves on a spiral path to cover all the brunches around the tree. A small petrol engine supplies the power for the wheels and the chain saw. The machine is remotely radio controlled. Meier (1996) reports producing the similar machine in Switzerland.

Roux et al (1994) developed a mechatronic tree pruning machine (SELA) with 5 wheels, 1 driver and 4 idlers. A petrol engine (1.9 HP) drives the circular saw and the electric generator to supply electric power for a battery. All the other components are electric powered such as: wheel drive motor and frame opening jack which grips the machine to the tree and adjusts the wheel spacing for different tree diameters. Electronic sensors control the saw speed, climbing speed drive wheel speed and tree diameter adjustment. The machine is equipped with a counter which records the number of pruned bunches. The data can be down loaded to a computer when the operations are finished.

1.2.6. Critical discussion of harvesting methods

There are difficulties with the hand and mechanised methods explained in section 1.2.2 and 1.2.3. The main problems with the hand harvesting method are:

1. It is a dangerous job. Climbing a height of 24 meters and working at this height is very difficult and while certain workers can do it, they would rather do safer and easier jobs (Loghavi 1993).
2. It is a slow operation. Machines can work and harvest faster than manual methods.
3. It is an expensive operation. Using machines can decrease harvesting costs.
4. There is the shortage of workers to do the cultural operations.
5. Most of the workers do not like climbing tall trees because it is dangerous and so dates are not harvested from these trees.
6. The tree trunk is rough and the leaves have very sharp spines which can injure the worker.
7. Snakes go into the tree crown to eat bird's eggs and chicks. They can bite the workers climbing the tree. Workers have fallen from trees as a result.
8. Bees also sting the workers climbing the tree. During the field study a young worker was seen who was stung by bees and had fallen from the tree when trying to escape from the attacking group of bees which had their nest up the trunk.

However machines which have been designed so far are not suitable for the date groves in Iran and many other countries for the following reasons:

1. Operator safety is not resolved because they must still work at a considerable height (Grosz et al 1985).

2. Tree spacing is not standardised, so the machine may not achieve the required turning circle.

3. In many places the spacing between trees is used to cultivate other crops, so the machine cannot move and work.

4. There are many irrigation channels and borders which machines can not pass over easily.

5. Most of the available harvesters are capable of lifting the operator to 12-15 metres which is not enough for the tallest trees (25 meters).

6. Most of the machines are heavy and use too much power to lift the heavy boom and platform to position the worker near the bunches. The ratio of payload to machine weight is very small for this type of machine (about one to forty).

7. Mechanised harvesters are expensive (Sial 1984).

The tree climbing harvesting machines explained in section 1.2.4 are not designed to be used in the date harvesting industry.

The tree climbing system of pruning machines explained in 1.2.5 are not suitable for a date harvesting machine because:

1. They are designed to move on a spiral path to reach all tree branches around the tree, but the required climbing machine must travel in a straight line. Straight line motion of the proposed machine requires mechanisms which are different than a spiral motion design.

2. They are equipped with petrol engines and hydraulic power and control systems which make the machine complex and expensive.

3. The climbing unit is not designed to carry a variable weight.

4. The wheel sizes and traction systems are designed to move on a smooth surface and cannot roll over the uneven surface of the date tree.

5. They cannot move on the ground which is one of the requirements of the proposed machine.

The problems with the existing systems presents the need to design a new alternative which can overcomes many of these deficiencies.

CHAPTER 2

SYSTEMATIC DESIGN AND CONCEPT THEORY

2. Systematic Design and Concept Theory

2.1. Design methods

To design the date harvesting machine a systematic design process was followed. To understand and select the method several design methods were studied. As the design process usually starts with a problem and ends with a solution, design methods offer a logical frame work and attempt to ensure that the problem is defined correctly and the selected solution is the best among available solutions for the defined problem.

Buhl (1960) defined the following procedure as a structure under which the problem area and past experiences may be combined effectively to produce a solution:

-establish a problem area (recognition)

-determine exactly the nature of the problem (definition)

-collect pertinent information (preparation)

-break down and study this information (analysis)

-assemble the analysed information into various configurations (synthesis)

-study the merits of each possible solution and select (evaluation)

-sell the chosen solution (presentation).

A more detailed design model with several steps for each phase was offered by Pahl and Beitz (1984) based on the following phases:

-clarification of the task

-conceptual design

-embodiment design

-detail design.

Clarification of the task: this phase involves the collection of information about the requirements and constraints to be embodied in the solution. It is followed by the drawing up and elaboration of the detailed specifications (requirement list) for the design problem.

Conceptual design: the conceptual design phase involves the establishment of function structures, the search for suitable solution principles and their combination into concept variants.

Embodiment design: during this phase, the designer, starting from the concept, determines the layout and forms, and develops a technical product or system in accordance with technical and economic considerations.

Detail design: this is the phase of design process in which the arrangement, form, dimensions and surface properties of the individual parts are finally laid down, the material specified, the technical and economic feasibility re-checked and all drawing and other production documents produced.

French (1985) offers a design model that starts with the statement of need and includes four main stages as Pahl and Beitz (1984) but concentrates on the conceptual design phase.

BS 7000 "Guide to managing product design" provides a breakdown of the design process as follow:

-design objective

-design planing

-conceptual design

-embodiment design

-detail design

-manufacture design

-design evaluation.

Design objective: ensure that all information required for the design are included. In this stage the product requirements should be stated.

Design planing: is to plan the design activity in terms of the design resource, it should set out the activities of the design team and the required completion dates of the various stages of the design.

Conceptual, embodiment and detail design are similar to Pahl and Beitz (1984) method.

Manufacturing design: represents the transfer of the idealised detail design into a practical product, capable of being manufactured.

Design evaluation: is the monitoring and assessment of design process, the product and the project management.

A more realistic design model with clear phases was suggested by Pugh (1991). The model includes following phases.

-needs

-specification

-conceptual design

-detail design

-manufacturing

-sell.

The first point with the model is the separate "specification" phase before conceptual design phase. The output of this phase is a product design specification (PDS) document which includes the demands for matters such as: performance, environment, maintenance, target cost, size, manufacturing facilities, safety and standards. In fact the output of this stage is clear objectives and system requirements.

The second point is that Pugh does not include the embodiment design as a stage in the model. This is realistic because there is always a feed back from the embodiment design to the conceptual phase and in the embodiment phase there is need for creation and selection among concepts, therefore it become part of the conceptual design phase.

Cross (1991) introduced a model including the following stages and suggested particular methods for each stage as follow:

-clarifying objectives (by objective tree method)

-establishing functions (by function analysis)

-setting requirements

-determining characteristics

-generating alternatives (by morphological chart)

-evaluating alternatives (by weighting/rating method)

-improving details.

Special methods which were suggested for use in this model (objective tree, function analysis, morphological chart) were applied to the date harvester. Reasons for these selections are explained in next paragraphs.

The "Need" for searching a new alternative method to harvest dates was discussed in Chapter 1. From the "need" analysis, the objectives for designing the new machine were set. To illustrate the objectives the "objective tree" was used which is a clear and useful format for such a statement of objectives. It shows the objectives and the general means for achieving them and clears important specifications of the machine. This was also the control criteria from which design decisions were made considering these objectives.

Conceptual design phase includes two important elements which are "concept generation" and "concept selection".

To generate concepts the machine overall function and sub-functions should be known. Then concepts can be generated to fulfil the sub-functions and overall function. For the date harvester the overall function is harvesting dates and sub-functions are, climbing, cutting, landing, etc. The functions and sub-function can be just listed, but to visualise it and to clear the relation between function and sub-functions, the usual method (Pahl & Beitz 1984; Ullman 1992) is to use block diagrams. This useful method which is "function analysis" was used for the date harvesting machine design. It helped to concentrate on what was to be achieved by the machine, not how.

To generate concepts for each function the following methods can be used:

-literature search (Journals, reference books, conference proceedings)

-analysis of existing technical systems

-analogies

-brain storming

-combination of solutions

-using design catalogues

-using patents as idea sources

-using experts to help generate concepts.

Only such solutions proposals should be pursued as:

- are compatible with the overall function and/or with each other

-fulfil the demand of objectives and specification

-are reliable in respect of performance , layout etc.

-are expected to be with in permissible costs.

When alternative concepts are not refined enough for direct comparison with the mentioned criteria, using the controlled convergence method introduced by Pugh (1991) which is fairly simple, has proven effective. The method of weighting/rating (Pahl & Beitz 1984) is also another aid for uncertain situations.

The clear statement and presentation of concept solutions is a helpful way of searching among concept solutions and looking for possible combined solutions. Function means tree(Tijalve 1979) and morphological chart (Pahl & Beitz 1984) are common methods which can be employed for this purpose. The advantage of the "morphological chart" is that it can show the complete set of combinations of solutions and gives a clear view of the possible solutions. It helps to create new solutions by combining existing solutions.

2.2. Date harvester design model

Investigating problems of date harvesting systems and the study of the design methods helped to set the steps shown in Table 2.1 which made a model to achieve the proper design solutions. Names of stages were taken from Pugh (1991) because of the comprehensive and clear definitions used and for the list of contents that should be mentioned at each design stage.

"Objective tree", "function analysis" and "morphological chart" methods were used at relevant design stages. Using these methods was suggested by Cross(1991) in his design model. It was a realistic approach, based on the advantages explained in the previous section. Most of the aids explained for generating and selecting concepts in the previous section were used in the design.

2.3. Need

The need to design a tree climbing date harvesting machine was discussed in Chapter 1. To abstract this need, potential harvesting methods were classified under hand and mechanised harvesting methods as shown in Figure 2-1.

Table 2.1 Date harvester design model

	Design stage	Method, data and tools
1	Need (Chapter 1 & 2)	literature search, writing to experts to obtain their view and publications, analysis of existing systems, previous works and experiences
2	Specification (Chapter 2)	objective tree and PDS
3	Establishing functions (Chapter 2)	function analysis
4	Concept generation (Chapter 2)	function analysis, morphological chart, analysis of existing systems, literature search, analogy, design catalogues, help from experts
5	Concept selection (Chapter 2)	morphological chart, PDS, objective tree, function analysis, mathematical and physical modelling
6	Interaction between machine and tree (Chapter 3)	mathematical model, physical properties and size measurement of the tree (done in Iran)
7	Detail design of experimental machine (Chapter 4)	reference books, design catalogues
8	Technical performance (Chapter 5 &6)	experimental machine, measurement system
9	Economical performance (Chapter 7)	field study data (done in Iran), cost/benefit analysis

Of the mechanised harvesting methods, the ground based harvesting method is an available technology but is not feasible for date grove conditions in Iran and many other countries. The summary of the problems are that:

The machine cannot enter to the grove and work in it, because of close tree spacing, irrigation channels and other cultivation under the shadow of the date tree. The machine can be used in some situations but still suffers from a high investment cost.

The air based methods are conceptual. They cover the idea of using flying machines to harvest dates. These methods will not be discussed because they are outside normal engineering methods and are likely to be expensive, complex and potentially unusable.

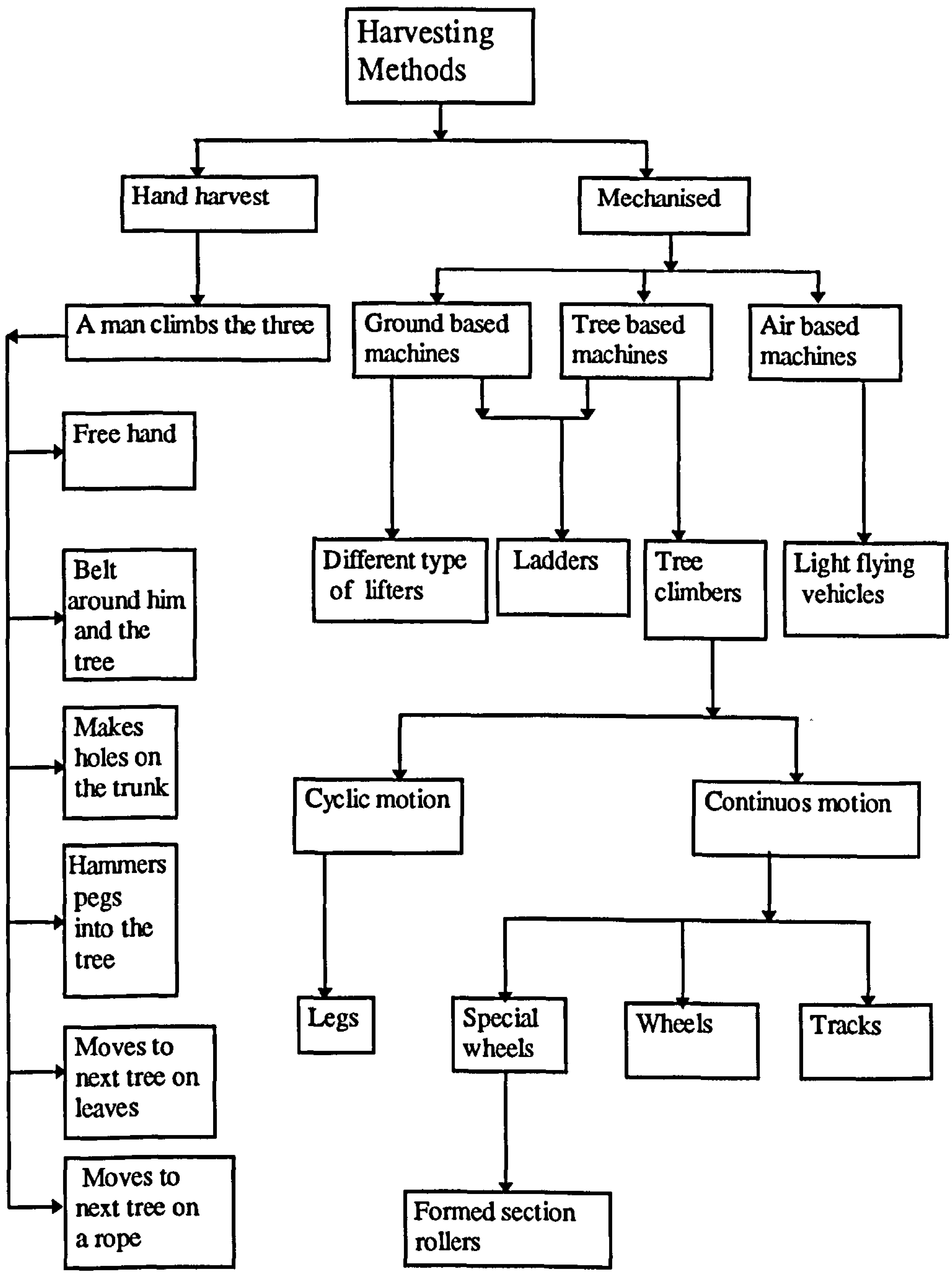


Figure 2-1 Alternative harvesting methods

In the presented mechanised methods, it is possible to have a combination of ground , tree and air based method. One example of the ground and tree base equipment is the ladder which uses the ground from the base and the tree trunk from the top.

The tree based or tree climber machines in the mechanised harvesting group are suitable alternatives. In order to show their advantages, from the previous data on tree climbing machines (Shamsi 1990; Niklen 1993), the proposed machine was compared with two existing machines; a tractor mounted lifting platform made in Iran and the date palm service machine made at Silsoe(Al-Suhaibani et al 1993). Some of the main improvements that can be achieved by the proposed machine were listed in Table 2.2.

Table 2.2 Comparison between proposed machine and existing systems

No	Feature	Existing systems	Proposed machine
1	Weight, kg	4000-8000	100-200
2	Payload, kg	100-200	25-50
3	Weight/payload ratio	20-80	2- 8
4	Power requirements, kW	50	2
5	Purchase price	from £ 16 000	cheaper than existing systems
6	Worker safety	-	better than existing systems
7	Maximum canal depth that can be negotiated, m	0.5	unlimited (using simple wooden bridge)
8	Maximum lifting height, m	12-15	unlimited
9	Turning circle radius, m	6	2
10	Width, m	3	1.5

From these data it can be concluded that even for the places that are using the existing systems it is possible that the proposed machine is a better alternative because of the following reasons:

1. The machine will be lighter and smaller than a truck and platform (lifter) and will weigh about 100-200 kg, when the truck and platform weights 4 to 8 tonnes. The comparison of weight and weight/payload ratio of existing and proposed systems

shows a significant weight reduction and efficiency improvements. The climber therefore can easily move in the groves where intensive cultivation are practised. If the irrigation channels are too deep for the machine to cross then the workers can pick it up and carry it over. Simple wooden bridges also can be used to allow the machine to pass over the channels.

2. Compare to the available mechanised methods, which were classified as "lifters" by Craig (1992), the proposed tree climbing machine will have few number of components and therefore it will be cheaper and easier to control than a "lifter" (truck and a platform).

3. The worker safety and severe working conditions will be improved because the worker does not climb the tree and can control the machine from the safety of the ground.

2.4. Product Design Specification (PDS)

The product design specification (PDS) (Pugh 1991) for the tree climbing date harvesting machine was developed by analysing disadvantages of the existed harvesting methods (Chapter 1) and considering technical, safety and economical issues for the new design. PDS was the main control criteria during the design stages and helped to select the right solution from the list of alternatives during the decision making phases of the design process. An objective tree (Figure 2-2) was employed to show and expand some of the design objectives of PDS to second, third or forth levels. It helped to find a means of achievement of those design objectives. For example expanding safety to the forth level showed that using a suitable traction element is a means to achieve the tree safety which is a PDS element. Considering required elements for PDS and using the above techniques following PDS was adopted for the tree climbing date harvesting machine:

Ability to move on the ground

A machine weight of about 100 to 200 kg (Shamsi 1991; Niklen 1993) is difficult to carry by one person, therefore, for the ease of transportation and moving from tree to tree the machine must be able to move on the ground. To achieve this goal it should be equipped with powered traction elements (wheels, track, etc.) to provide mobility on the ground. The ground speed should be near to the worker ground speed.

Ability to cross irrigation channels

In many parts of the world where dates has been planting for centuries the surface irrigation is adopted to water the trees. To cross the irrigation channels the machine should have a high ground clearance and suitable traction elements.

Small overall dimensions

The machine should have dimensions as small as possible to be able to pass between the close tree spacing in the groves.

Ability to work on uneven tree surfaces

The tree trunk is covered with leaf bases which, the machine must be able to move over. Values of the surface roughness (leaf bases dimensions) will be measured in a field test on trees in Iran.

Ability to work on different tree diameters

The tree diameter depends on the tree age and variety. To harvest different size ranges of tree the machine should be equipped with a mechanism which enables it to work on the tree diameter domain. This domain will be defined from field measurements.

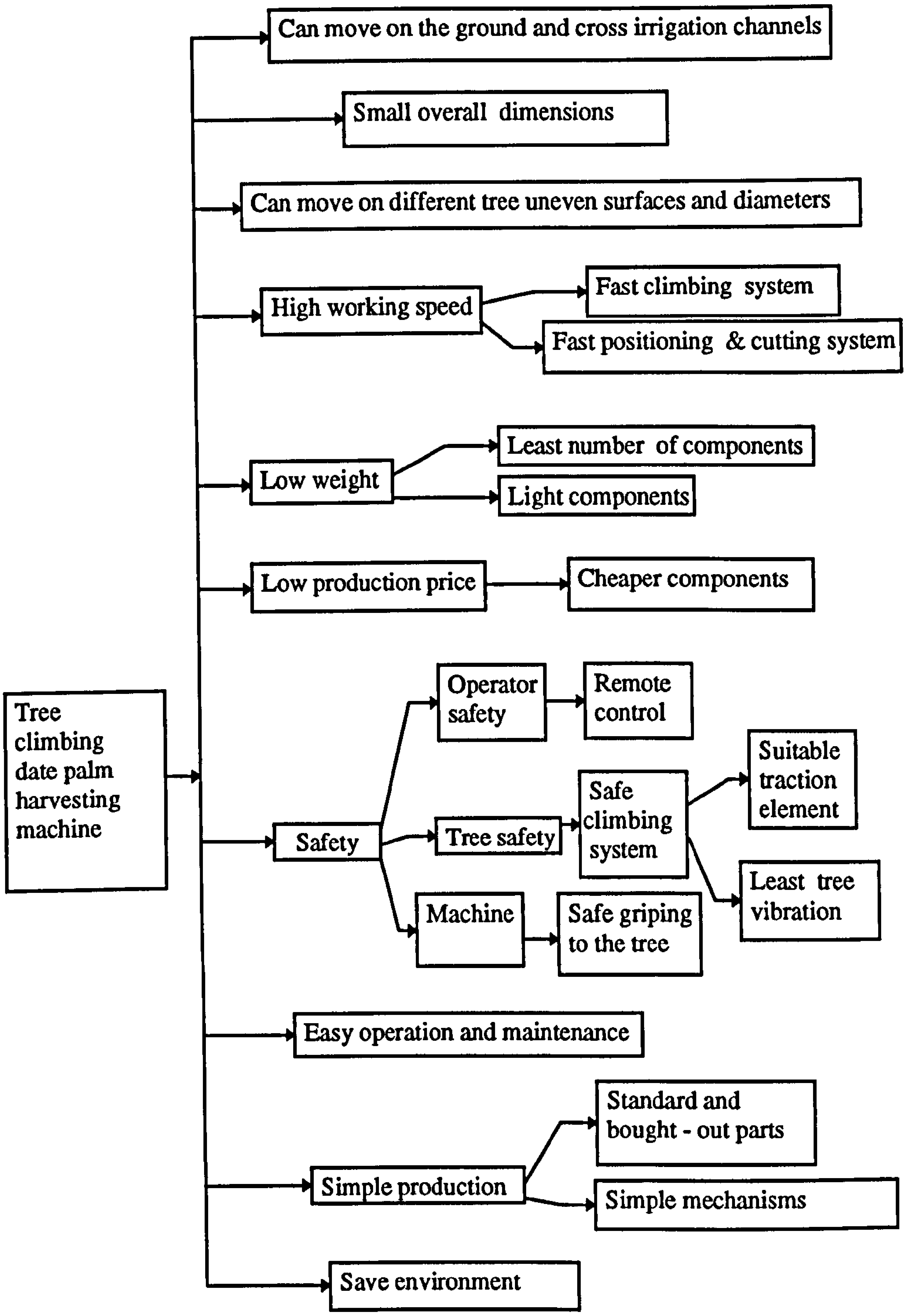


Figure 2-2 Objective tree for the tree climbing date harvesting machine

Fast climbing speed

The machine should climb the tree with a speed which is considerably higher than the worker climbing speed. This target value will be found from the field measurements. Fast climbing speed is one of the criteria for selecting climbing concept among alternative systems.

Fast positioning and fast cutting system

To achieve a high harvesting speed the positioning system which brings the cutter to the bunch stalk and the cutting system should move quickly.

Low weight

For the minimum damage to the tree and for the ease of transportation it is better to have the machine as light as possible. To achieve this goal the least number and light weight components must be selected for the machine.

Low production price

The machine price is an important factor. If it is cheap more farmers can afford to buy it, therefore, selecting cheaper components among alternatives was one of the strategies provided it did not limit the machine reliability.

Operator

To reduce the costs the machine should be able to do the main harvesting tasks with one operator.

Operator safety

The machine should provide operator safety better than hand harvesting and other mechanised harvesting methods. To achieve it the machine should be remotely controlled from the safety of the ground. Using high voltages should be avoided. Mechanical transmission systems and rotating shafts should have guards. In the manufacturing phase national safety standards should be applied to the product.

Tree safety

The machine should not damage the tree. The traction element is a key part which touches the tree. It should be suitable and should not exert stresses levels greater than the tree can tolerate. The tree stress limits should be determined by tests on real tree trunk and leaf base samples.

The machine should not vibrate the tree because this detaches individual dates from the bunches before the machine can harvest them.

Machine safety

To provide the machine safety the griping mechanism should be selected carefully to provide a robust and rigid attachment to the tree and will prevent the machine from detaching from the tree when it is climbing.

Ease of operation and maintenance

The machine should be easy to operate. Complex and difficult to operate systems should not be used in the design. Parts and mechanisms with minimum maintenance should be selected to make the machine suitable to use in remote areas.

Simple production

To produce the machine in local workshops using standard and bought-out parts and simple design solutions are required for the concept selections and parts design.

Save environment

Those power sources should have priority which are environmentally friendly and do not pollute the environment.

Machine harvesting capacity

Machine harvesting capacity in tree/h or kg/h should be equal or more than the hand harvesting method. The criterion is hand harvesting method because mechanised harvesting methods (lifters) are not suitable for many groves but if the machine can harvest faster than lifters it may be able replaces these systems as well.

Target cost

The machine cost should be low enough to replace the traditional harvesting methods. The machine harvesting cost must be less than or equal to hand harvesting method. This cost target value will be found from the field study of hand harvesting systems. The existing mechanised harvesting system in Iran costs approximately £16 000 (£10 000 for a tractor and £ 6 000 for a tractor operated lifter). The proposed machine should cost significantly less than this and with an enhanced performance, then it can be used instead of the existing machines in Iran and other countries.

Competition

The literature was searched and in the early phase of the study. Dr Brown, an international expert in date mechanisation systems and manager of date mechanisation project for the U.S. Department of Agriculture in 1964 was consulted by letter for feasibility and availability of tree climbing date harvesting machines. He said that a tree climber would be feasible, a market search also showed that there is no other competitor in the market at the moment and the machine will be a totally new product.

Ergonomics

All worker operating inputs should be at a the suitable height and loads on the worker should be within the capacity range of an average sized and healthy worker. The date basket should be at a suitable height so that the worker can easily reach the dates to unload the machine.

The above PDS was the design control criteria but the limiting values for some of them were unknown. A field study and set of experiments were conducted to find these values. Steps for this study is explained in Chapter 3 and results are used in Chapter 4, "detail design of experimental machine", to complete the design process.

2.5. Conceptual design

2.5.1. Establishing functions

Function analysis for the climbing date harvester was the main step and guide to generate alternative solutions which is shown in Figure 2-3. It illustrates the sequence of operations that the harvester must perform. Small squares show the OR statement and dotted lines show less probable choices of the next operation. "Take position" is small up and down, left and right movements of the climbing unit or the cutting unit before the cutter moves to grab and cut a bunch.

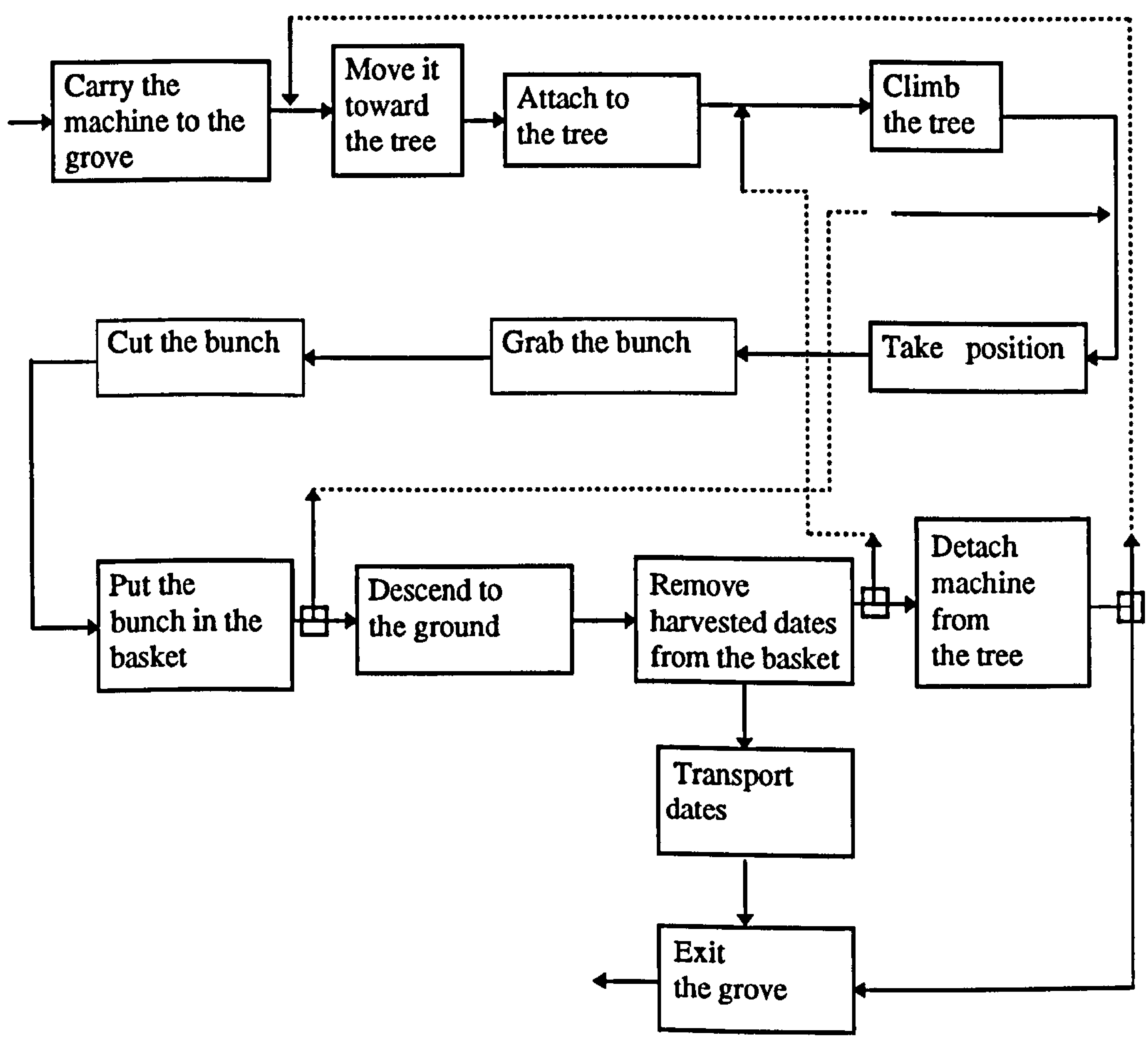


Figure 2-3 Function analysis of the tree climbing harvesting machine

2.5.2. Concept generation

The goal is to generate the complete range of alternative design solutions for the machine, and hence to widen the research for potential new solutions. A "morphological chart" was used to illustrate the concept solutions. It includes a classification of all available and conceptual solutions applicable to the design. To generate these concept solutions several aids were used such as: literature search, analysis of existing systems, analogy, design catalogues and help from experts.

The function analysis (Figure 2-3) illustrates the operations that the machine should perform to harvest dates. These operations were rearranged into the groups. Based on these groups a "morphological chart" was employed to illustrate alternative solutions for functions of each group (Figure 2-4). The groups and the rows of the chart allocated to each group are as follows:

Climbing: row A to H

Attach to, and detach from, the tree: row I and J

Positioning: row K to M

Cutting: row N

Descending: row O

Transport: row P to R

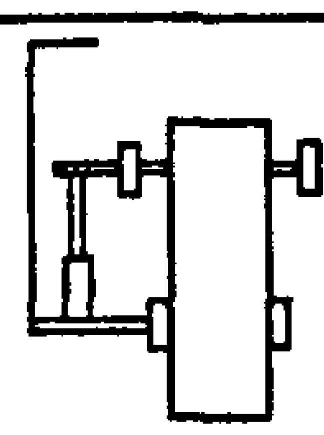
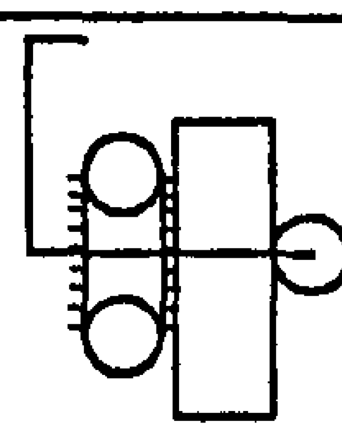
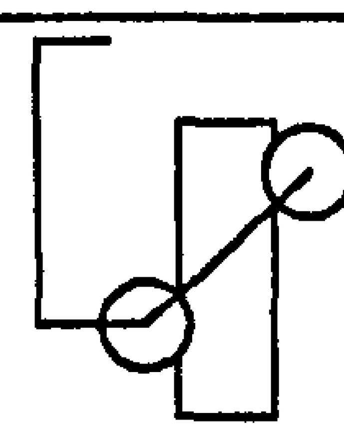
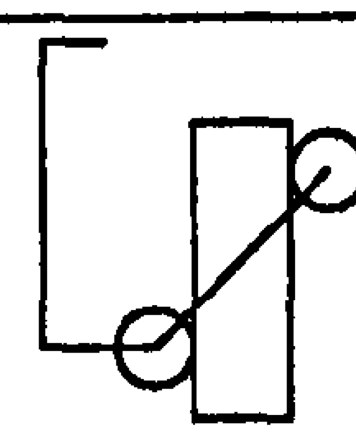
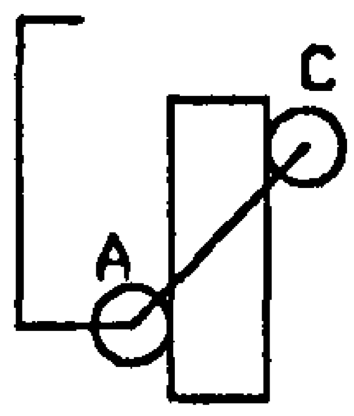
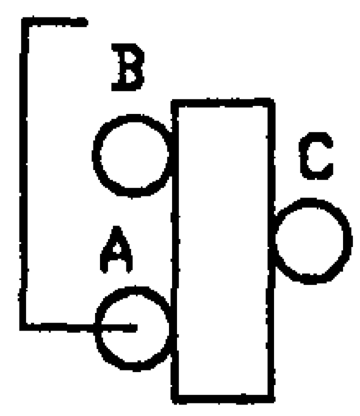
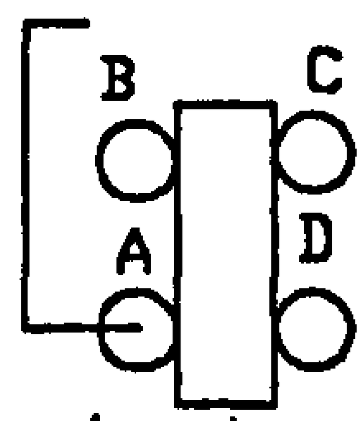
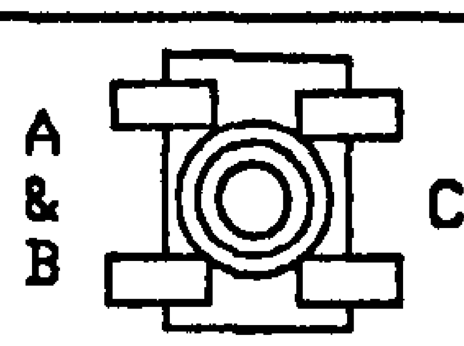
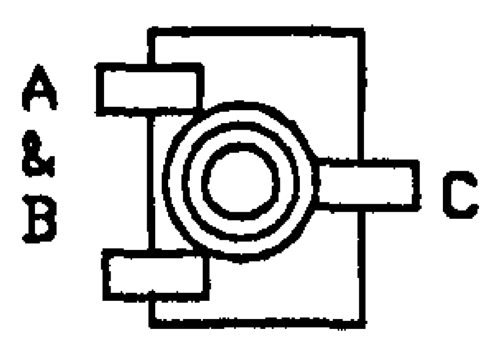
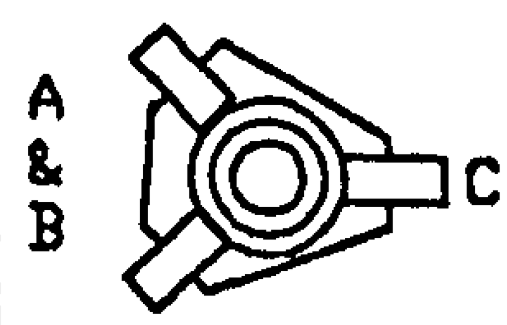
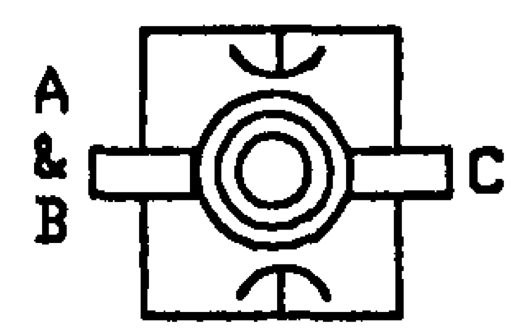
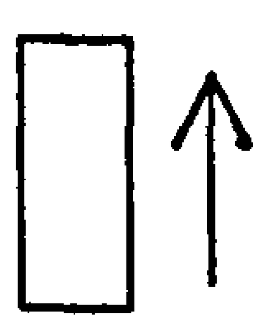
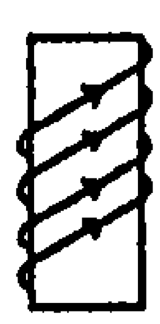
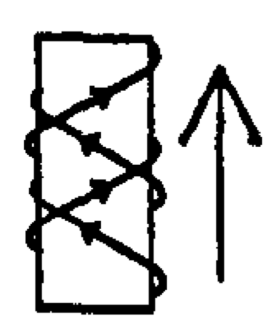
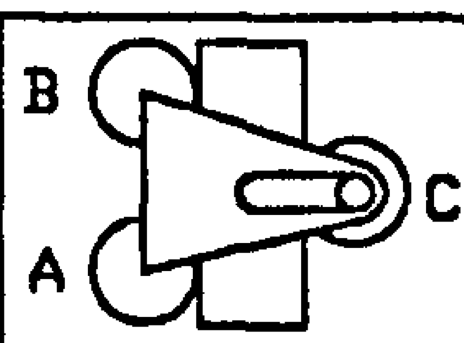
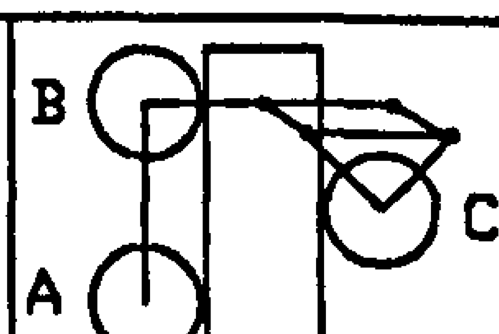
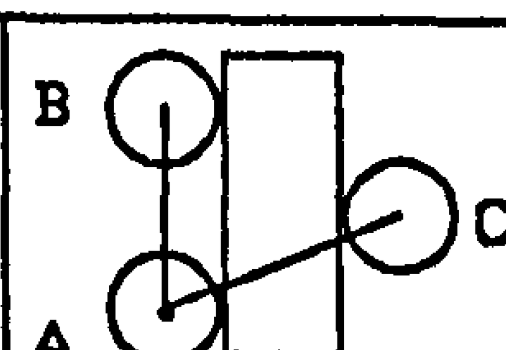
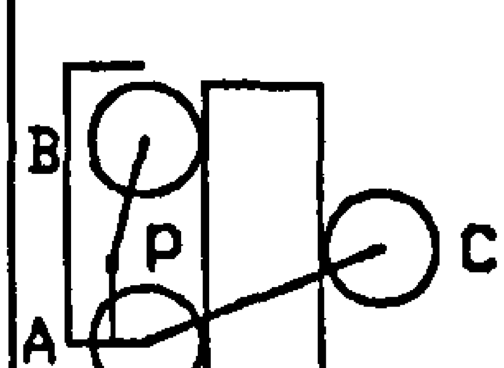
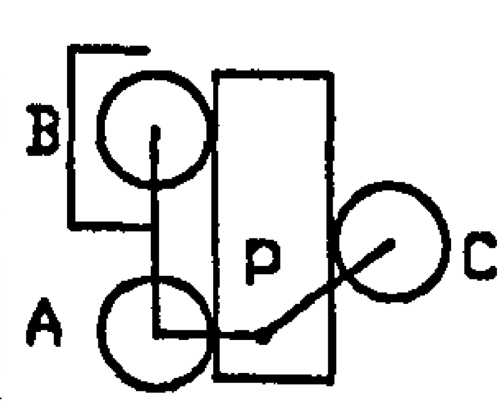
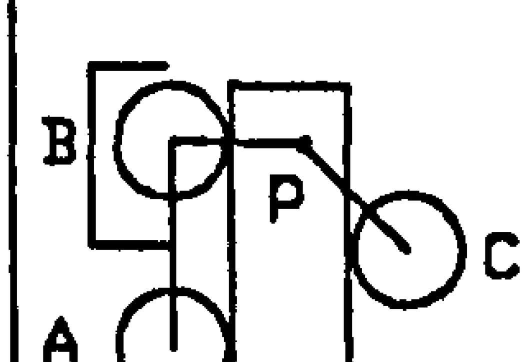
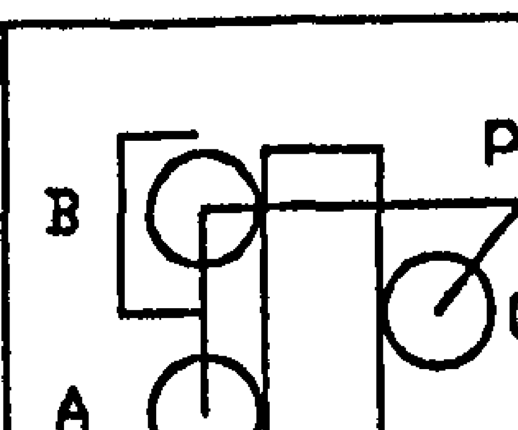
Power: row S to W

Control: row X and Y

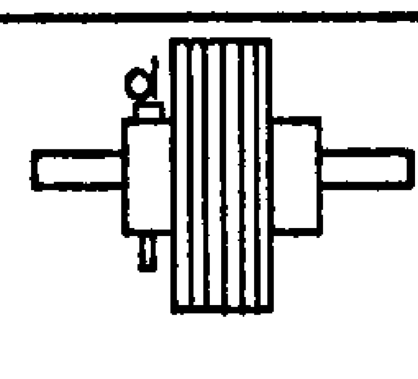
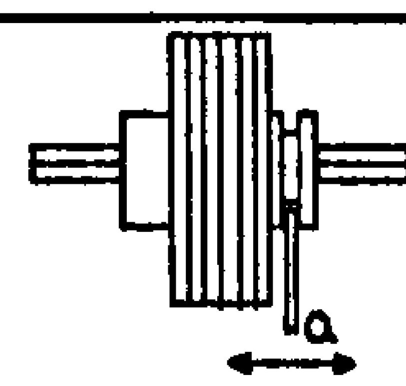
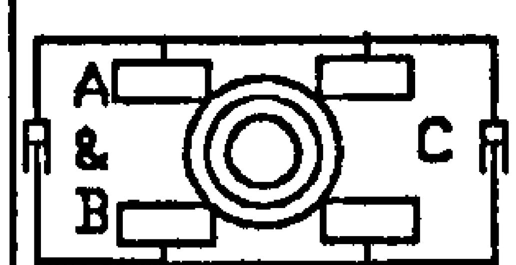
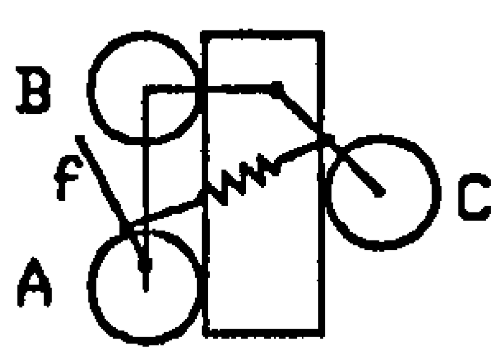
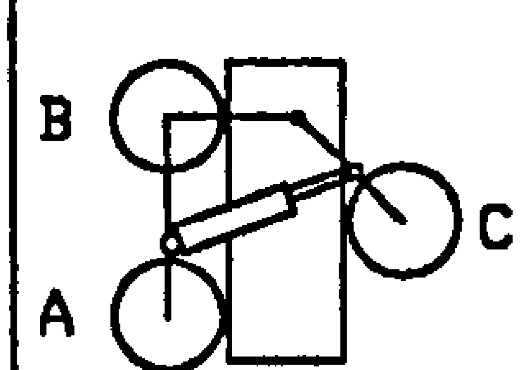
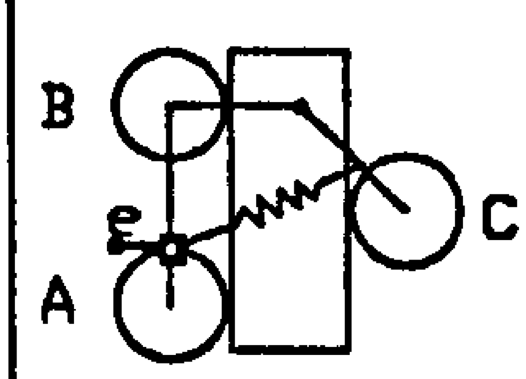
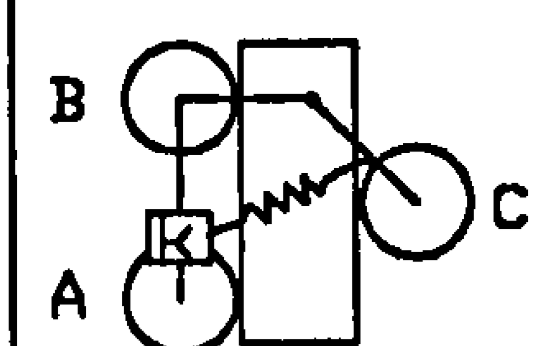
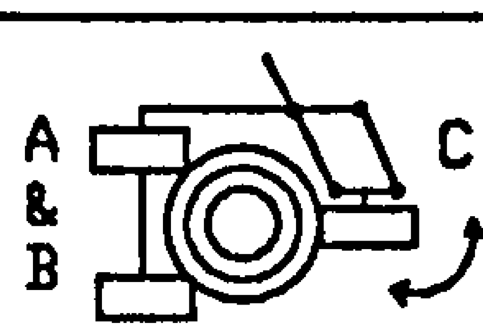
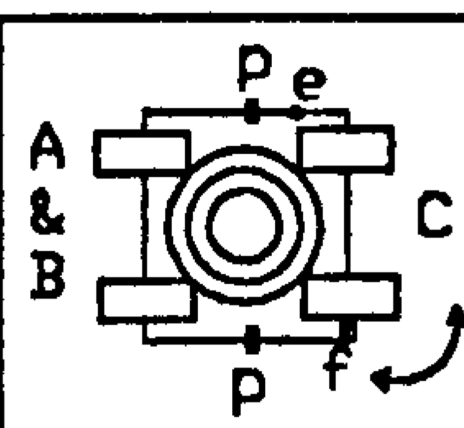
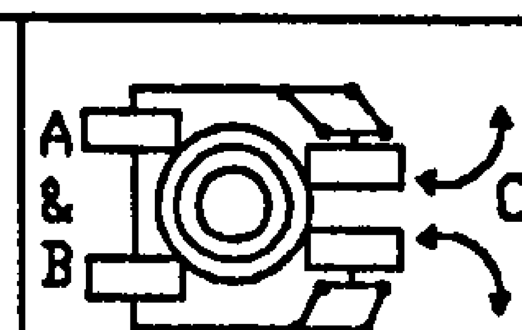
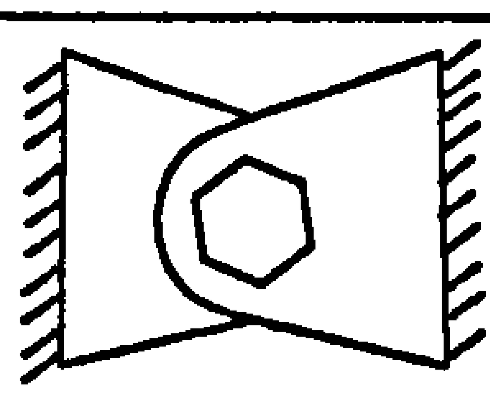
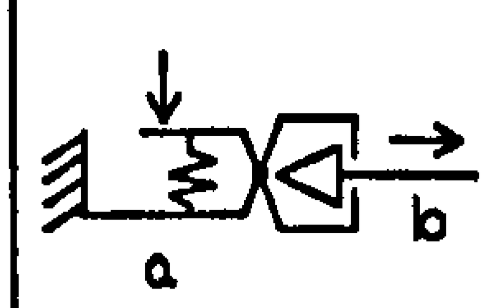
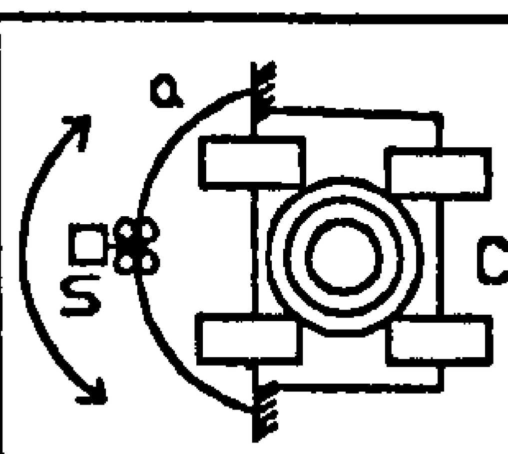
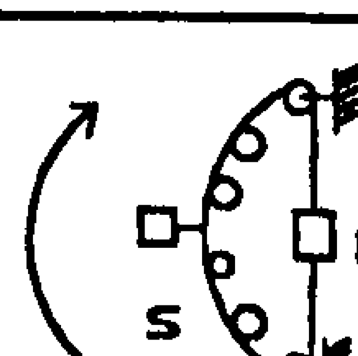
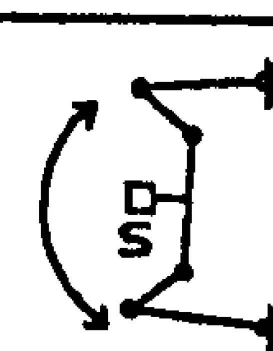
Braking: row Z

Transmission: row AA

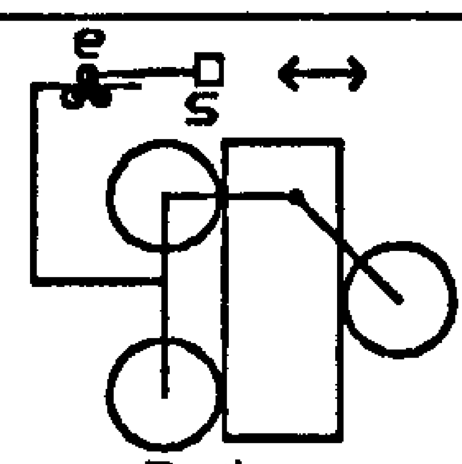
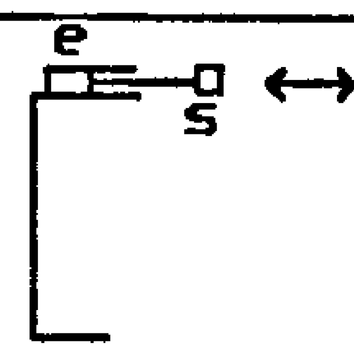
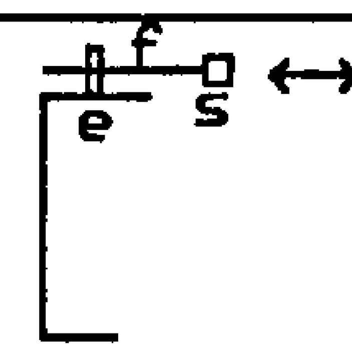
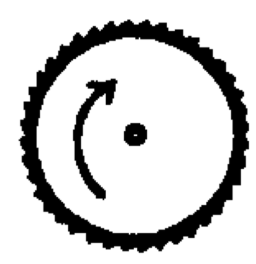
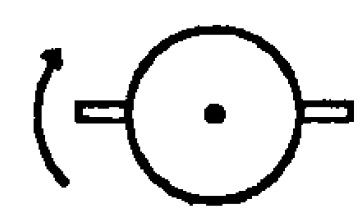
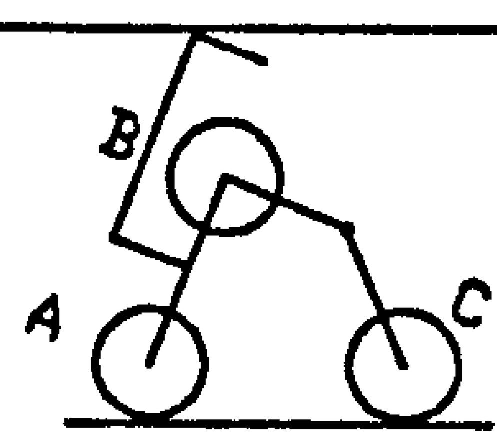
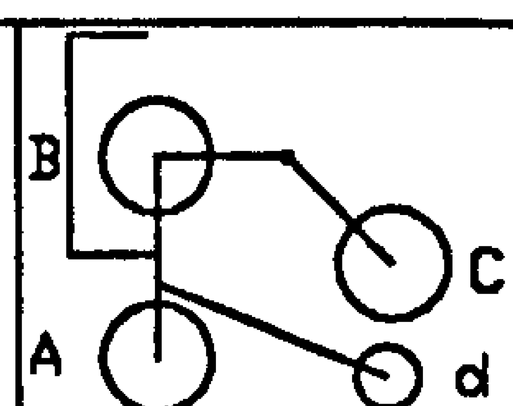
Figure 2-4 Morphological chart for the tree climbing date harvesting machine

	Function	Solutions 1	2	3	4
A	Traction	 Leg	 Track	 Concave roller	 Wheel
B	Vertical plane stability	 2 axles	 3 axles	 4 axles	
C	Horizontal plane stability	 6 wheels	 5 wheels	 6 wheels oblique	 Guiding shoes
D	Climbing method	 Vertical	 Spiral	 Vertical and two way spiral	
E	Diameter adjustment in vertical plane	 Slider mechanism	 multi linkage	 Arm and hinge	
F	Hinge position				

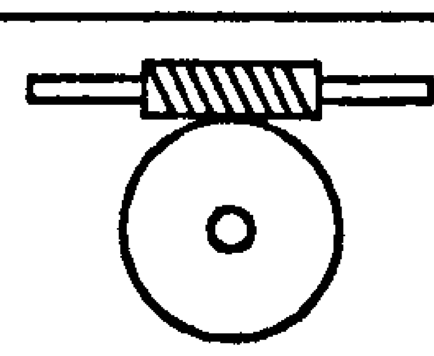
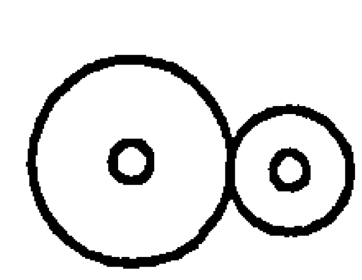
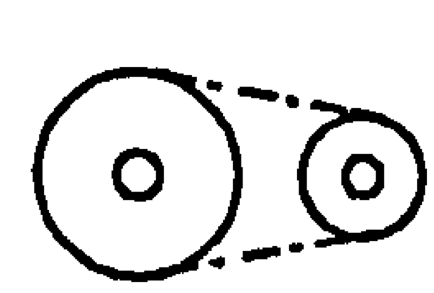
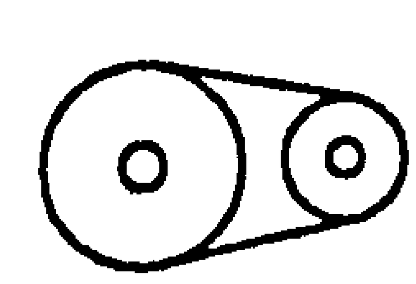
Morphological chart for the tree climbing date harvesting machine (continued)

	Function	Solutions			
		1	2	3	4
G	Diameter adjustment in horizontal plane	 Manual	 Automatic	 Chassis move	
H	Traction control	 Lever	 Linear actuation	 Hand winch	 Powered winch
I	Attach to and detach from the tree	 Multi linkage 5 wheels	 axle rotation	 Multi linkage 6 wheels	
J	Locking 'one axle unit'	 Bolt and nut	 Quick latch		
K	positioning	Climber turn	Cutter arm turn	Climber and cutter arm turn	Machine repositioning and cutter arm turn
L	Cutter arm positioning	 Rail	 Pulleys	 Multi linkage mechanism	

Morphological chart for the tree climbing date harvesting machine (continued)

	Function	Solutions			
		1	2	3	4
M	Cutter positioning	 Rail	 Linear actuation	 Screw	Rack and pinion
N	Bunch cut	 Circular saw	 Chain saw	 Shear blades	 Rotating blades
O	Lower dates	On machine	On winch		
P	Transport mode	 Rotation	 Stabilizers		
Q	Steering on the ground	Fixed wheels	Caster wheels	Turning wheels	
R	Transport	Towing	Special trailer	Van	Tractor trailer

Morphological chart for the tree climbing date harvesting machine (continued)

	Function	Solutions			
		1	2	3	4
S	Power	Electric	Petrol	Diesel	Self powered
T	Electric Motor type	DC motor	AC motor		
U	DC type	Series	Shunt	Permanent magnet	Others (explained In the text)
V	Electric power source	Batteries	Generator	Tractor batteries	Grove main electric power
W	Batteries place	On board	On the ground		
X	Control	Electric	Electronic (PLC)	Mechanical	Hydraulic
Y	Control signals Transmission	wire	Radio		
Z	Braking	On motor	Along the transmission system	On driving axle	
AA	Transmission	 Worm and gear	 Simple gear	 Chain	 Belt

2.5.3. Concept selection

The concept for each function was selected from the morphological chart using the design objectives and specifications (PDS). The advantage and disadvantages of each concept was mentioned carefully and feasibility of some concepts were checked using mathematical and physical model analysis.

(A) Traction

Row A shows different traction elements that could be fitted on a tree climbing machine. The angled line represents the cutter unit of the machine. Legs (shown in cell A1) are the traction element of walking machines. The main advantage of legs is their ability to climb over obstacles and uneven surfaces. A legged vehicle can achieve a smooth ride on a rough tree surface. The zigzag shape of the tree surface can also function as an anchorage for the leg and can help it to climb, but, causes extra rolling resistance for continuos motion elements like wheels or rollers. Legged vehicles are complex and expensive because of the actuation and control system (Todd 1985). A legged climbing machine needs an extra mechanism to relocate the frame to provide the ground motion for the climber. Legs usually move slower than wheels.

The main advantage of tracks as the traction element is providing a traction force bigger than wheels or rollers if they are designed to carry an equal load (Liljedahl et al 1989). Tracks were rejected because they move slower than wheels and rollers and are more complex and expensive elements. It is partly because of the shape and need for a number of idler rollers on the chassis.

The main advantage of the concave roller is that it serves as both a traction element and a lateral movement controller. The rounded inwards shape of the roller prevents the machine from moving to one side of the tree. Disadvantages of rollers are firstly, stress concentration at the contact points of roller and tree. Nicklin (1993) reported this problem, which damaged the rubber layer of the driving rollers which was used to reduce stress level on the tree surface. It can cause the same problem to the date tree trunk. Secondly, rollers are not suitable for moving on the ground and were rejected because of possible damage to the tree and ground mobility problem.

A wheel (A4) with a pneumatic tyre is another alternative. It was selected among the concepts because it minimises the stress concentration and exerts a more uniform stress on the tree surface. When the load on the wheel increases its contact surface with the tree expands and keeps the stress level constant and equal to the tyre inflation pressure. This feature decreases the tree damage and is the unique advantage of the wheel. A wheel is cheaper than a track and a leg. It can be used for the ground motion without relocating the wheels position. In fact considering the design objectives wheels were selected because of, providing the tree safety, suitable price, easily available bought-out part, simple mechanism and good ground mobility.

(B) Vertical plane stability

Row B in the morphological chart shows the different layouts of wheels to solve the vertical plane stability. B is the simplest model for a climbing machine. This model was used by Nicklin (1993) and Davis (1979) to lift a person to the tree crown. As it is a simple model a mathematical analysis was used to check if it is suitable for remote control date harvester.

Figure 2-5 shows the model which has axles A and C. It carries the total weight of W. If axle A is driving and B idle from the static equilibrium (Meriam & Craige 1993) the following equations can be derived:

$$\sum F_x = 0 \Rightarrow A_x = C_x$$

$$\sum F_y = 0 \Rightarrow W = A_y \qquad (2.1)$$

$$\sum M_A = 0 \Rightarrow C_x h = x W$$

$$A_x = C_x = W(x/h) \qquad (2.2)$$

Equation 2.2 is valid when :

$$A_y \leq \mu A_x \qquad (2.3)$$

where μ is the coefficient of friction between the tree and the tyre.

If the above condition are not satisfied the machine will slip down the tree.

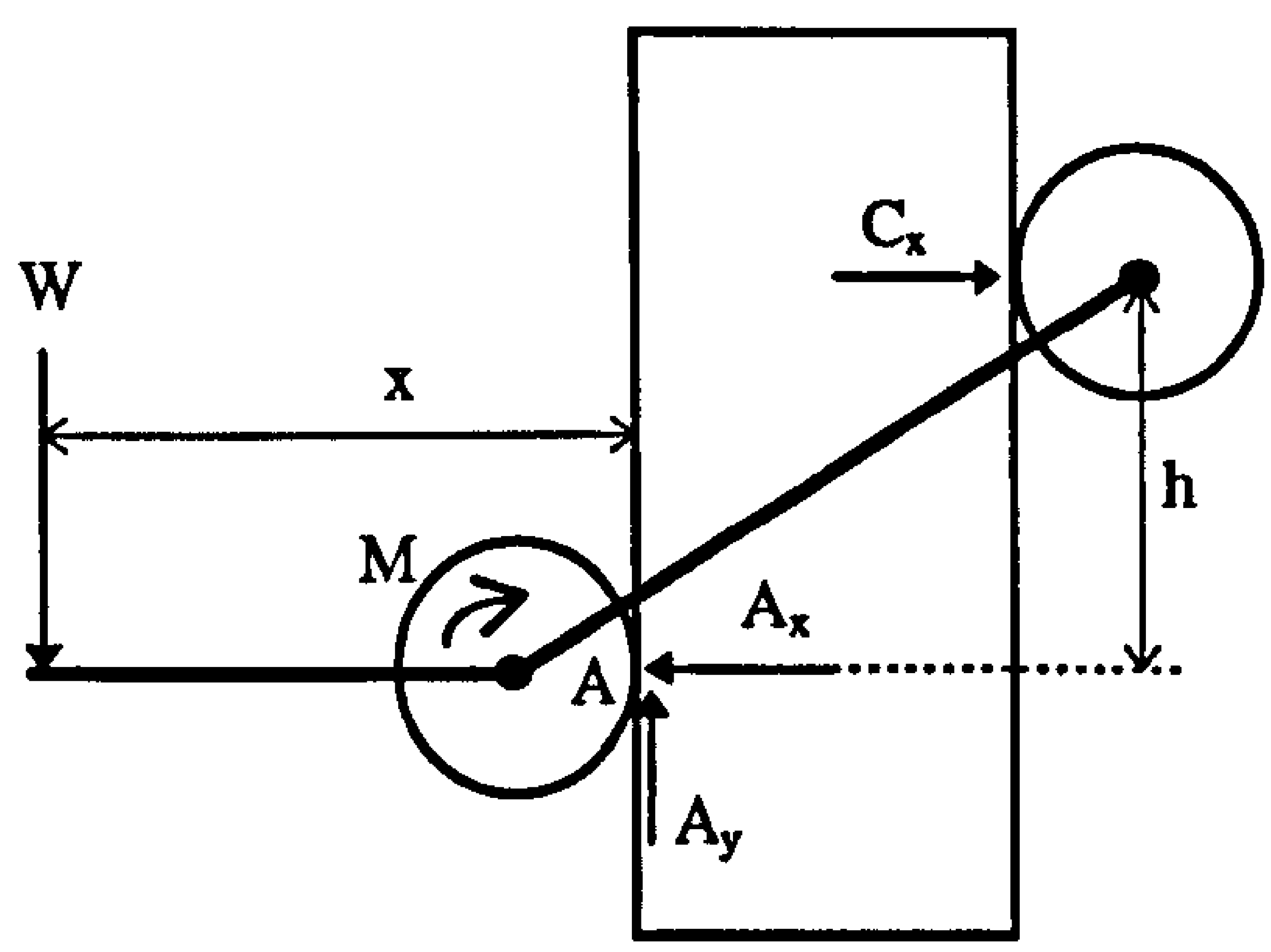


Figure 2-5 Free body diagram of the two wheel model

Substituting Equation 2.1 and Equation 2.2 in Equation 2.3 thus working condition for the machine can be shown as follows:

$$W \leq \mu W (x / h)$$

or

$$x \geq (h / \mu) \tag{2.4}$$

Equation 2.4 is an important equation because it is independent of loading and relates the coefficient of friction to the machine main dimensions. It shows the condition that is required for the machine to be able to work. The following results can be concluded from this analysis:

1. It is a one side loaded model and as a clockwise moment will detach the machine from the tree, it is not stable in the vertical plane.

2. Equation 2.2 shows that for a particular dimension only the machine weight, W controls the traction force, therefore if the machine is carrying a known load and

driving wheels fail to climb and start to spin it is not possible to increase the wheel force against the tree to increase the traction force to over come the problem.

3. Equation 2.4 shows that the dimensions of the machine play an important role in the model and create dimension limits for it. For example if the coefficient of friction is 0.7 and the wheels height difference, h, is 0.5 m then the length of centre of gravity, x, should be greater than 0.7 m. If these conditions are not satisfied the machine will slip down the tree.

4. The machine is one side loaded. It will not work if loaded from the other side.

5. The harvester arm and bunch cutter are displaced from the original position as the tree diameter changes.

The above results showed that the model B is not suitable for a remote control date harvester. B2 is a three axle model, axles A and B one side and axle C the other side of the tree at a height between axles A and B which is the minimum axles requirement for resisting the moment and having stability in the vertical plane. To increase the stability and to provide a better guiding system, 4 axle model such as B3 can be used. The model B2 which can provide the requirements was selected for the simplicity. Decisions on the number of wheels on each axle considering horizontal plane stability is made in the next section.

(C) Horizontal plane stability

Row C of the morphological chart shows different solutions for horizontal plane stability for the selected layout B2. It can be achieved with 2 wheels on each axle, or the total of 6 wheels (layout C1) it is also possible to have one wheel on axle C, but there is some doubt that it will work correctly. C3 is the possible layout with 5 wheels with bent axles A and B, in fact 5 wheels and 5 axles, this arrangement improves the problem of tyres side wear but this layout can not move on the ground. Shaft supports, diameter adjustment mechanism and power transmission will be more complex and expensive if axles A and B are driving axles which are contrary to the design objectives. C4 is a layout with one wheel on each axle with guide shoes to control the horizontal plane stability. This layout is a cheaper solution but guiding shoes cause extra friction and may damage the tree surface. the layout is not suitable for ground locomotion.

Layout C1 (two wheels on each axle) was selected for the design. One wheel on axle C gives a simpler and cheaper solution, so the test rig will be designed to enable layout C2 to be tested as well.

(D) Climbing methods

Three methods of climbing are shown in row D. Vertical climbing D1, demands the simplest mechanical devices and is the shortest way of reaching the dates. Spiral motion (D2) is used by pruning machines to reach branches around the tree trunk but this is not necessary for date harvesting. This method can not be used when the machine is being powered from the ground by hydraulic or pneumatic tubes or electric wires because they would also turn around the tree causing problems. Two direction spiral and vertical motion, D3, gives the highest flexibility to the machine . It gives the ability of "positioning" to climbing unit, but it is more complex than spiral motion (D2). Turning wheels on the tree especially in the static condition can damage the tree surface, therefore it is safer, cheaper and simpler to give the positioning job to the harvester arm. The harvester arm is a light part which includes an arm and a cutting device while the climbing unit is the heavy part. The harvester arm can turn around on its base to find the bunch, thus consuming less energy than turning the whole climber around the tree. The tree safety also increases with this method because the climber does not move on the tree for positioning. The limitation is that only half of the tree can be harvested and providing a mechanism which enables the harvesting arm to turn 360° around the tree is not easily possible. Therefore, the vertical method (D1) is selected, with the assumption of harvesting half of the tree at each run and transfer "positioning" to the "harvesting arm".

(E) Diameter adjustment in vertical plane

The machine should be equipped with a mechanism to adjust the wheel spacing according to the tree diameter both in vertical and horizontal planes. Three methods were considered for adjustments in vertical plane as E1, E2 and E3 (Artobolevisky 1975). E1 is slider mechanism that provides the facility of sliding axle C in a pair of grooves for diameter adjustment. It is a perfect mechanism for moving on a straight line or any curve type path. One disadvantages is the requirement for an extra mechanism to ensure that both ends of the axle move simultaneously, without this mechanism the slider mechanism can not work. Extra friction in the moving joints and requirement of lubrication are other problems. A multi-linkage mechanism (E2) is another alternative which can provide any path of movement. These mechanisms were found to be complex and a simpler solution was required. An arm which can rotate around a hinge located at point such as centre of axle A, would be a simple and effective solution (E3) and was selected for the design.

(F) Horizontal hinge position

The location of the hinge has an important effects on the machine performance. F1 shows hinge P near axle A, it is actually model B1 with the extra axle B, it inherits problems of B1 and is not a good choice. F2 has axle C connected to the rest of the machine by an arm and hinge P. This model is good for ground travel because axle C can be turned and fixed in a position level with axle A, then the machine has four wheels (axle A and C) on the ground. Low ground clearance is a disadvantage for the model because it limits the ability to cross uneven surfaces as the chassis touches the ground. The other problem is decreasing vertical distance between axle A and C with increasing the tree diameter. It increases the force, A_x, by which driving wheels A are pushed on the tree and it may also damage the tree. The mathematical model of the concept solution B1 which is partially valid for this analysis was employed to clarify the situation.

Re writing Equation 2.2:

$$A_x = w(x/h)$$

shows that A_x increases when W increases and when h decreases. Mature trees have large diameters and have more yield than a young or small diameter tree, therefore W is bigger and h is smaller on a big diameter which result in a large A_x which might damage the tree. The other weakness of the concept is low ground clearance, therefore the concept was rejected. F3 shows hinge P positioned near axle B, the constraint of low ground clearance is resolved and the tree damage which is a problem for F2 is changed to an advantage because with this position h increases with the tree diameter. F4 is a concept with hinge P placed on the right hand side of axle C for all the range of tree diameters. This position of P offers a self griping system to the tree if axles A and C both are driving and wheels of axle C rotate a bit faster than axle A. This system is interesting but needs an intelligent system to control the griping force. The concept was not considered as griping can be provided simper by a pair of springs. The F3 layout was selected for its explained advantages and also for its simplicity and safety.

(G) Diameter adjustment in horizontal plane

G1 offers a simple manual method to set the wheel spacing between two wheels of each axle. Pin (d) passes through the rim hub and driving shaft holes. The driving shaft should be drilled in equal spacing to obtain the desired wheel spacing.

G2 is an automatic version of G1. The driving shaft has a hexagonal cross section and wheel can move axially on it but cannot rotate relative to it. Powered guiding tool, a, placed in the rim hub groove and can move the wheel to achieve the required spacing. It is easier and faster but more expensive. In the G3 method the wheels are in a fixed position but the two parts of the chassis move to create the desired spacing. It is the quickest adjustment method because all wheels are adjusted simultaneously, but it is complex. For the matter of simplicity, reliability and cheapness G1 was selected.

(H) Traction control

An adjustable system to force the wheels in contact with the tree is required to gain traction. H1 is a lever mechanism which pulls a spring to produce this force. Setting the lever at different positions controls the force of the wheels. H2 is a linear actuator which can be powered by hydraulic, pneumatic or electric systems and has a sensor to control the force. It is a quick but expensive system. H3 is a hand winch, which tensions the spring with a steel rope. H4 is a powered winch, which can tension the spring very quickly. H2 and H4 were rejected because of the complexity. A decision between H1 and H3 will be made after knowing the lever displacement range considering maximum and minimum diameter tree and the force required to pull the lever. The lever mechanism or H1 is simpler and cheaper than hand winch, H2, but if the displacement or the force to pull the lever is too large it will be replaced with H2.

(I) Attach and detach from the tree

I1 and I3 use a 4 bar parallel linkage for ease of attachment to the tree. These mechanisms provide a quick attachment which improve the harvesting time but they are not compatible with the selected concept, F3, as rotation of axle C around pivot p and its movement in the plane of the 4 bar linkage when it is being attached to the tree creates interaction problems. I2 was selected, turning axle C and its supports around point (e) and opening "one axle unit" as a gate. The operation is reversed to detach the machine from the tree.

(J) Locking "one axle unit"

Selected I2 requires a lock at point (f) of this model. J1 and J2 are two types of these locks or connections. A bolt and nut (J1) was selected for the ease of manufacturing, but a quick attach latch may be required later if tests show that connection and disconnection times are too great.

(k) Positioning

Vertical climbing, D1, was selected, for the machine so methods K1 and K3 are not feasible because they need the spiral motion of the climbing system. K2 or "only the cutter arm turn" cannot be used because a complex system should be designed to turn the cutter 360° around the tree and it interferes with the attachment to and detachment from the tree which requires extra complex mechanisms.

K4 is applicable to the design but the machine only harvests half the bunches, returns to the ground, and after unloading dates, is attached to the tree from the opposite side and climbs to harvest the other half of the tree. It is a lot easier to design the cutter arm for rotation of 180° instead of 360° and it does not interfere with the attachment and detachment mechanisms.

(L) Cutter arm positioning

To cut a bunch the climber moves to a height so that the cutter (s) is level with bunch stalk. The Rail mechanism, L1, can be employed to move the cutter in the horizontal plane to reach the required position. L2 is a motor (m), pulleys and rope system to bring the cutter (s) near the bunch. L3 is a multi linkage mechanism, to provide a half circular movement path for cutter(s). It requires more than one degree of freedom which is more expensive and difficult to control than L1 and L2. L1 was selected for the machine because it is more reliable than L2 due to the complexity of the detail rope and pulley system.

(M) Cutter positioning in vertical plane

The last movement before cutting a bunch is cutter positioning in the vertical plane. In solution M1 the cutter frame has small wheels which move on a rail by the power of a small motor. A linear actuator (M2) was also considered. Solution M3 is screw F connected to the cutter (s) which can move forward and backward by motor (e) and finally a rack and pinion mechanism (M4) can be employed to generate the linear movement for (s). M2 was selected because of the simplicity and being a standard and bought-out part available for electric, hydraulic or pneumatic systems.

(N) Bunch cut

Circular saw (N1) is a simple method and was used by Roux et at (1994) on a tree pruning machine to cut extra branches of the tree. A chain saw (N2) was mounted on a tree climbing pruning machine made by Serei Koyo (1996) Company in Japan. Personal experience showed that chain saw are not suitable for cutting date bunches because the wet bunch fibres jam the saw chain.

Shear blades (N3) are a well known device to cut stalks and tree branches and are produced in manual and powered versions. Rotating blades (N4) work on the impact principles and requires a high peripheral speed from 51 to 76 m/s. It is not a suitable method because of the impact force which shakes and detaches dates from the bunch. Circular saw (N1) and shear blades (N3) both are acceptable methods but N3 was selected because two blades approach to the stalk from opposite sides to grip it positively which a circular saw does not have this characteristic and may require an additional holding device.

(O) Lower dates

Two concepts for lowering dates were studied. O1 is keeping dates in a basket and lower them when machine did harvest one side of the tree or half of the yield. O2 is lowering the yield bunch by bunch using an automatic winch. This method reduces the machine weight and required power significantly because it does not carry any dates, but it was rejected because the required time to lower bunches was considerable and the winch had a complex design. Concept O1 was selected which is lowering after harvesting half of the yield. A light basket holds the cut bunches.

(P) Transport mode

P1 shows the machine rotated around axle A so that axle C is on the ground. P2 shows the second concept of transport mode which stabilisers D hold the machine on the ground in the correct position for climbing. P2 is more complex than P1 as it requires a pair of stabiliser wheel and frames but it was selected because it is easier and safer for the worker to attach the machine to the tree with stabiliser wheels. To attach the machine to the tree without stabilisers, axle C is lifted from the ground and turned around the horizontal hinge (selection L2). This is a difficult and unsafe job for the worker because at the same time that he is holding the machine weight he must operate the " one axle unit" lock. P2 solves this problem because machine is on stabilisers and is already in climbing position. The second advantage of P2 is cutter safety because with P1 when machine is being connected the cutter might touch the tree and cause damage.

(Q) Steering on the ground

Stabilisers shown in cell P2 can have fixed wheels (Q1) or free caster wheels (Q2) or operator controlled turning wheels (Q3). As the machine is light and the operator walks behind the machine he can easily push the machine to right or left therefore Q1 was selected If the test machine shows that it is difficult to steer the machine, Q2 and then Q3 will be chosen.

(R) Transport the machine

To transport the machine from one grove to another, method R1, towing by a tractor or a car is possible. Method R2 uses a special trailer, designed for the harvesting machine. R3 and R4 use a van or tractor to move the machine for long journeys. The machine can be moved into a van or tractor trailer using two small wooden ramps. As both vehicles are usually available to transport dates to stores and processing factories, R3 and R4 both are suitable solutions.

(S) Power

The climber can be powered by electric motors (S1) or petrol and diesel engines (S2 & S3). The choice of self powering (S4) was also considered as a method. S4 is using the potential energy of dates, which is equal to the date weight multiplied by their height from the ground. This energy gain and can be stored when machine lowers harvested bunches to the ground. This is a natural and environmental friendly energy. Machine potential energy is also considerable. These energies can be saved and used to supply part or all the required energy to harvest dates, but it was not considered in this stage of the design, as the validity of the main idea is not proven. Petrol engine (S2) is better than diesel (S3) because it is easier to use and cheaper and lighter. Neither S2 nor S3 was chosen for the design because these engines vibrate the tree and it may separate some of the dates from bunches before machine can harvest them. For bunch shaking harvesting method explained in Chapter 1, one of the devices is a 7 to 8 kg hydraulic powered vibrator which can be carried up to the tree by a worker. It is attached to the bunch and produces 600 to 1100 pulses per minute to shake and harvest dates (Perkins & Brown 1964). This example shows that using S2 or S3 on the machine was not a good choice. Electric motor (S1) is accepted as the best alternative because it is principally vibration free. There are many other advantages for this concept. It is simpler than a petrol engine and can be easily controlled by a small and light electronic controller, and no need for extra equipment for reverse motion, a clutch system or variable speed gear box. It is more reliable than S2 and S3, because, there is only one moving part, the rotor shaft. It is easy to maintain because there is no cooling, exhaust, ignition, fuel and lubrication systems. It is a very flexible system which in emergency can use different power sources such as batteries, the grove main power supply, tractor and car batteries and portable generators. Considering the progress of renewable energy projects in Iran to produce electricity from sources such as water, wind and sun, an electric motor saves the environment and prevents the green house effect because it does not generate carbon dioxide (CO_2) and other polluting gases and does not consume oxygen (O_2) and it does not need to toxic liquids such as antifreeze to work (Brant 1994).

(T) Electric motor type

T1 and T2 are DC and AC electric motors. T1 was selected for the machine because a DC motor can work with low voltages of 12 or 24 volt batteries which are standard and readily available part of electric vehicles, cars and tractors.

The operator safety is guaranteed with this voltage and it is much safer than usual 110 or 220 volt AC motors. Electronic controllers also have developed faster for DC motors than AC ones (Brant 1994). A catalogue and enquiry search showed many companies which make electric vehicles in the range of 1 to 4 kW power such as "Sun Gift" (1996) and "Batri Car" (1996) use DC permanent magnet motors for their products.

(U) DC motor type

Among different types of DC electric motors which are, series, shunt, compound, permanent magnet, brushless and universal, the permanent magnet was selected. The criteria of selection were simplicity, lightness, efficiency and high starting torque. These motors are increasingly used today because new technology (various alloys of Alnico magnet material, ferrite-ceramic magnets, etc.) enables them to be made smaller and lighter in weight than equivalent wound field motors (such as series and shunt) of the same power rating. While commutator and brushes are still required, the complexity and expenses of fabricating a field winding is saved and efficiency is gained because no current is needed for the field magnets (Brant 1994).

(V) Electric power source

The electric power source for the machine can be batteries (V1) or portable generator (V2) and it is possible to use the tractor electric system (V3) as tractors with 12 or 24 volt electric systems are usually available in the groves for transporting dates and other agricultural operations. It is also possible to use the main power supply of the grove (V4). The best alternative is (V1) but as suitable sized batteries do not last for eight hours of a working day, extra sets of batteries are required. The second priority (V2) is using a portable generator which is placed on the ground and the climber is connected to it by wires. The possibility of using long wires will be checked with the test machine. The advantage of V2 to V3 and V4 is that it makes the machine totally independent of other power sources but the machine still has the chance to use V1, V3 and V6, if they are available.

(W) Batteries position

As the ability of the model to lift loads was not completely clear at this stage of the design therefore the light weight of the machine was very important. To keep the weight low W2, having batteries on the ground, was selected. The machine is connected to the batteries by wires. Wires are unlikely to turn around the tree and cause a problem because the model climbs the tree vertically.

(X) Control

To control the chosen DC motor, X3 and X4 are irrelevant concepts. X1 is the simplest and cheapest concept because for a basic climber an on-off and a forward-reverse switch is enough, but to increase the motor safety against excessive current and to provide different speeds an electronic controller, X2, is the best choice for the machine. A programmable logic controller (PLC) can also control the motor parameters such as, speed, maximum current, acceleration, reverse motion and the electric brake. The brake control functions are acting and releasing and delay time to act after the motor is switched off. PLCs have replaced relays and electric switches in many industrial systems. It is computer programmed to implement logic functions (Auslander & Kempf 1996). Using electric motors with an electronic controller (PLC) is a mechatronic approach to the design which, compared to the traditional mechanical method, offers a compact system, simplified mechanism , programmable movements, variable speed drives, electronic synchronisation, light structures, accuracy achieved by feed back and automatic and programmable controls (Bradley et al 1991).

(Y) Control signal transmission

The model developed so far needs four input signals from the operator. Up-down for the climber, left-right for harvester arm in horizontal plane (concept L2), forward-backward for harvester arm in vertical plane (concept M2), close-open for the cutter shear blades (concept N3). A standard radio control (Z2) system with 4 channels costs about £150 and is the most convenient method of control. Using wires is the cheaper method and is preferred. The feasibility of using wire will be tested with the test rig then if it works both methods are valid and using each method is only the matter of simplicity, cheapness or convenience.

(Z) Braking

Three places for brake system was thought of at the early stage of the design, but at the stage which a DC electric motor and PLC (X4) were chosen as the driver and control system, the "on motor" brake ,Y1, was selected. It is suitable for the design because it is readily available with this type of motors.

(AA) Transmission

DC motors run usually up to 7000 rpm with tow torque which must converted to low speed and high torque for a vertical climbing system. Worm and gear transmission (AA1) is a very compact and light system compare to other systems for the same reduction ratio. Gears (AA2) occupy more space but have better efficiency than worms. They need a very accurate casing. Chain (AA3) are good when shafts are not close to each other and can tolerate more errors in the shaft positioning. Belts (AA4) have less efficiency than chains but are the cheapest concept and are suitable for high speeds. Belts are not suitable for the machine because of the slippage that may become a problem for the braking system and reduces the worker, tree and machine safety. Then best choice for the machine is (AA1) worm and gear transmission system, because it is light and many electric motors are coupled with this type of gear box. The second advantage is the braking action of the gear and worm system which does not let the motor turn easily when the machine is on the tree and the power is off. It helps the brakes to stop the machine more efficiently by reducing the brake load. A chain transmission was used at the final stage of the transmission as it is easy to change the speed and torque ranges easily by changing the sprockets. This is very convenient for the test rig but enclosed gearing would be preferred for the production machine.

2.5.4. Driving axles

The selection of the driving axles and the number of them is important for the correct working of the machine. The movement of a wheel on a flat tree surface and an uneven tree surface was analysed to determine the wheel behaviour and decide on the driving axles and wheel diameter. The uneven surface results from the presence of old leaf bases on the tree trunk. The wheel in both conditions is illustrated in Figure 2-6. The friction theory applies to the contact surface. To carry load W on a flat surface the wheel should be forced on to the tree by P where:

$$W = \mu P \tag{2.5}$$

Where μ is coefficient of friction between the tyre and the tree.

If the wheel radius is r the required torque for the static equilibrium will be $M = Wr$

Where the same wheel faces a step (Figure 2-6, b) it will lose some part of its load carrying ability. The new load capacity, W_s, decrease with the height of the step, s, and the coefficient of fiction between the tyre and the edge of the step, μ_s. Ageikin (1987) reported that minimum load capacity occurs when the wheel just loses its contact with the flat surface and is supported by the edge of the step (point a).

The following equations can be used to find the reduction in load capacity of the wheel and the maximum height of the step that a wheel can climb when other parameters are known:

$$\sum F_x = 0$$

$$P_s - T_s \sin\alpha - N_s \cos\alpha = 0 \tag{2.6}$$

Where T_s is step tangential reaction force to the tyre at point a; N_s is step normal reaction force to the tyre at point a; P_s is horizontal load on the wheel when it is leaving the flat surface.

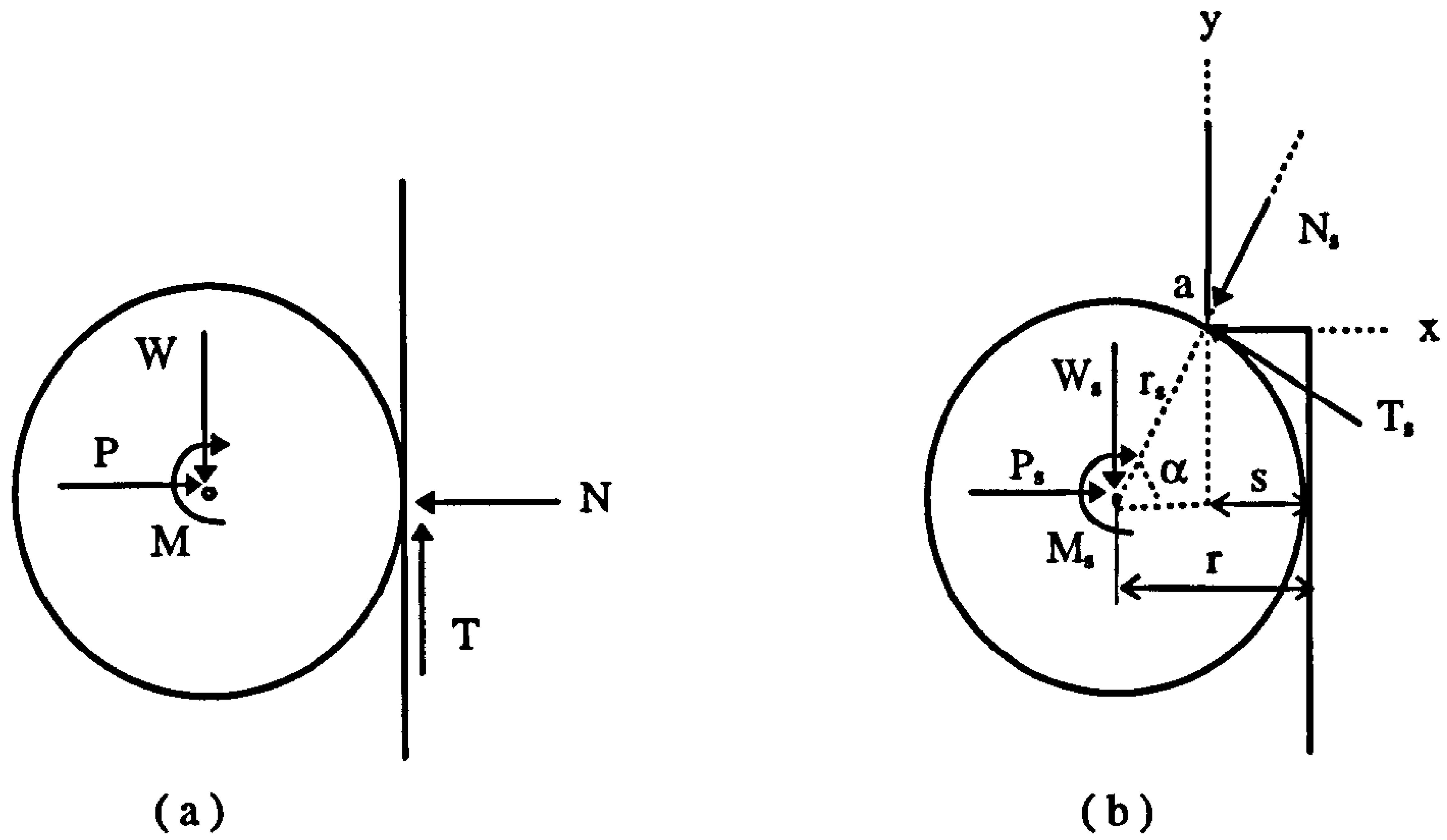


Figure 2-6 Wheel climbing a tree, (a): flat surface (b): step (worse case of a leaf base)

The friction controls the maximum value of the T_s so:

$$T_s = \mu_s N_s$$

Where μ_s is the coefficient of friction between the wheel and the step edge. It is equal to $1.2 \times \mu$ (Ageikin 1987). Substituting T in Equation 2.6 gives:

$$P_s \ -\mu_s N_s \sin\alpha - N_s \cos\alpha = 0$$

$$N_s = \frac{P_s}{\mu_s \ \sin\alpha + \cos\alpha} \tag{2.7}$$

Where N_s is normal reaction from the step edge.

$$\Sigma F_y = 0$$

$$-W_s - N_s \sin\alpha + \mu_s N_s \cos\alpha = 0$$

$$W_s = N_s (\mu_s \cos\alpha - \sin\alpha)$$

Substituting N_s from Equation 2.7 in above equation gives:

$$W_s = \left(\frac{\mu_s \cos\alpha - \sin\alpha}{\cos\alpha + \mu_s \sin\alpha}\right) P_s$$

Dividing numerator and denominator of the term in bracket by $\cos\alpha$ gives:

$$W_s = \left(\frac{\mu_s - \tan\alpha}{1 + \mu_s \tan\alpha}\right) P_s \tag{2.8}$$

The term in bracket was called coefficient of traction so:

$$W_s = k\, P_s \tag{2.9}$$

and

$$k = \left(\frac{\mu_s - \tan\alpha}{1 + \mu_s \tan\alpha}\right) \tag{2.10}$$

The relation between α, wheel radius and step height can be drawn from the dotted line triangle in Figure 2-6:

$$\cos\alpha = (r - s) / r_s \tag{2.11}$$

Where r is wheel radius, mm. s is height of the step, mm. r_s is wheel radius on the step edge in mm, which is smaller than r because of tyre deformation at the step edge.

Figure 2-6 and Equation (4.11) show that for a constant wheel radius, α increases when s increases. Equation (4.8) shows that for a constant value of μ_s and p_s the load capacity decreases when α increases. It means that if the wheel is carrying its maximum capacity on the flat surface it cannot continue carrying it when it arrives at a step. To compensate for the reduction of the load capacity of the machine when it faces a step, the machine needs some assistance from another driving axle, therefore, the climber requires a minimum number of two driving axles to keep its original load capacity. The second axle pushes the climbing wheels and help them to roll over the step. Equation 2.10 can be used to calculate the coefficient to determine the new load capacity. From the geometry of the model axles A and axle C were chosen to be the driving axles as they carry the highest load. The other great advantage of axle A as a driving axle is providing the ground mobility for the machine as it is the lowest axle. If the machine is designed to move on a smooth surface which is not a date tree, only one axle is required and that is C because it carries the highest load of the three axles therefore can provide the highest traction force (lifting force).

2.5.5. The machine layout

The objective of this section is to summarise the machine concept layout, which elements were selected from the morphological chart, and discussions of previous Sections. The sketch of the machine and the required elements are briefly illustrated in Figure 2-7 and Figure 2-8. It can be seen from the front view that the machine has a "two axle unit" with two wheels on each axle. The stabilisers (extra ground wheels, D), harvester arm, cutter and the fruit basket are also connected to this main unit. On the other side of the tree there is the "one axle unit" with two wheels on it which provides the forcing and gripping system and also a proportion of the traction force because it has a driving axle. Wheels of axle A and C are driving wheels and wheels of axle B and stabiliser wheels D are idle. Stabiliser wheels assist the machine for ground movement and are not used in climbing. They can be detached from the machine, if required, to increase the machine load capacity.

Looking from above, to harvest the tree the wheel spacing shown in Figure 2-8 should be adjusted for all axles to suit the tree diameter.

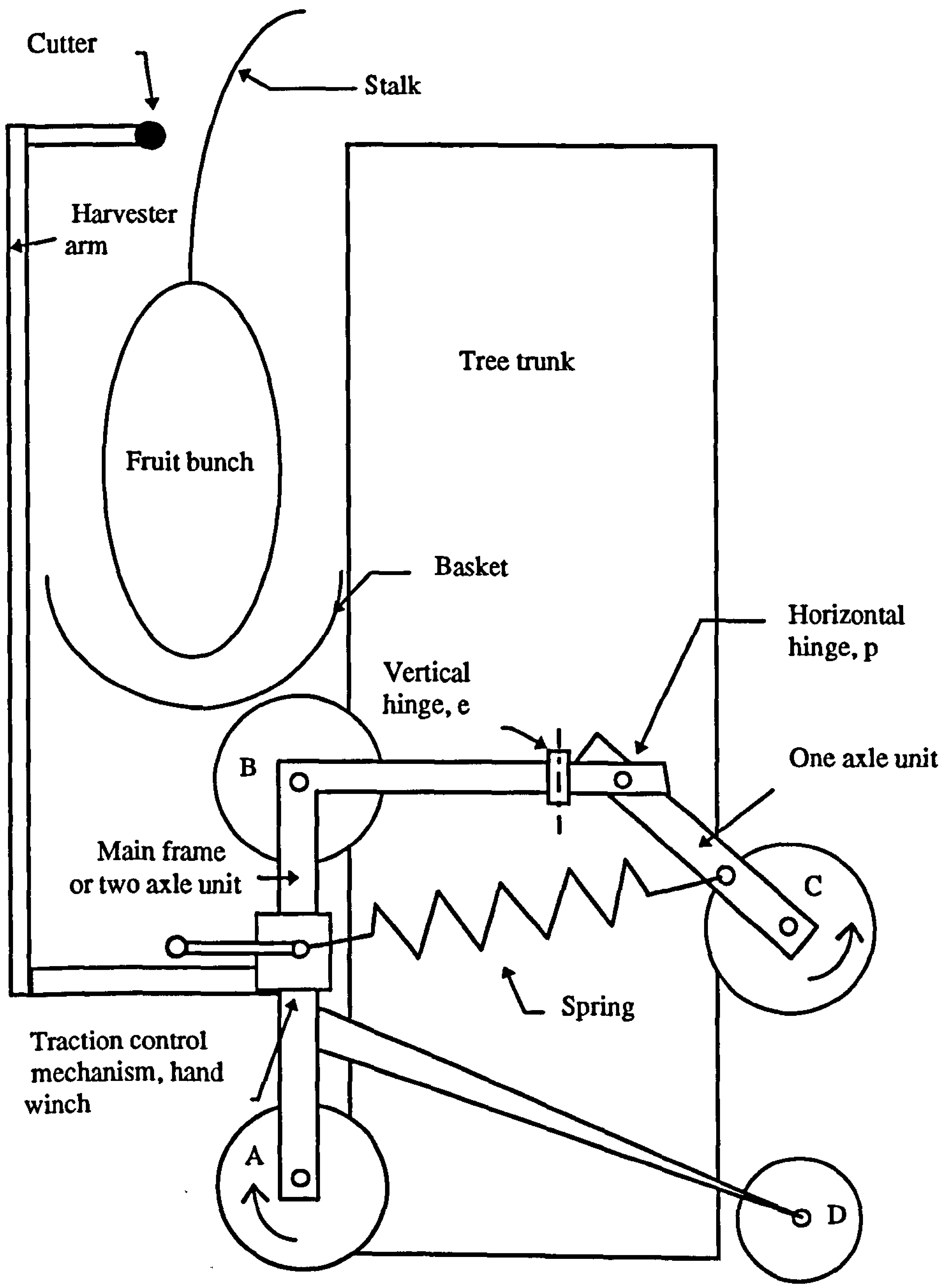


Figure 2-7 Front view of the machine including important elements

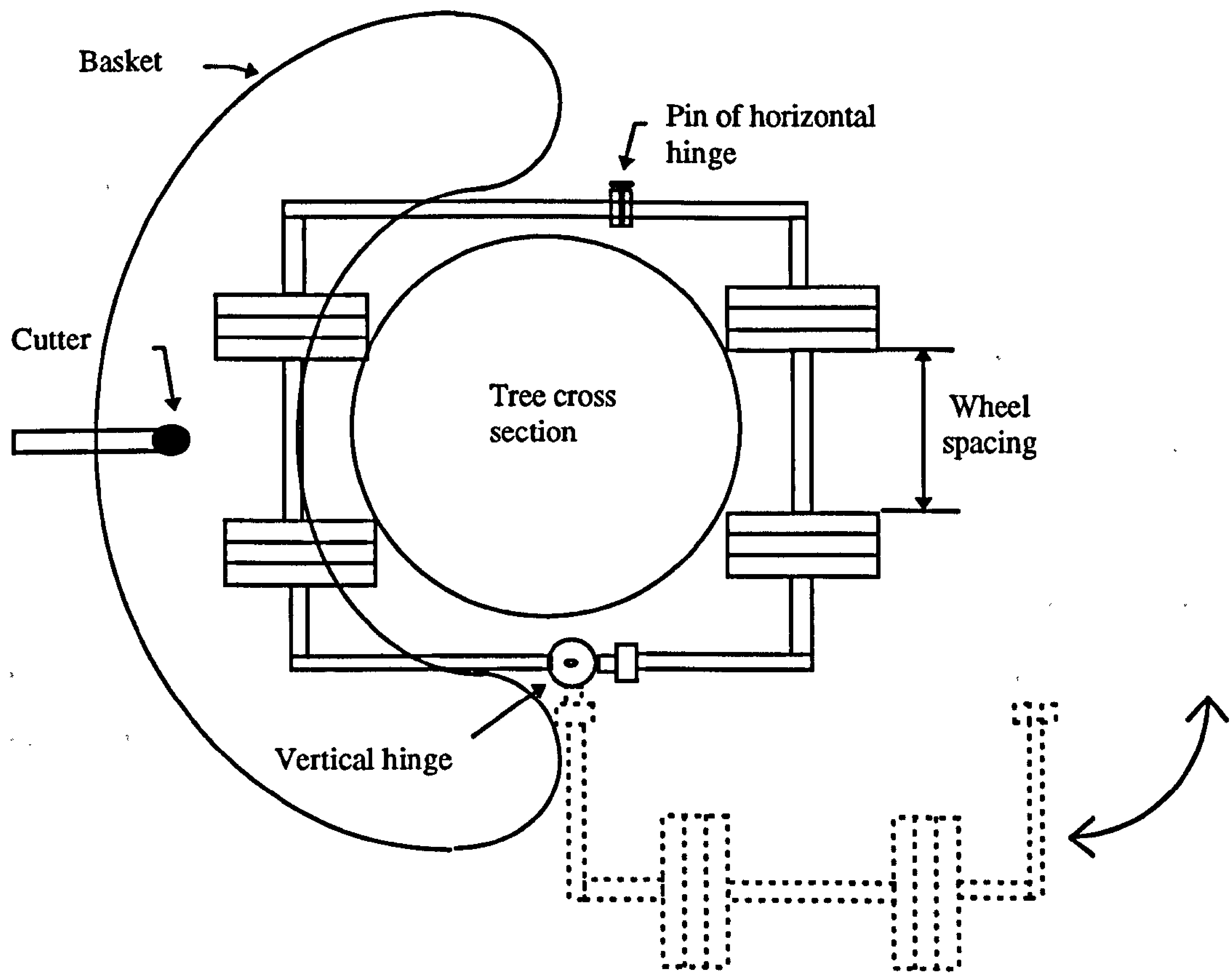


Figure 2-8 Top view of the machine

To attach the machine to the tree "one axle unit" can be rotated around its vertical hinge e which is illustrated in Figure 2-8. This action can be undertaken once the horizontal hinge (p) pin is pulled out; the machine is then moved towards the tree. When the machine surrounds the tree the "one axle unit" will be brought back to its original position and the horizontal hinge (p) pin will be fitted in its hinge.

The traction control mechanism shown in Figure 2-7 pulls the pair of springs and keeps them in tension to produce the gripping force P, to provide the required traction force for the machine driving wheels.

The machine climbs the tree vertically. When it reaches the tree crown the harvester arm can rotate on a circular path a total of 180° to search for a bunch. The cutter is then moved forward and backward to reach the bunch stalk. The cutter is then operated and the bunch falls into a very light basket which extended 180° under the arm. With this arrangement the machine can harvest half of the tree yield at one time and comes back to the ground for the dates to be unloaded. The machine will then be detached from the tree, repositioned and re-attached to the other side to complete harvesting of the remaining fruits.

2.5.6. Model force analysis and general parametric equations

The general parametric equations of the three axle date harvesting tree climbing machine were developed to predict the machine behaviour and dimensions.

Based on the force diagram of Figure 2-9 the static equilibrium equations were written with co-ordinate system located at point A, the contact point of wheels of the lowest axle with the tree, as follows:

$$\Sigma F_x = 0 \Rightarrow$$

$$C_x = A_x + B_x \tag{2.12}$$

$$\Sigma F_y = 0 \quad \Rightarrow$$

$$-W + A_y + C_y = 0$$

$$W = A_y + C_y \tag{2.13}$$

Taking a moment around point A gives:

$$\Sigma M_A = 0 \quad \Rightarrow$$

$$x_w W + y_B B_x - y_c C_x + x_c C_y = 0 \tag{2.14}$$

From the friction theory, the optimum condition to avoid the wheel slippage or extra forcing on the tree is when:

$$C_y \;= \mu\, C_x \tag{2.15}$$

$$A_y \;=\; \mu\, A_x \tag{2.16}$$

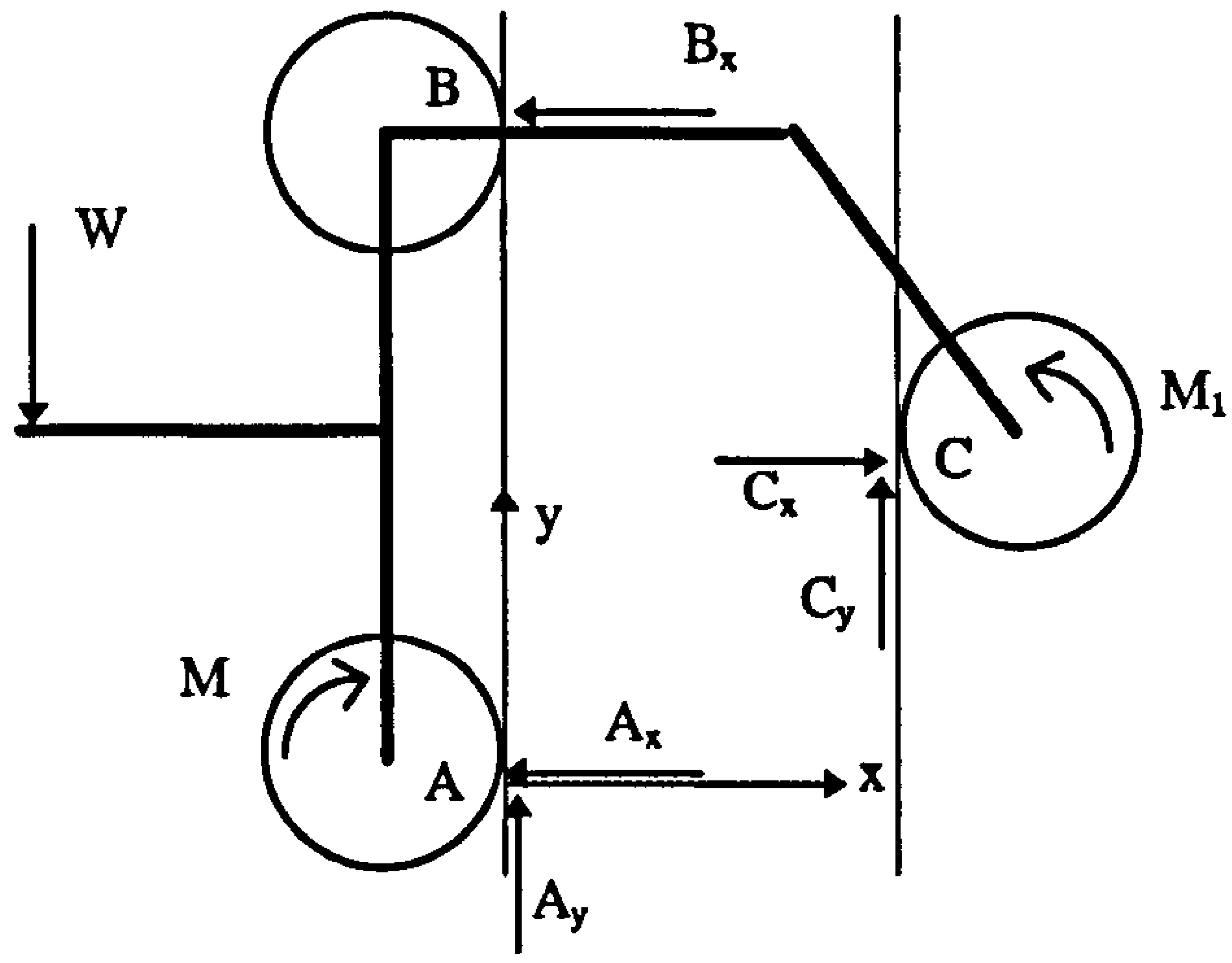


Figure 2-9 Force diagram of the model

Where C_x is horizontal reaction of the tree on the driving wheels C. A_x is horizontal reaction of the tree on the driving wheels A. B_x is horizontal reaction of the tree on the idle wheels B. W is the total weight of the machine including the dates; A_y is vertical reaction of the tree on the driving wheels A. C_y is vertical reaction of the tree on the driving wheels C. x_w is the length of the centre of gravity, horizontal distance from point A. y_B is the height of the contact point of wheels B, vertical distance from point A. yc is the height of the contact point of wheels C, vertical distance from point A. x_C is length of the contact point of wheels C, horizontal distance from point A. μ is coefficient of friction between tyre and machine.

Substituting Equation 2.15 and Equation 2.16 in Equation 2.13 gives:

$$W = \mu A_x + \mu C_x$$

Substituting A_x from Equation 2.12 and transforming the equation for C_x gives:

$$W = \mu (C_x - B_x) + \mu C_x$$

Writing the equation for C_x gives:

$$C_x = 0.5(W/\mu + B_x) \qquad (2.17)$$

Substituting C_x in Equation (2.15) gives:

$$Cy = \mu C_x$$

$$C_y = 0.5(W + \mu B_x) \tag{2.18}$$

Substituting equation (2.17) and equation (2.18) in equation (2.14) and transforming the equation for B_x gives:

$$x_w W + y_B Bx - ycCx + x_c C_y = 0$$

$$x_w W + y_B B_x - y_c(0.5(W/\mu + B_x)) + x_c(0.5(W + \mu B_x)) = 0$$

$$2x_w W + 2y_B B_x - y_c(W/\mu + B_x) + x_c(W + \mu B_x) = 0$$

$$B_x(2y_B - y_C + \mu x_C) = W(y_C/\mu - x_C - 2x_W)$$

For x_c = tree diameter (d) :

$$B_x = \left(\frac{\frac{y_C}{\mu} - d - 2x_W}{2y_B - y_C + \mu d} \right) W \tag{2.19}$$

Equation (2.19) is an important equation for the selection of the machine dimensions and determination of the reaction force B_x exerted from the tree to the machine. It shows that B_x is W multiplied by a coefficient.

For the proper work of machine, B_x should be positive because it is the machine guiding force that prevents the machine to moving to one side of the tree. B_x is positive when the coefficient is positive. The coefficient is a combination of machine dimensions, tree and wheel diameter and the coefficient of friction between the tyres and the tree. This combination should be selected such that the coefficient is positive, a negative B_x means that the machine is unstable.

2.5.7. Wheel spacing

A self guiding system was designed for the machine to prevent the lateral movements in directions shown in Figure 2-10 (a). If this movements are not controlled the machine hits to one side of the tree, may cause damage and can not continue climbing. This idea works when there are two wheels on the axle and the wheel spacing is more than a particular value, which can be determined from the friction theory. Figure 2-10 (b) shows the principles of the system. If wheel spacing is wide enough then wheels slide to the correct position shown in the Figure 2-10 (b) independent of the value of F. To satisfy this condition:

$$T \geq \mu N \qquad (2.20)$$

Writing static equilibrium equations gives:

$$T = F.\sin\beta$$

$$N = F.\cos\beta$$

substituting T and N in Equation (2.20) gives:

$$F.\sin\beta \geq \mu F.\cos\beta \quad \Rightarrow \quad \tan\beta \geq \mu$$

or

$$\beta \geq \tan^{-1}\mu \qquad (2.21)$$

From the small triangle inside the tree cross section:

$$\sin\beta = \frac{\text{wheel spacing}}{\text{tree diametre}}$$

or

$$\text{Wheel spacing} = \sin\beta \times \text{tree diameter} \qquad (2.22)$$

Equation 2.21 in Equation 2.22 gives:

$$\text{Wheel spacing} \geq \sin\tan^{-1}\mu \times \text{tree diameter} \qquad (2.23)$$

Equation (2.23) predicts the minimum wheel spacing for the self guidance system.

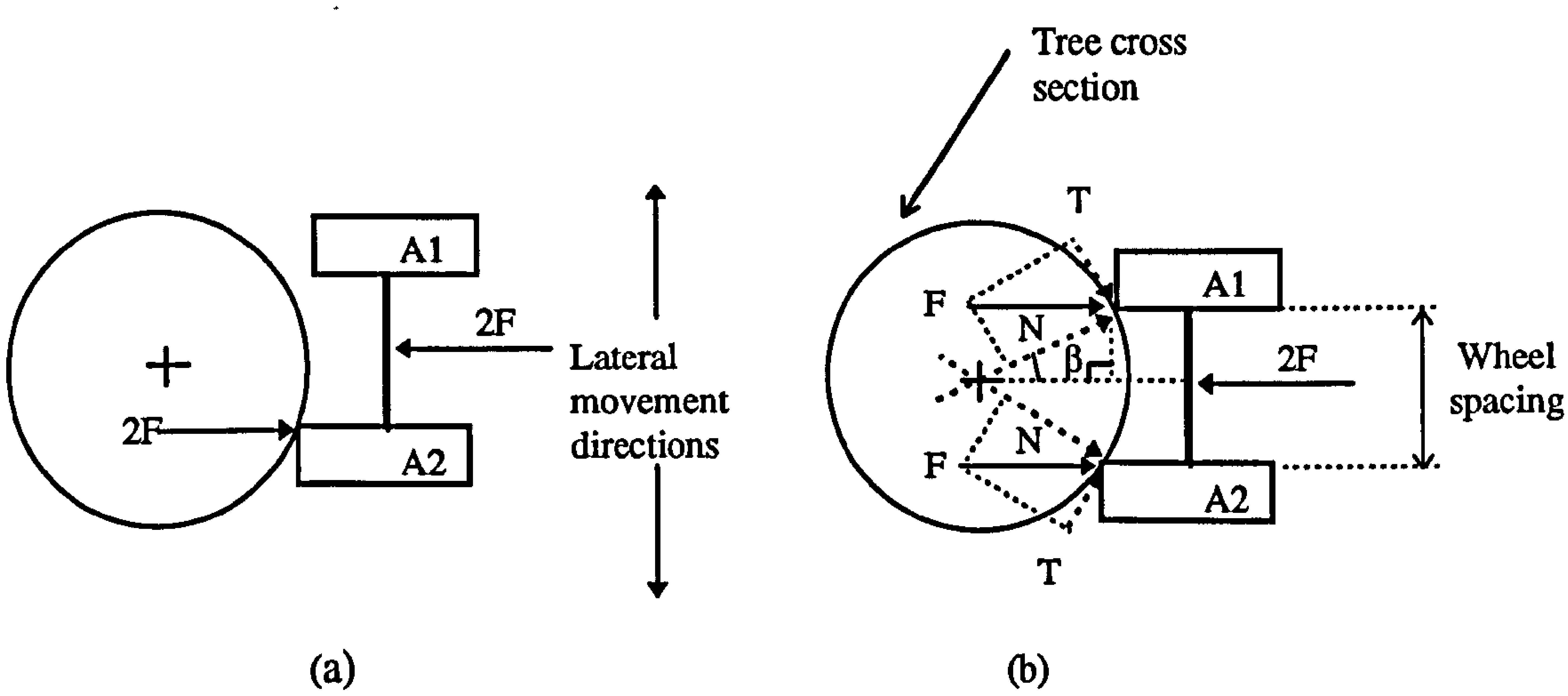


Figure 2-10 Machine lateral control: (a) problem of no control (b) analysis of control

2.5.8. Machine wheel radius

An optimum wheel radius was required for the machine. Wheel with small radius can not climb properly and big diameter wheels are heavy and expensive. The optimum wheel radius was found by considering:

1. the maximum leaf base pitch
2. the tree to axle clearance
3. the maximum leaf base height

The largest wheel radius found from the three consideration was selected as the wheel radius.

2.5.8.1. Wheel radius in relation to the leaf base pitch

Different combinations of leaf base dimensions were tested to determine the worst climbing condition for a wheel. The worst climbing condition or the maximum horizontal wheel displacement (in Figure 2-11) was when a maximum pitch and height leaf base followed after a minimum height leaf base. This condition with leaf base shape and a small and big wheel are illustrated in Figure 2-11.

It is seen that a wheel can anchor to the edge of the next leaf base, point a, better if the radius is greater that the pitch of that leaf base (length db). If it is smaller than db the wheel cannot anchor to the leaf base edge, therefore wheel radius must be greater than the maximum leaf base pitch, db.

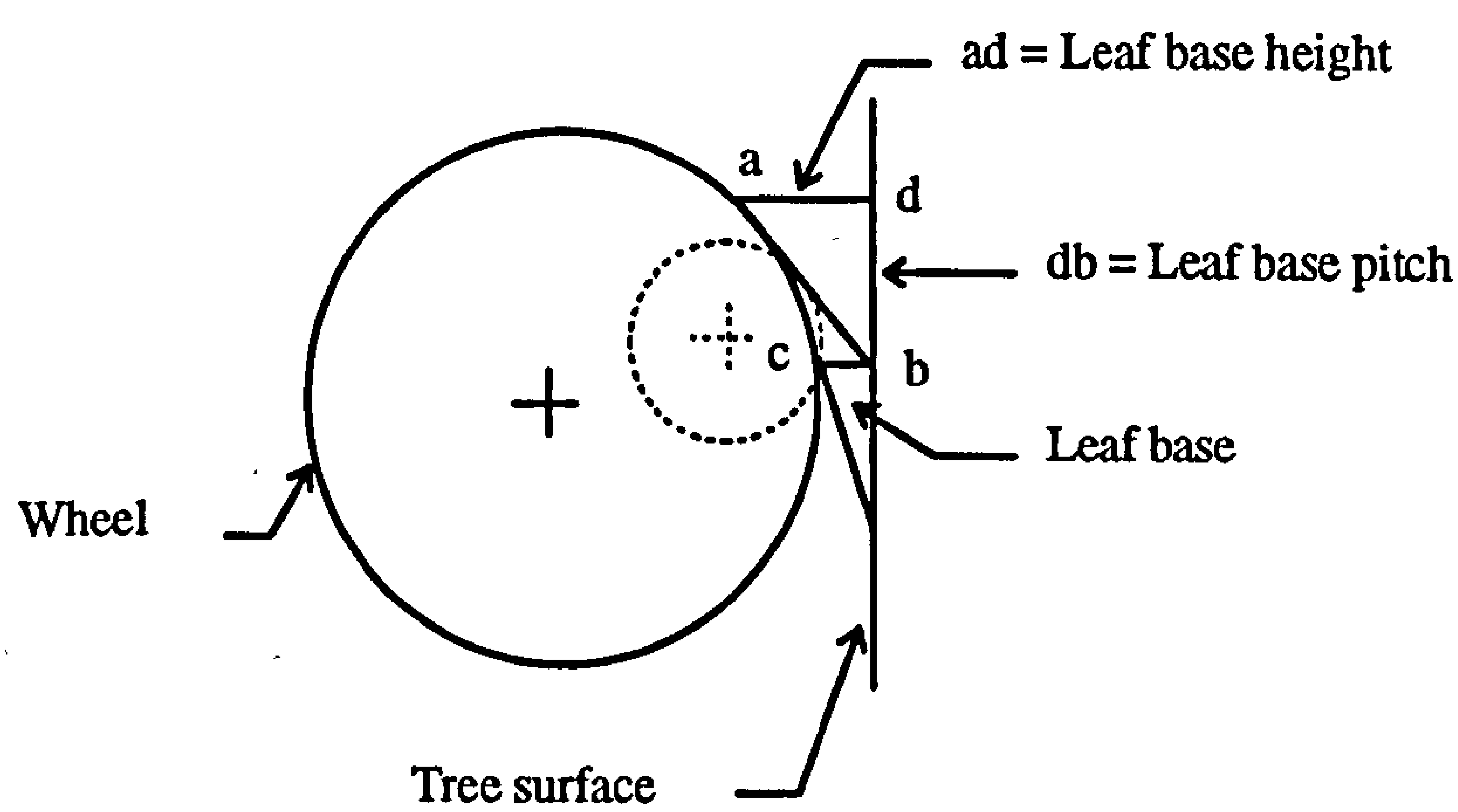


Figure 2-11 Leaf bases shape and ability of a big wheel to garb to the edge of the next leaf base (point a)

2.5.8.2. Wheel radius in relation to the tree, axle clearance

There are two wheels on each axle of the machine and the spacing is adjustable to allow working on different tree diameters. The wheel radius, r, in Figure 2-12 should be such that when the spacing is set at the maximum, the wheel axle does not touch the tree. to satisfy this condition the following analysis was used:

$$r = od - oa \qquad (2.23)$$

Where r is wheel radius in mm.

Figure also shows that:

(2.24)

od = tree radius, mm + clearance between the tree and the axle, mm

To ensure that the axle shaft does not touch the tree under normal circumstances a 70 mm clearance between the tree and the axle was allowed.

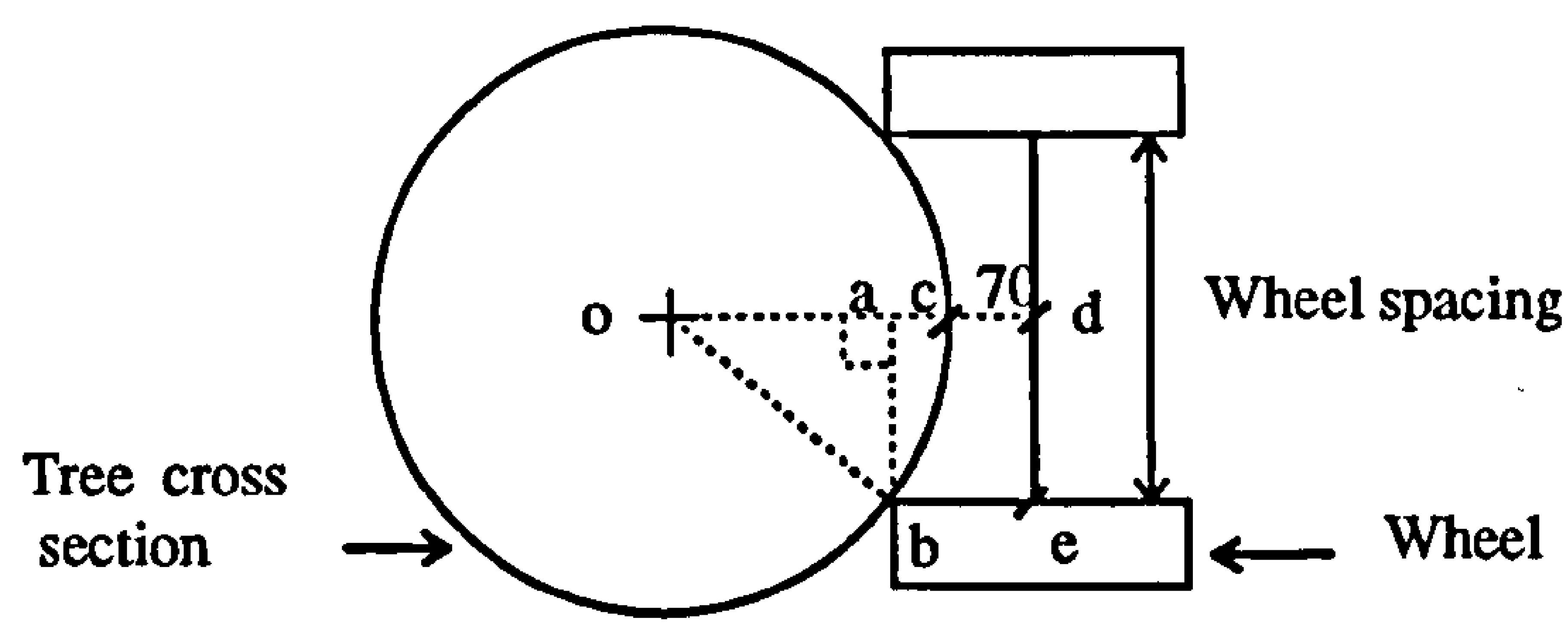


Figure 2-12 Wheel radius calculation based on the axle clearance

From triangle oab is seen that:

$$oa = \sqrt{ob^2 - ab^2} \qquad (2.25)$$

Where ob is tree radius, mm. ab is half of wheel spacing calculated from Equation 2.22 in mm.

Substituting these parameters in Equation 2.25 and then substitute Equation 2.25 and Equation 2.24 in Equation 2.23 gives:

(2.26)

$$r = (\text{tree radius, mm} + 70) - \sqrt{(\text{tree radius, mm})^2 - (0.5 \times \text{wheel spacing, mm})^2}$$

Equation 2.26 calculates the wheel radius considering the axle clearance. In this equation wheel radius, r, increases with tree radius and wheel spacing, therefore, for the wheel to be able to work on a range of date trees, tree radius and wheel spacing must be related to values of the maximum diameter date tree.

2.5.8.3. Wheel radius in relation to the leaf base height

The analysis of the wheels on the flat and uneven surface introduced in section 2.5.4 will be continued here to find another criterion for selection of optimum wheel diameter.

When the machine is lifting a load W on a flat tree surface and reaches a leaf base the load capacity W reduces to W_s. Dividing Equation 2.9 by Equation 2.5 gives:

$$\frac{W_s}{W}=\frac{kP_s}{\mu P} \qquad (2.27)$$

For the tree climber wheels P_s is almost equal to P because it is the force by which the climber wheel is forced on the tree and stays nearly constant when wheel position changes a maximum of 41 mm to climb a leaf base, therefore:

$$\frac{W_s}{W}=\frac{k}{\mu} \qquad (2.28)$$

Substituting k from Equation 2.10 and μ_s = 1.2μ (Ageikin 1987) in Equation 2.28 gives:

$$\frac{W_s}{W}=\frac{1.2\mu-\tan\alpha}{\mu+1.2\mu^2\tan\alpha} \qquad (2.29)$$

Rewriting Equation 2.11 for r gives:

$$r = r_s.\cos\alpha + s \qquad (2.30)$$

Selecting a ratio for $\frac{W_s}{W}$ and knowing μ for a design situation, Equation 2.29 can be used to determine α. Then substituting α in Equation 2.30 and knowing s, height of step (leaf base) and tyre deflection (r-r_s), the wheel radius, r, can be calculated and if wheel radius is known the required tyre deflection can be calculated.

In the real condition s in Equation 2.30 is the maximum horizontal displacement or absolute step (leaf base) height that the wheel radius r is designed to climb (refer to

Figure 2-11). For the tree climber driving wheels height of step, s, is equal to the difference between the maximum leaf base height (length ad in Figure 2-11) and the minimum leaf base height (length cb in Figure 2-11). To design wheel diameter these values were measured in the field study in Iran.

Selecting a value for $\frac{W_s}{W}$ in Equation 2.25 is a compromise between required extra power and wheel radius. Smaller ratios require more extra force or push from the second driving axle and according to Equations 2.25 and 2.26 a bigger wheel is also required. For the current study the ratio was selected 0.5 which suggests a reasonable wheel diameter. With the assumption that at each moment one driving axle is on the worst push demanding condition (which it leaving its contact with smooth surface and suspending from the step edge Figure 2-6 ,b) and the other one is on the smooth surface ratio 0.5 suggest that each driving axle should be able to provide an extra traction force equal to half of the capacity of the other axle for critical condition of climbing leaf bases. There are two driving axles and two motors, therefore, the motor design power should be selected for lifting a weight, 1.5 times bigger than the power required to lift the machine actual weight on a smooth tree surface.

The wheel radius must be calculated by the first two explained methods (maximum leaf base height, Section 2.5.8.1 and axle clearance, Section 2.5.8.2). The larger one must be chosen and substituted in Equation 2.30 of the third method to calculate the tyre deflection. If the tyre deflection is less than 50% of the tyre section height (Ageikin 1987) the wheel radius is accepted, otherwise, a bigger wheel radius must be selected using Equation 2.30 to achieve the proper value of tyre deformation.

CHAPTER 3

INTERACTION BETWEEN MACHINE AND TREE

3. Interaction Between Machine and Tree

The study of interaction between machine and tree was conducted in Iran in order to establish values for PDS and design equation parameters developed in Chapter 1 and Chapter 2. The study was divided into four stages:

1. Tree failure stresses: to check that the stresses caused by the machine on the tree do not damage the tree, in order to satisfy the PDS statement of tree safety. The method was to define a safety factor, S, which is equal to the tree failure stress divided by the working stress applied to the tree by the machine.

2. Hand harvesting speeds and tree yields: to evaluate the machine working speeds and select the machine capacity explained in PDS.

3. Tree and fruit sizes measurements: to design the machine sizes directly or using design equations to satisfy PDS elements such as ability to climb the tree uneven surface and work on different tree diameters.

4. Measurement of coefficient of friction between tyre and the tree surface: to select the wheel spacing, tyre size and machine lifting capacity.

The study was carried out in Bam and Shahdad, two Cities in Kerman province during summer 1995. The failure stress tests were conducted on a newly cut tree trunk. This was cut into the small samples which were tested by equipment in the Material Physical Testing Laboratory of Kerman University. Two special tools which had been designed and constructed in Silsoe were used for tests on the cut tree and on the leaf bases of living trees according to the methods described in Section 3.1.4 and Section 3.1.5.

3.1. Tree failure stresses

To establish the method of tree stress measurements the wood stress systems were studied. Explained by Silvester (1967), there are three kinds of direct stresses to which timber can be subjected, tensile, compressive and shearing. Timber is not isotopic and has three structural axes and consequently has three different sets of values for mechanical properties in the three directions. The three structural axes of wood are longitudinal, radial and tangential (Figure 3-1). When considering the structure of wood in relation to its structural axes it is easy to appreciate the importance of axes orientation in respect of mechanical properties.

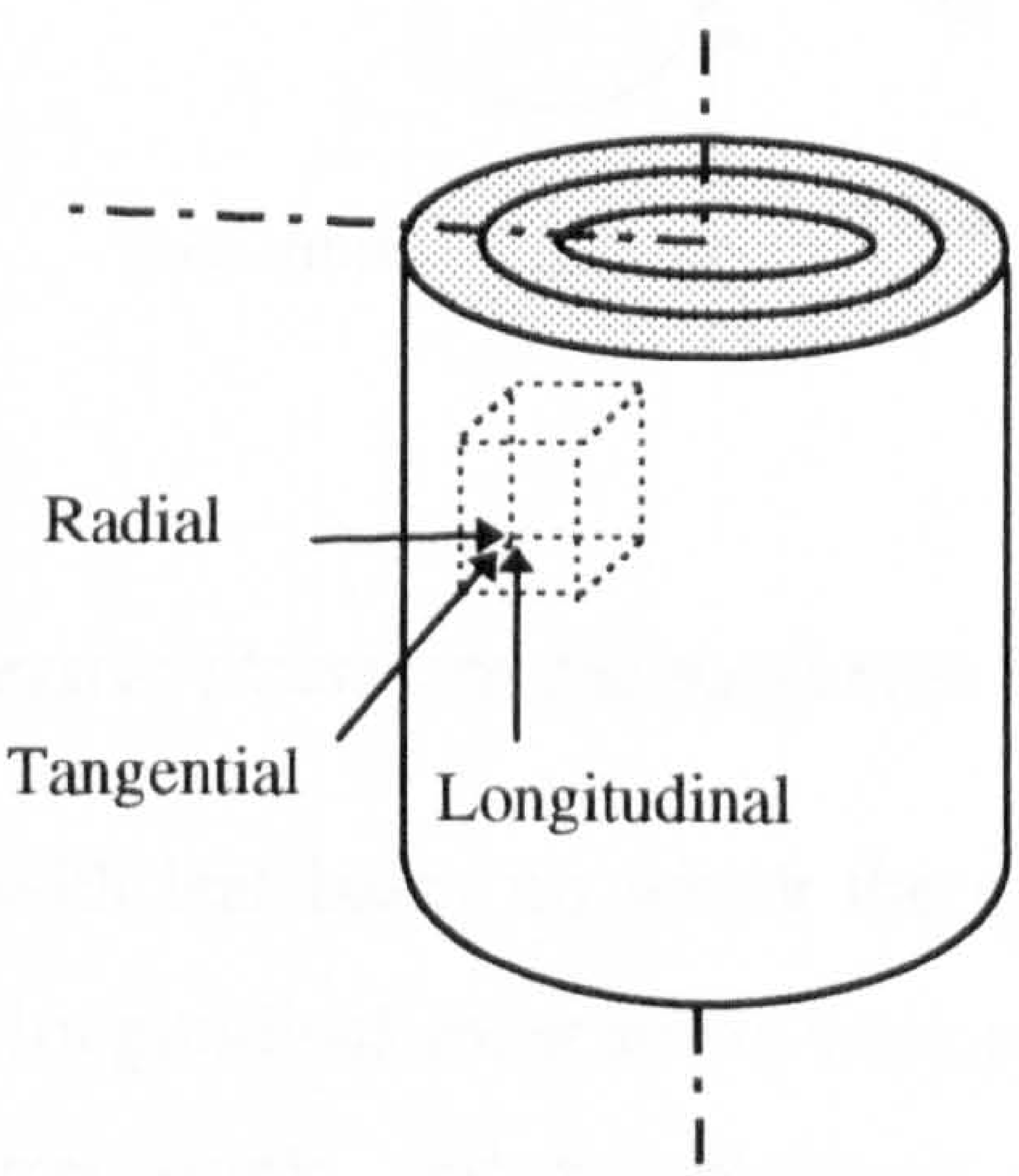


Figure 3-1 Main stress directions in the tree trunk

The difference in the strength properties of wood on the radial and tangential axes, however, is not considered to be of significant importance. It is usual, therefore, in structural design to consider the strength of timber only parallel with the grain, i.e. loaded in the direction of the longitudinal axis, or perpendicular to the grain when loaded on the radial or tangential axes.

To find which stresses should be measured, the behaviour of a driving wheel and the complete machine on the tree were analysed.

The wheel exerts a radial stress, σ_r, and the maximum longitudinal shear stress, $\tau_l = \mu\sigma_r$ to the tree as is seen in Figure 3-2 where μ is the coefficient of friction between tyre and tree. The contact surface of the pneumatic tyre on the tree surface changes with load such that σ_r stays constant and equal to the tyre inflation pressure, therefore:

$$\sigma_r = \text{tyre pressure} \tag{3.1}$$

$$\tau_l = \mu \times \text{tyre pressure} \tag{3.2}$$

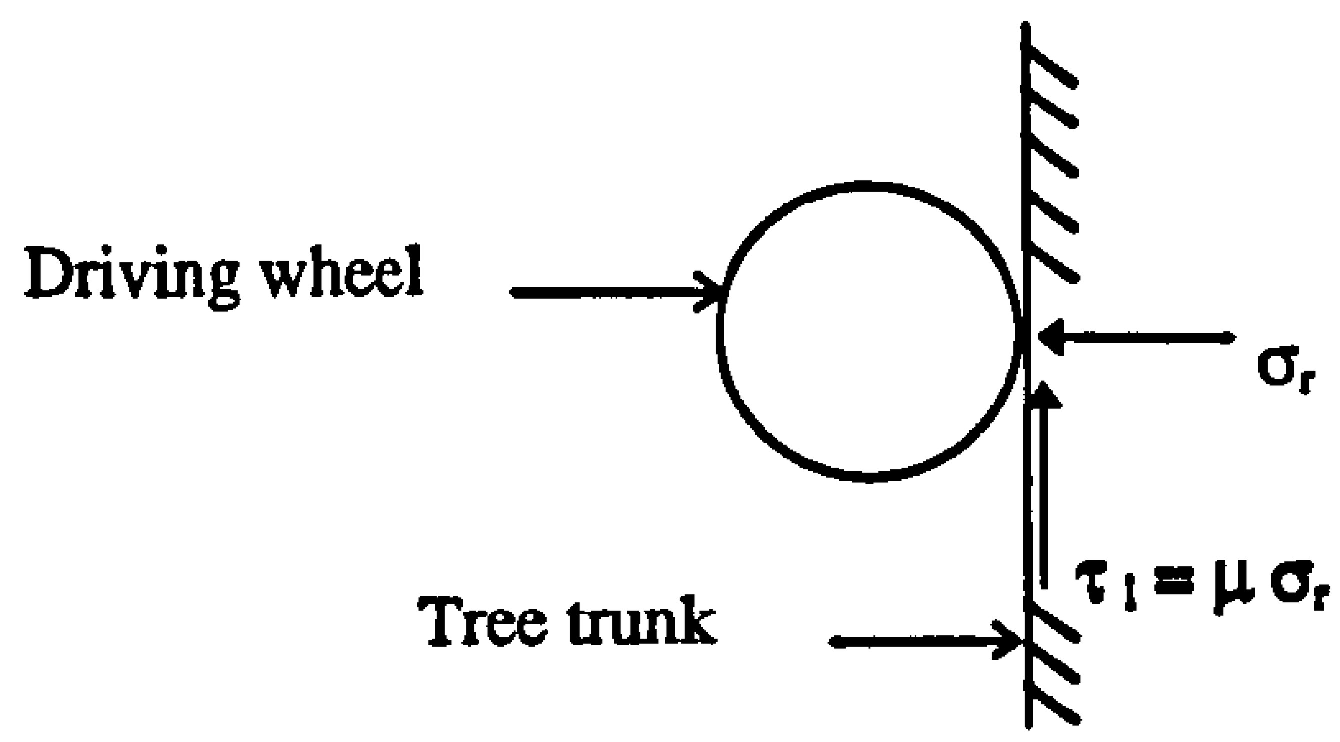


Figure 3-2 The radial compressive stress and the maximum longitudinal shear stress

The tree surface is covered with leaf bases on which the tyre works, therefore, the radial compressive stress and longitudinal shear stress were measured for both the tree surface (leaf bases) and the tree trunk and the minimum values were used for the design.

The complete machine applies its total weight as an eccentric load to the tree. Thereby producing varying stress levels on different parts of the tree. The maximum value of the stress was calculated from the principles of columns stress analysis (Beer & Johnson 1981), According to this analysis the eccentric load of machine weight, W, on the tree which is considered as a column can be replaced with a central load of the same value and a moment equal to the product of the load and the eccentricity as is shown in Figure 3-3. Then the maximum tensile and compressive longitudinal stresses, σ_t and σ_c, on the tree can be calculated from the following equations:

$$\sigma_t = \frac{W}{A} - \frac{My}{I} \tag{3.3}$$

$$\sigma_c = \frac{W}{A} + \frac{My}{I} \tag{3.4}$$

Where A is the tree cross section area in mm^2 and is equal to πr^2, where r is tree radius in mm. M is equal to Wx, maximum moment, in N mm. y is farthest element distance from the neutral axes in the tree cross section which is equal to the tree radius in mm. I is moment of inertia equal to $\frac{\pi r^4}{4}$ for a circular (tree) cross section.

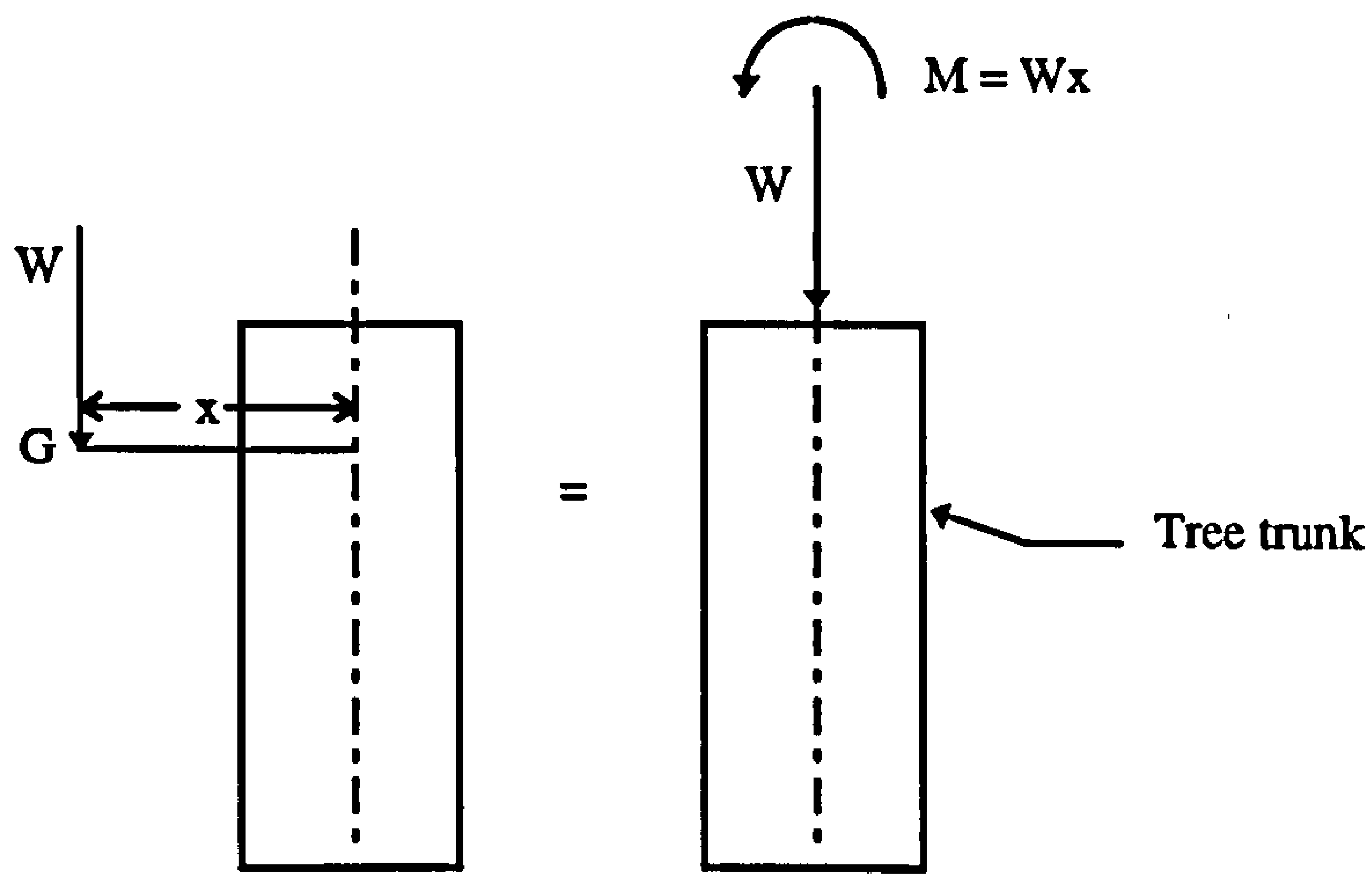


Figure 3-3 Machine load transformation

Equations 3.3 and 3.4 were used to calculate the maximum stress for the critical condition, that is when the machine moves on a tree with the least diameter.

Date palm trunk samples were prepared and tested using above considerations and British Standard 373 (1957), "Method of testing small clear specimens of timber", was used to select the sample sizes and test methods. From the above information the tree trunk samples were tested for:

1. Tensile stress in longitudinal direction
2. Compressive stress in longitudinal and radial direction
3. Shear stress in longitudinal direction

The leaf bases were tested for:

1. Compressive stress in radial direction
2. Shear stress in longitudinal direction.

Nine samples were selected at random from different locations within the trunk for each test.

3.1.1. Tree trunk tensile stress in longitudinal direction

The tree trunk and bark are composed mainly of bundled fibres which transfer water and nutrients to the leaves and fruits. An individual fibre has a diameter of approximately 0.2 mm and can carry considerable load. To measure the trunk tensile stress one fibre was separated from each $20 \times 20 \times 300$ mm sample and loaded by 1 N increments until separation. Figure 3-4 shows the test method and the results are presented in Table 3-1.

Table 3-1 Tree trunk tensile stress in longitudinal direction

No.	1	2	3	4	5	6	7	8	9	Average	Standard deviation
Force, N	72	77	62	57	59	72	81	65	69	68	8
Fibres per $mm^2 \times 10^2$	92	83	95	90	89	92	85	80	87	88	5

The tensile stress from Table 3-1 is equal to the average load that one fibre can carry until separation (68 N) multiplied by the average number of fibres per unit of area (0.88) which is 60 MPa.

3.1.2. Tree trunk compressive stress in longitudinal direction

The tree trunk test samples of 50 × 50 × 50 mm were prepared and loaded in longitudinal direction with a hydraulic press until sample starts to deform and release water. Figure 3-5 shows a sample and the experiment. The results are shown in Table 3-1 with an average stress of 5.34 MPa.

Table 3-1 Tree trunk longitudinal compressive stress

No.	Load, N	Dimensions, mm	Stress, MPa
1	13000	51 × 51	5.00
2	13300	50.5 × 50	5.27
3	13400	50 × 50	5.36
4	13300	49.5 × 50.5	5.32
5	13500	49 × 50	5.51
6	13200	51 × 50.5	5.13
7	13800	49 × 49	5.75
8	13500	50.5 × 49.5	5.40
9	13400	50 × 50.5	5.31
Average			5.34
Standard deviation			0.21

3.1.3. Tree trunk compressive stress in radial direction

The tree trunk compressive test was carried out by applying a force in the radial direction, the samples failed due to shear force which developed in the 45° plane relative to the compressive force direction (Figure 3-5). The results show that the average radial compression stress is 2.96 MPa (Table 3-2).

Table 3-2 Tree trunk compressive stress in radial direction

No.	Force, N	Dimensions, mm	Stress, MPa
1	7800	51×49	3.12
2	6800	48.5×50	2.80
3	7200	50.5×49	2.90
4	7300	50×50	2.92
5	7400	51.5×49	2.93
6	7300	49.5×50	2.95
7	7700	50.5×50	3.05
8	7500	51×51	2.88
9	7400	49×49.5	3.05
Average			2.96
Standard deviation			0.10

3.1.4. Tree trunk shear stress in longitudinal direction

The hydraulic and mechanical press could not measure the shear stress directly. A shear tool was designed in order to convert the compressive force of the press to shear force (Figure 3-7). The principle involves a moving jaw with an opposing stationary jaw, each jaw holds half of the sample. Loading the moving jaw creates a shear force equal to the compressive load on the sample in a vertical plane. Shear stress measurements were conducted in Iran using the tool. Longitudinal samples of $20 \times 20 \times 20$ mm were cut from the trunk fixed in the shear tool which was loaded gradually up to the braking point where complete shear occurs in the sample. The results are shown in Table 3-3 with a mean value of 1.1 MPa.

Table 3-3 Tree trunk shear stress in longitudinal direction

No.	Force, N	Dimensions, mm	shear stress, MPa
1	425	20 × 19	1.12
2	390	21 × 19	1.08
3	410	19 × 19.5	1.11
4	450	20 × 20.5	1.10
5	405	21 × 19	1.02
6	455	21 × 19.5	1.11
7	430	20 × 18.5	1.16
8	415	21 × 19.5	1.01
9	470	20 × 20	1.18
Average			1.10
Standard deviation			0.06

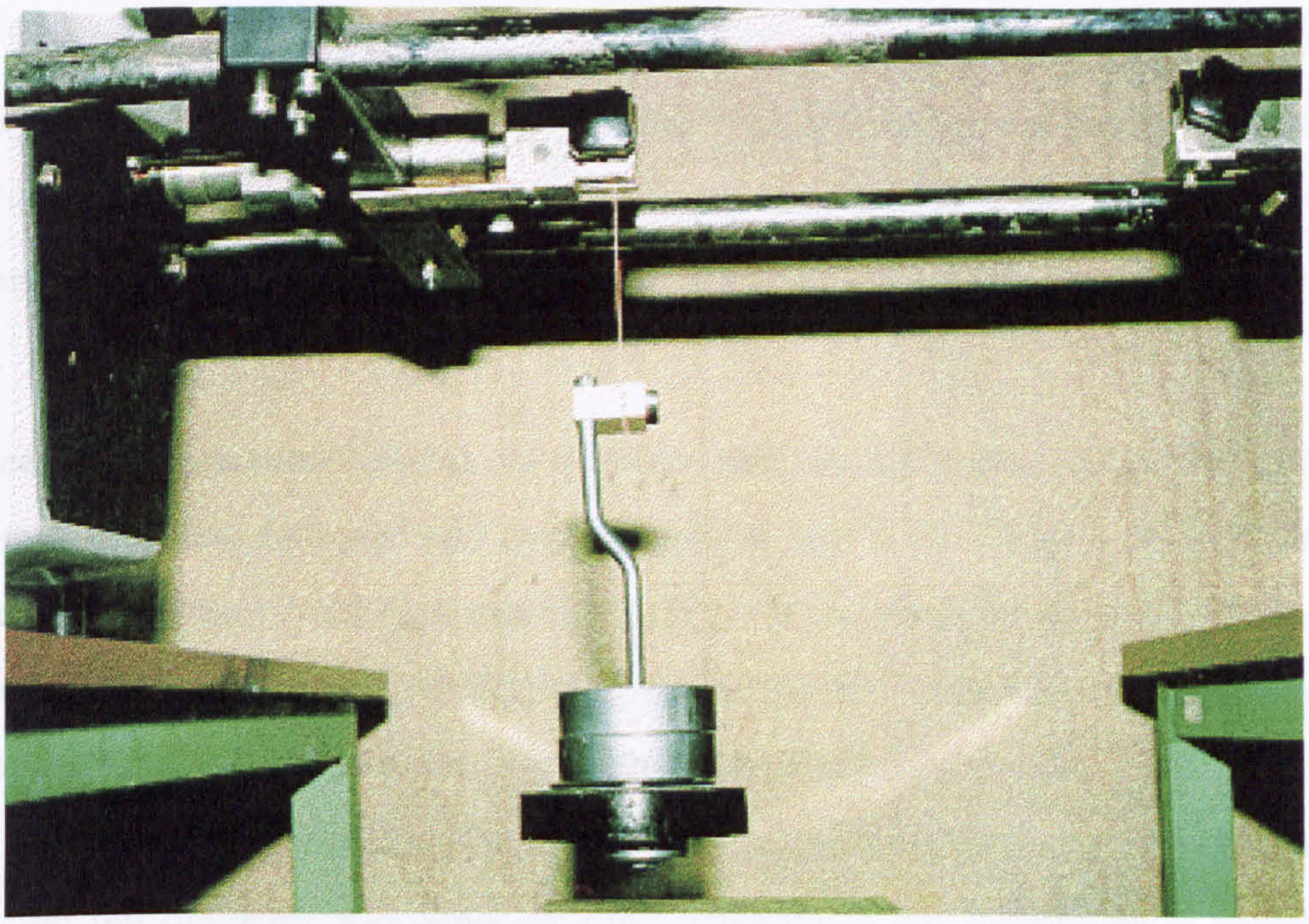


Figure 3-4 Tensile stress test on a fibre

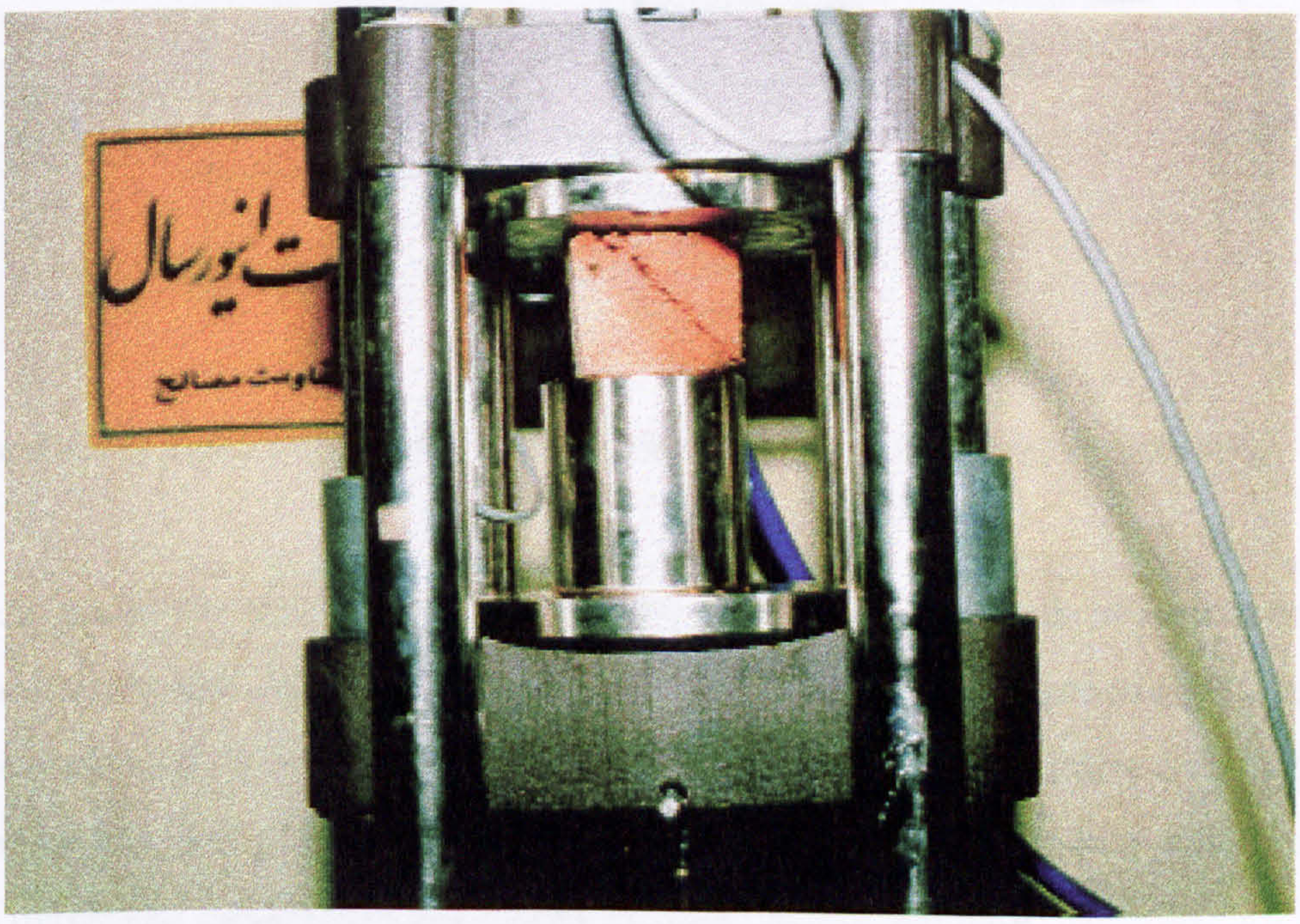


Figure 3-5 Compressive stress test on the trunk sample

3.1.5. Leaf base compressive stress in radial direction

Climber wheels move on the tree surface which is covered by leaf bases. The load on the wheel and the traction force developed under each driving wheel will exert a radial compression stress and also a longitudinal shear stress to the tree surface or leaf bases.

To measure the maximum radial compression that the leaf bases on the live tree can tolerate a tool was designed and constructed for use in the field. As is shown in (Figure 3-6) the tool comprises mainly of a lever in the shape of an angled beam. At one end (A) the angled beam is hinged to a plate which is attached to the tree and at the opposite end (C) weights are suspended on the tool. The weight action pushes a 4 mm diameter pin at point B to the leaf base in the radial direction. The criterion to select the pin diameter was reaching the failure stress by suspending a reasonable weight from point C.

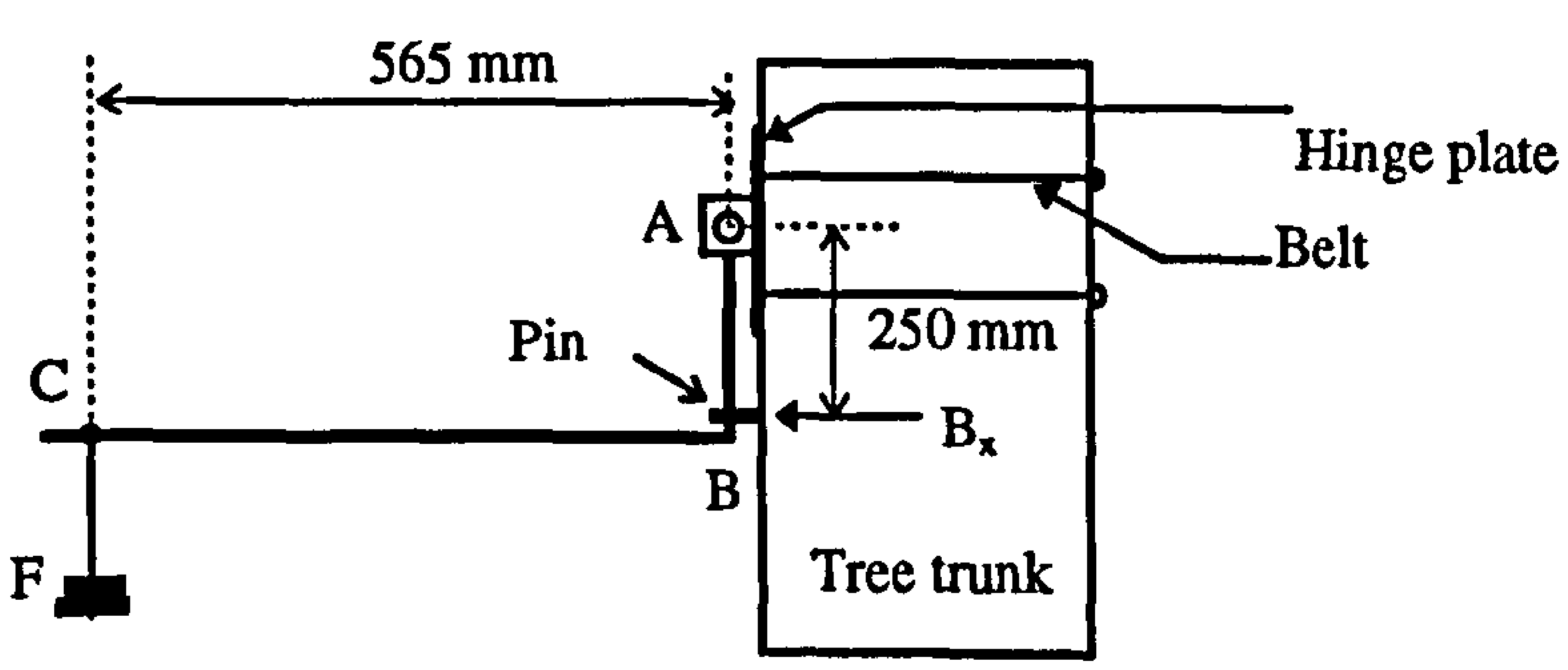


Figure 3-6 Mechanics of the tool constructed to measure leaf base compressive stress

A clock gauge connected to the tool is used to measure the depth of penetration of the pin into the leaf bases. It can be also used to draw the force, penetration diagram for the leaf bases, it was not used for the tests because the pin clearly and suddenly penetrates leaf base at a specific load. The stress at the point of failure was recorded as the maximum radial stress that can be tolerated by the leaf base.

Connecting the tool to the tree requires some skill. The tool is held onto the tree by two belts which are connected to the hinge plate and positioned around the tree as can be seen in Figure 3-8. After tying the belts around the tree and fastening them four adjusting bolts are used to align the plate in a vertical plane. The pin length should then be adjusted to ensure the tool lower beam is horizontal. Following these adjustments, the tool is ready to be loaded.

To convert the loading at point C to the radial compressive stress at point B the following equation was used:

$$\sigma_r = (B_x + \text{tool weight reaction force}) / (\text{pin cross section}) \qquad (3.5)$$

To find B_x a moment was taken about point A in Figure 3-6:

$$\sum M_A = 0$$

or

$$565\, F - 250\, B_x = 0$$

or

$$B_x = 2.26\, F \qquad (3.6)$$

Where F is force applied at point C.

Using a spring scale showed that the tool weight exerts a horizontal force equal to 7.5 N at point B. Substituting this value and B_x from Equation 3.6 in Equation 3.5, for a pin diameter of 4 mm:

$$\sigma_r = 0.59 + 0.18\, F \qquad (3.7)$$

Equation 3.7 was used to convert the loading, F, at point C to the compressive stress in radial direction, σ_r developed on the leaf base at point B.

The results of the loading, F, and conversion to compressive stress are shown in Table 3-4 with a mean value of 6.38 MPa.

Table 3-4 Leaf base compressive stress in radial direction

No.	F, N	Stress, MPa
1	27	5.45
2	40	7.79
3	32	6.35
4	31	6.17
5	29	5.81
6	39	7.61
7	37	7.25
8	28	5.63
9	30	5.99
Average		6.38
Standard deviation		0.96

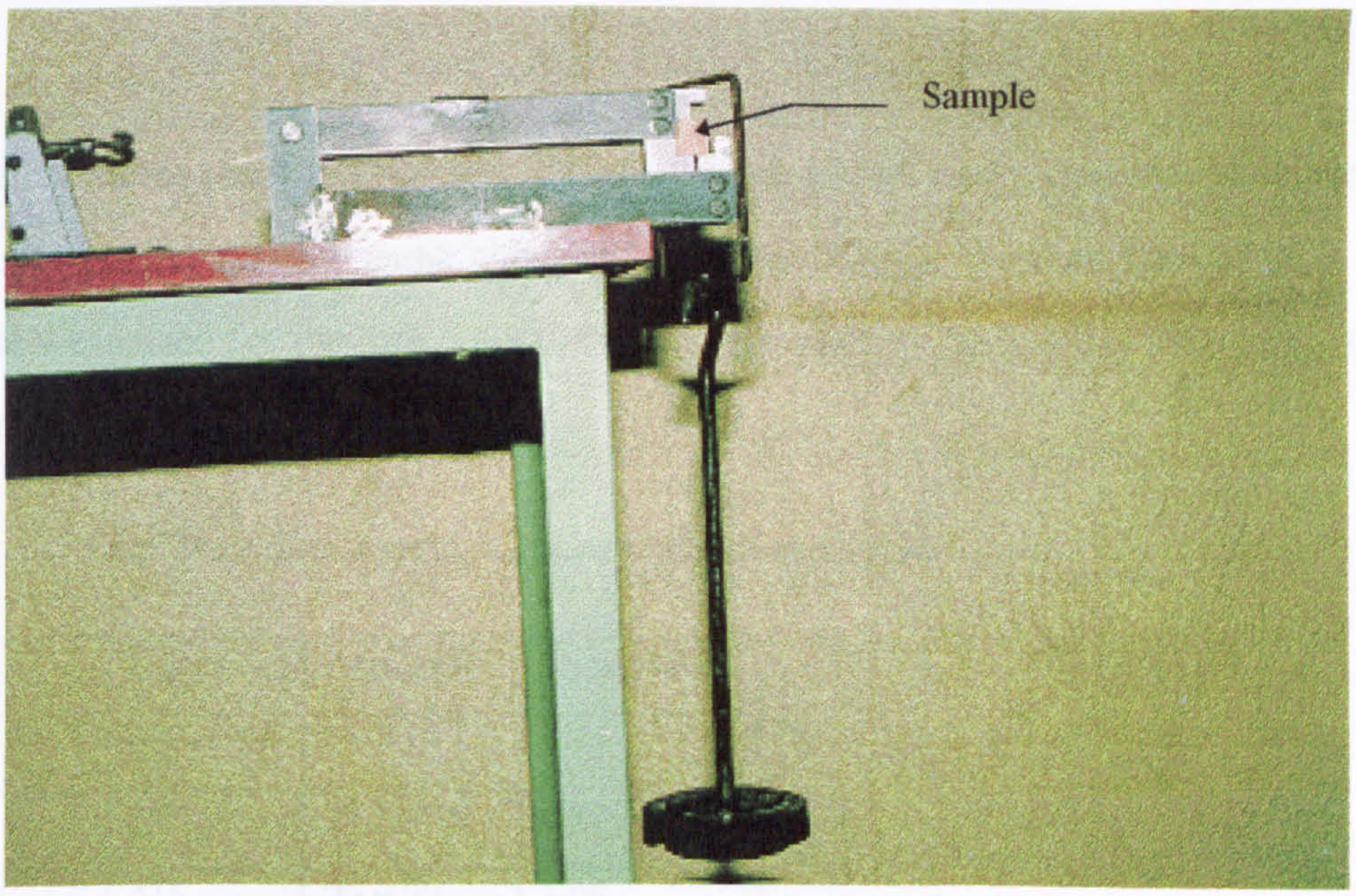


Figure 3-7 Shear stress test on the trunk sample

Figure 3-8 Leaf base compressive stress tool

3.1.6. Leaf base shear stress in longitudinal direction

The shear stress testing of the leaf bases was conducted employing the same method and tool used for the tree trunk shear test (Section 3.1.4.). Sample sizes of 20 × 20 × 20 mm were used for the test. The test results are shown Table 3-5 with a mean stress value of 1 MPa..

Table 3-5 Leaf base shear stress in longitudinal direction

No.	Force, N	Area, mm^2	Stress, MPa
1	450	21 × 19	1.13
2	380	19.5 × 20	0.97
3	415	20 × 20	1.04
4	420	20 × 20	1.05
5	410	20.5 × 19	1.05
6	435	19 × 19	1.20
7	290	19.5 × 21	0.71
8	355	21 × 20.5	0.82
9	430	21 × 19.5	1.05
Average			1.00
Standard deviation			0.15

3.2. Summary of the stress test results

The stress test results are summarised in Table 3-6. These results were used for the design purpose, as they define the limits of the permissible machine loads applied to the tree. These are the stress limits at which damage to the tree occurs. To satisfy the design criteria (PDS), the machine should exert forces and stresses smaller than these values. Because the fibres are positioned in longitudinal direction and mostly parallel to each other for the whole length of the tree trunk they exhibit poor resistance to longitudinal shear stress and it can be seen from the test results.

The leaf bases on which the machine is moving can tolerate greater radial compressive stress than the tree trunk therefore, the lower stress value which is the tree trunk was adopted as the design criterion.

Table 3-6 Stress test results summary, MPa

Stress	Tree trunk	Leaf base
Tensile in longitudinal direction	60.10	
Compressive in longitudinal direction	5.34	
Compressive in radial direction	2.96	6.38
Shear in longitudinal direction	1.10	1.00

3.3. Harvesting time of bunch cutting method

The total harvesting time for 9 trees harvested by the bunch cutting method were recorded. During harvesting the worker climbs the tree and gets into the crown. He then prepares the crown by cutting off the sharp spines and excess foliage in order to prevent injury and have a clear view for cutting a bunch. The bunch is then cut and tied to a rope and lowered to the ground. The rope is pulled up to the crown and the preparation, cutting and lowering is repeated for the next bunch. The total harvesting time in Table 3-7 was calculated by addition of the following times:

Total harvesting time = climbing time × 2 + Number of bunches × (cutting time + lowering time + preparing time) + time to move to the next tree.

The partial times and the total harvesting time can be seen in Table 3-7 which gives an average time of 1231 seconds (20 minutes and 31 seconds) and a maximum yield of 50 kg.

Table 3-7 Partial and total harvesting time of bunch cutting method

Factor or action	Tree number									Ave.
	1	2	3	4	5	6	7	8	9	
Height, m	14.1	15.3	12.1	16.2	11.2	10.5	9	11	10.4	
Climb, sec	45.5	56.7	60.5	81	46.7	30.9	36	36.7	29.7	
No. Bunch	8	7	6	7	8	9	7	7	8	
Bunch weight, kg	6	6.49	5.68	5.6	6.8	5.1	7.21	6.32	5.81	
yield, kg	48	45.4	34.1	39.2	50.4	45.9	50.0	44.3	46.5	45
Cut, sec	7.6	8.7	7.2	10.2	12.4	13	11.1	14	8	
Lowering, sec	80.5	87.3	70.9	87.8	83.1	60	104.5	82	58	
Preparation, sec	56.7	62.1	67.1	54.9	72.3	74.8	49	81	46.3	
Move to next tree, sec	6.8	5.7	5.8	7.9	5.7	8	6.8	8.6	9	7.1
Total harvest, sec	1255	1226	998	1240	1441	1400	1231	1321	967	1231

The total harvesting time for bunch cutting method from the table is 1231 seconds (20.5 minutes) considering an efficiency of 75 % the actual average harvesting time for a tree is 1641 seconds (27.3 minutes).

3.4. The tree and fruit size measurements

Relevant tree and fruit sizes were measured during the field study from 25 yielding trees of specified varieties. The trees were selected randomly from several groves in the two cities. The data are presented in Appendix A and were used for the machine design purpose. Table 3-8 shows a summary of the more important results. As it is seen from this table the tree trunk was measured at two points, near the base and near the crown, the date tree diameter decreases with the tree height and all of the trees have a smaller diameter near the crown.

The tree spacing measurement shows that the distance between the trees is not uniform. In many places other trees such as orange or lemon, were planted between the date trees. To investigate the problem of low tree spacing, the shortest distance between the measured date tree and the nearest tree, which may be a different species, was recorded. It is seen from the table that the minimum distance of 0.74 m in "distance to nearest tree" row exists in the table. This is very small for operation of the conventional machines even if other obstacles such as irrigation channels do not present a problem.

To measure the crown diameter and other parameters on the tree crown a worker climbed the tree, his climbing speed was recorded to be used as a criterion for selection of the machine speed.

The tree trunk surface is covered and made from the leaf bases of the dry leaves which are pruned every year and presents an uneven surface for the machine to move on. The leaf bases grow in a spiral path on the surface, so their pitch, depth and width were measured and were used to select wheel diameter for the machine. The yield was measured to estimate the machine payload and total weight.

Three harvesting method identified in the groves are: bunch cutting (B.C.), bunch shaking (B.Sh.)and fruit picking (F. P.) and they are specified for each tree in the tables of appendix A. A combination of main methods was also mentioned for those varieties where date growers harvest the fresh fruits by fruit picking method, keeping some fruits on the tree until at the end of the season, growers cut the whole bunch and collect the dry fruits.

The thesis work was concentrated on the bunch cutting system but the data for other systems were also collected because the project development can then consider the other harvesting methods.

Table 3-8 Summary of data on date tree and fruit sizes

Feature	Average	Standard deviation	Max.	Min.
Tree height, m	10.3	3.6	17	6.5
Diameter at ground, mm	639	104	850	500
Diameter at crown, mm	441	93	640	300
Row spacing, m	4.6	0.69	5.7	3.5
Across row spacing, m	5.8	0.96	7.3	4.1
Bunch & stalk weight, kg	9.7	2.66	14.5	6
Bunch diameter, mm	510	74	650	400
bunch length, mm	772	178	1110	420
Number of bunches	8	2	11	7
Leaf base pitch, mm	94.3	19.9	115	60
Leaf base height, mm	60	10	75	41
Leaf base width, mm	262	30	301	234
Yield, kg	66.9	28.2	127.6	34.1
Distance to nearest tree, m	3.66	0.74	5.2	2.5
Climbing speed, m/sec	0.31	0.08	0.41	0.18

3.5. Measurement of the coefficient of friction

The aim of this measurement was to establish the coefficient of friction between the wheel and the tree trunk surface. Three leaf base samples were prepared using a hand saw. The test was done with three replications for each sample.

To simulate the real conditions a piece of rubber from a tyre was glued to a steel holder. The rubber sample was loaded to 20 N. The weight of the rubber holder and the rubber sample was 2.1 N. The total weight of the unit was 22.1 N. One end of a plastic rope was tied to the rubber holder and the other side to a spring balance.

The sample was fixed onto the table by a clamp. The rubber was placed on the sample and was pulled by the spring balance which displayed a maximum of 100 N with 1N resolution. At the point that the rubber starts to slip over the sample the amount of force was read from the spring balance. Results of the test can be seen in Table 3-9.

The coefficient of friction (μ) could be calculated from the equation:

$$\mu = F_h / F_v \qquad (3.8)$$

where F_h is force read from the spring balance, N. F_v is total weight of the unit, N.

Table 3-9 Horizontal force needed to pull the rubber on the leaf base sample

sample		A			B			C		Average	Standard deviation
replication	I	II	III	I	II	III	I	II	III		
F_h , N	17	18	17	21	20	20	19	18	19	18.8	1.4

Substituting 18.8 for F_h and 22.1 for F_v in Equation (3.8) hence μ has an average value of 0.85.

CHAPTER 4

DETAIL DESIGN OF EXPERIMENTAL MACHINE

4. Detail Design of Experimental Machine

The detail design of the concept elements was conducted using concept theory and equations and data generated from the study of the interaction between machine and tree. The PDS was used as a guide and control criteria.

A full size experimental machine was manufactured in the Silsoe College workshop and was used to investigate the theory and the machine performance.

4.1. Wheel radius

The wheel radius was calculated by three methods explained in Section 2.5.8 and the greatest one was selected for the machine as follows:

1. Wheel radius in relation to the leaf base pitch: By this method (Section 2.5.8.1) wheel radius should be bigger than the maximum leaf base pitch (length db in Figure 2-10) which is 115 mm from Table 3-9.

2. Wheel radius in relation to the tree, axle clearance: By this method the wheel radius was calculated from Equation 2.26 which is:

   $$r = (\text{tree radius, mm} + 70) - \sqrt{(\text{tree radius,mm})^2 - (0.5 \times \text{wheel spacing,mm})^2}$$

   From Table 3-9 maximum tree radius is 425 mm. Wheel spacing from Equation 2.23 for μ equal to 0.85 is 550 mm. Substituting these values in Equation 2.26 gives r equal to 171 mm.

3. Wheel radius in relation to the leaf base height: Using this method (Section 2.5.8.3) the greater wheel radius calculated from methods 1 and 2 which is 171 mm was substituted in Equation 2.30 and a tyre deflection of 23 mm was calculated. A tyre with an approximate radius of 171 mm has a height section of 80 mm. 23 mm is 29% of this size tyre section and according to Section 2.5.8.3, 171 mm is an acceptable because the deflection is less than 50% of the section height. To calculate tyre deflection by Equation 2.30, α was determined 22.5° from Equation 2.29 for $\frac{W_s}{W}=0.5$ (refer to Section 2.5.8.3) and μ = 0.85 from Section

3.5. s in Equation 2.30 is 34 mm which is the absolute surface roughness i.e. the difference between the maximum and minimum leaf base height which are relatively 75 and 41 mm from Table 3-8.

The second method, r equal to 171 mm, gave the reasonable wheel radius of 171 mm and the nearest standard wheel radius was 175 mm which was selected for the machine. The load capacity of this wheel is 1500 N with the inflation pressure of 0.17 MPa.

4.2. Machine main dimensions

The dimensions were selected using the tree and fruits maximum and minimum sizes and date bunch weights presented in Table 3-9 and Appendix A. Using these values, the main machine dimensions can be seen in Figure 4-1. The spacing between axles A and B , y_B ,was selected at 1000 mm. If the height was more than this value, unloading the dates from the basket may prove to be difficult. Axle C was decided to have 500 mm height (y_C) relative to axle A when it is on the minimum tree diameter which is 300 mm. These values determine the exact location of the horizontal hinge which connect the one axle unit to the two axle unit. The point G_d is the centre of gravity of the harvested dates. The presented dimensions of y_B and y_C were optimised by a spread sheet. The criteria were resulting a positive value for B_x (Equation 2.19) as will be explained more in Section 4.4, an acceptable stress on the tree and ergonomics factors.

4.3. Machine weight

It was necessary to predict the machine weight in order to select the motor power, bearings, power transmission system and also to design the frame, fasteners and other machine elements. The weight was predicted by an estimation of the component weights from catalogues, or from similar parts available in the workshop.

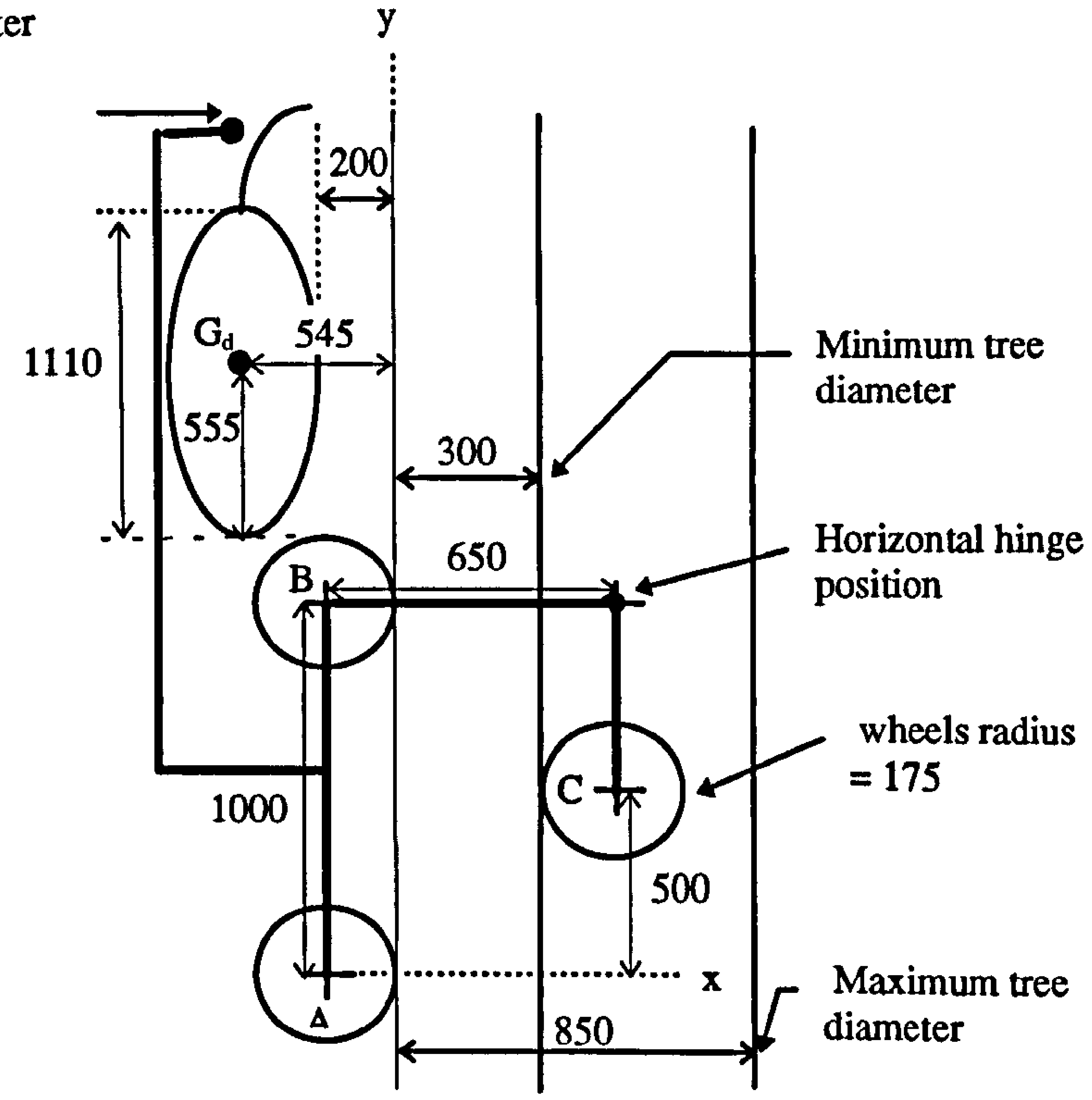


Figure 4-1 Model dimensions, mm

Main beams used in construction of the climber frame are shown in Figure 4-2. A second frame similar to that illustrated in the figure is on the opposite side of the tree. For this reason weight calculations for these sections were multiplied by a factor of two.

Small squares on Figure 4-2 refer to five horizontal beams which were designed to connect the two sides of the frame together. The frame weight was calculated using the hollow square steel 25 × 25 × 3 mm which weigh 2 kg/m (AHA 1994). Detail frame weight calculations are presented in Appendix B.

The length of the centre of gravity, x_w, was also required. To simplify the calculation of x_w, the centre of gravity of dates was assumed to be at point G_d in Figure 4-1 and the machine components centre of gravity were assumed to be at:

1. Centre of axle A, for components near or in the vertical plane that passes through the axle A and also for the "two axle unit" frame weight.

2. Centre of axle C, for the components near it's centre or in a vertical plane that passes through axle C and also the "one axle unit" frame weight.

Half of the stabiliser weight was located in point A and half in point C.

Table 4-1 shows the predicted component weights and their location.

The total machine weight is 115.2 kg which exerts a force equal to 1152 N from which 737 N act from point A and 415 N from point C. The results were used to calculate the length of centre of gravity, x_w.

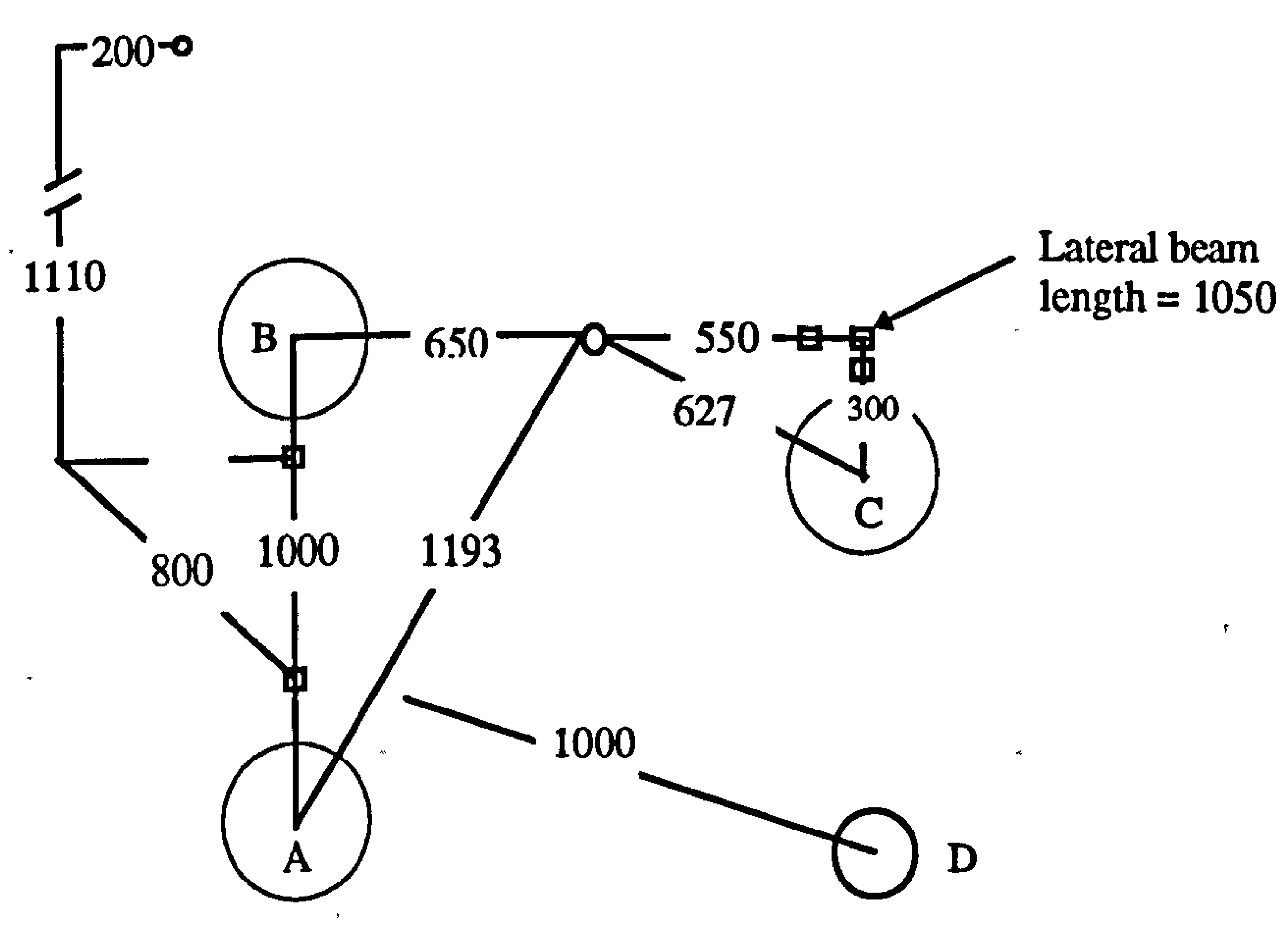


Figure 4-2 Machine frame beam lengths, mm

4.3.1. The centre of gravity

The length of the centre of gravity, x_w, was needed for design calculations. To find x_w the total machine weight was assumed to be a vector acting from point G in Figure 4-3.

The co-ordinate system is located at the contact point of the lower wheels (A) with the tree. External forces and positions were imported from Table 4-1. x_W was calculated using the following equations:

Table 4-1 Machine elements weight and weight vector acting point

No.	Element	amount	Element weight force (N) and the location	
			Centre of axle A	Centre of axle C
1	Motor A		42	
2	Controller A		10	
3	Chain and sprocket A		100	
4	Motor C			42
5	Controller C		10	
6	Chain and sprocket C			100
7	Wheels A	2	56	
8	Wheels B	2	56	
9	Shafts A and B	2	91	
10	Shaft bearings, A,B	4	40	
11	Wheels C	2		56
12	Shaft C	1		45
13	Shaft bearing, C	2		20
14	Spring	2	60	
15	Spring tension mechanism	2	30	
16	Main Frame	1	156	
17	Two wheel frame	1		122
18	Arm weight	1	56	
19	Stabiliser	2	30	30
	Total		737	415

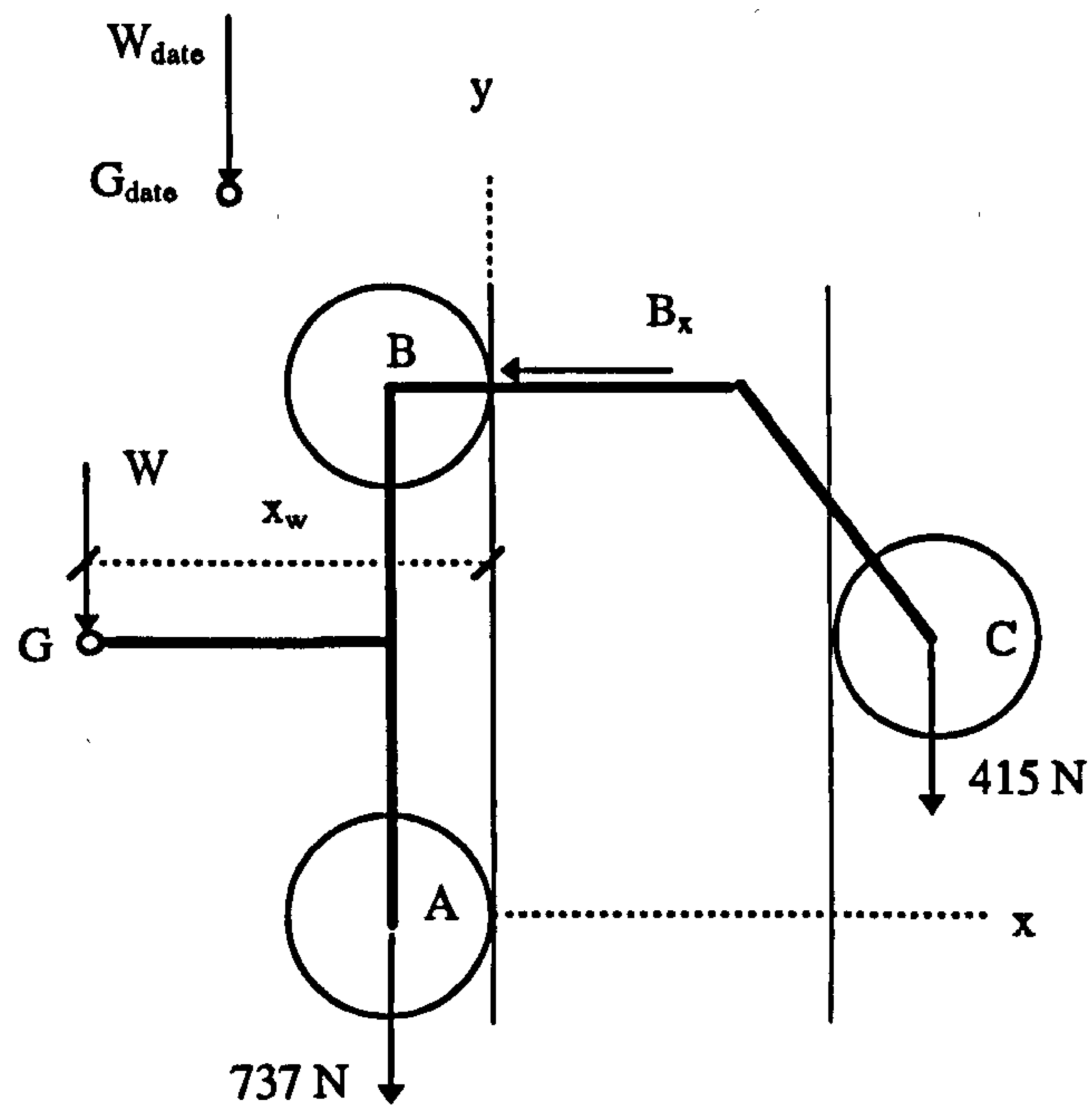


Figure 4-3 The distribution of date and component weights on the machine frame

$$W = W_{machine} + W_{dates} = 1152 + W_{dates} \tag{4.1}$$

x_W is equal to (Meriam and Craige 1993):

$$x_w = \frac{\sum xF}{\sum F} \tag{4.2}$$

Where F is any external force including the dates and the machine component weight and x is the horizontal distance of F from the co-ordinate system in Figure 4-3. Substituting force values from Figure 4-3, dimension sizes from Figure 4-1 and W from Equation 4.1 in Equation 4.2 gives:

$$x_W = \frac{(d+r)\times 415 - 545\times W_{date} - r\times 737}{1152 + W_{date}} \tag{4.3}$$

Where d is tree diameter and r wheel radius in mm.

4.4. Checking dimensions and finding external reaction forces

The dimensions selected for the machine can be checked since the unknown parameter length of centre of gravity, x_w, was formulated by Equation 4.3. If substituting parameter values including x_w in Equation 2.19 for a minimum and minimum tree diameter produce a positive value of B_x as was explained in Section 2.5.6 the dimensions are suitable for the machine. Table 4-2 shows the summary of data and calculated results including B_x.

To calculate parameters Figure 2-9 and the parametric Equations were used in the following order: Equation 2.19 for B_x, Equation 2.17 for C_x, Equation 2.15 for C_y, Equation 2.12 for A_x and Equation 2.13 for A_y.

Table 4-2 Known dimensions and data from the machine and the calculated results from the data in the model equations

Data						Results					
d mm	μ	y_c mm	y_B mm	W_{date} N	$W_{machine}$ N	x_w mm	B_x N	C_x N	C_y N	A_x N	A_y N
300	0.85	500	1000	500	1152	123	40	992	843	952	809
850	0.85	900	1000	500	1152	-15	216	1080	918	864	734

The values shown for Y_c and y_B in the table are results of several attempts on a spread sheet to find suitable values which are reasonably high and also give positive value for B_x in Equation 2.19.

For the shown values of data in the table B_x was positive, therefore, the selected dimension sizes for the machine were acceptable.

The values of C_x and C_y were multiplied by a factor of 1.5 for the design purpose to produce extra traction required when the other axle is climbing a step (leaf base) as it was explained in section 2.5.6.3.

4.5. Checking tree safety against machine stresses

At this stage that the machine weight, centre of gravity and tyre forces had known the tree safety was checked against the stresses applied by the machine. To calculate the working stresses Equations 3.1 to 3.4 were used. The tyre pressure used in Equations 3.1 and 3.2 was 0.17 MPa from Section 4.1. The critical condition of stresses is when the machine with full payload (50 kg) is working on a minimum diameter tree(300 mm), therefore, the data of minimum diameter tree from Table 4-2 were used for Equations 3.3 and 3.4.

Table 4-3 shows the results of the calculations. The tree relevant failure stresses of Table 3-6 were used for the analysis. The safety factor (S) column which is failure stress divided by working stress shows that even a small young date tree can tolerate different stresses with the safety factors range from 7 to 400, therefore the machine is unlikely to damage the tree, and PDS on tree safety is satisfied by this analysis.

Table 4-3 Safety factor of stresses applied to the tree

Working stress			Failure stress			Safety factor (S)
Type	Calculation source	Value, MPa	Type	Calculation source	Value, MPa	
σ_r	Equation 3.1	0.17	$\sigma_{r\,max}$	Table 3-6	2.96	17
τ_L	Equation 3.2	0.14	$\tau_{L\,max}$	Table 3-6	1.00	7
σ_c	Equation 3.3	0.19	$\sigma_{c\,max}$	Table 3-6	5.33	28
σ_t	Equation 3.4	0.15	$\sigma_{t\,max}$	Table 3-6	60.10	400

4.6. Motor and power supply design

The machine needs sufficient power to lift the total vertical load and to overcome the rolling resistance of the wheels. The machine has two driving axles and as it was discussed in Section 2.5.6.3 a factor of 1.5 is required in power equation for the moment that wheels of one driving axle are climbing a step (leaf base) and lose part of their traction force. At this moment wheels of the other driving axle have extra reserve traction power to assist the climbing axle and move the machine, therefore, the power equation is:

$$\text{Power} = 1.5\ Fv \qquad (4.4)$$

where F is the total weight that the machine must lift plus rolling resistance and is equal to the summation of following forces in Figure 2-9:

$$F = C_y + A_y + 0.1(C_x + A_x + B_x) \qquad (4.5)$$

v is maximum worker climbing speed, 0.41 m/sec, which was set as the target machine climbing speed. The wheel rolling resistance is estimated to be 10 % of the horizontal load on it (Liljedahl et al 1989), therefore 0.1 in Equation converts the horizontal load on the wheel to the vertical rolling resistance. To calculate F, values from Table 4-2 for the maximum diameter tree were substituted in Equation 4.5, because F with these values is greater than values from the minimum diameter tree. The resulted F is 1868 N. Substituting F and V in Equation 4.3 gives the design power of 1149 Watt.

4.6.1. Motor and controller type

Two, 24 V permanent magnet dc electric motors, type PM63, were selected (EMD 1995) for the machine. The motor is brush type and can produce a maximum power of 700 watt. The control system is a logic programmable controller (PLC) less than 0.5 kg in weight. Neither hydraulic, pneumatic nor mechanical control systems can compete with the weight advantages offered for this size of multi-function control system. The PLC is a computer programmed to implement the logic functions. The controller controls the motor speed, direction, acceleration, and the maximum current to protect the motor. It controls the output power by a full bridge, ultrasonic pulse width modulation circuit (Dynamic 1993). The motor was equipped with an electric brake that weights only 0.5 kg which is controlled by the PLC and also with a 12.5

to 1 reduction ratio worm gear box. Motor and gear box weigh 4.2 kg. The total motor, gear box, brake and control system weigh 5.2 kg. Figure 4-4 shows the conventional (brush type) permanent magnet dc motor and worm gear box. The electric brake can be seen in the figure. The handle on top of the electric brake is the manual brake release for use in case of a break down and emergency landing. The prototype moved downward slowly when two hand brakes were released. A small plug-in programmer is used to down-load the desired values for speed , acceleration, maximum power , brake delay, and other parameters to the PLC by selecting the required values from setup menu on the programmer screen. It is also possible to upload a programme from the controller to the programmer for future storage and use. Figure 4-5 shows the controller, the small plug-in programmer can be seen which was used to down-load the control parameter values into the controller.

One of the main features of the controller is the constant speed selection . It runs the motor with a constant speed independent of the payload using the Motor Resistance value. This feature was helpful and was used in all experiments because with the constant climbing speed power calculations are easier.

Two 12 V batteries of 40 Ah, connected in series were used to supply the power for the motors.

Permanent magnet motors, PLC and battery operated system is a mechatronic approach in engineering design. It provides a system which can be more versatile and of higher performance than a traditional mechanical system (Bradley et al 1991). It is also useful for development of the project as an intelligent machine, making the system suitable for fitting required sensors and vision system in the future.

4.7. Power transmission system

The role of this system is to reduce the motor speed by a proper reduction ratio for the driving axles. This reduction ratio was calculated from the motor specification curves. From Section 4.6 the design power is 1149 watt. Two motors on axles A and C produce this power, each motor provides half of the total power, or 574 watt.

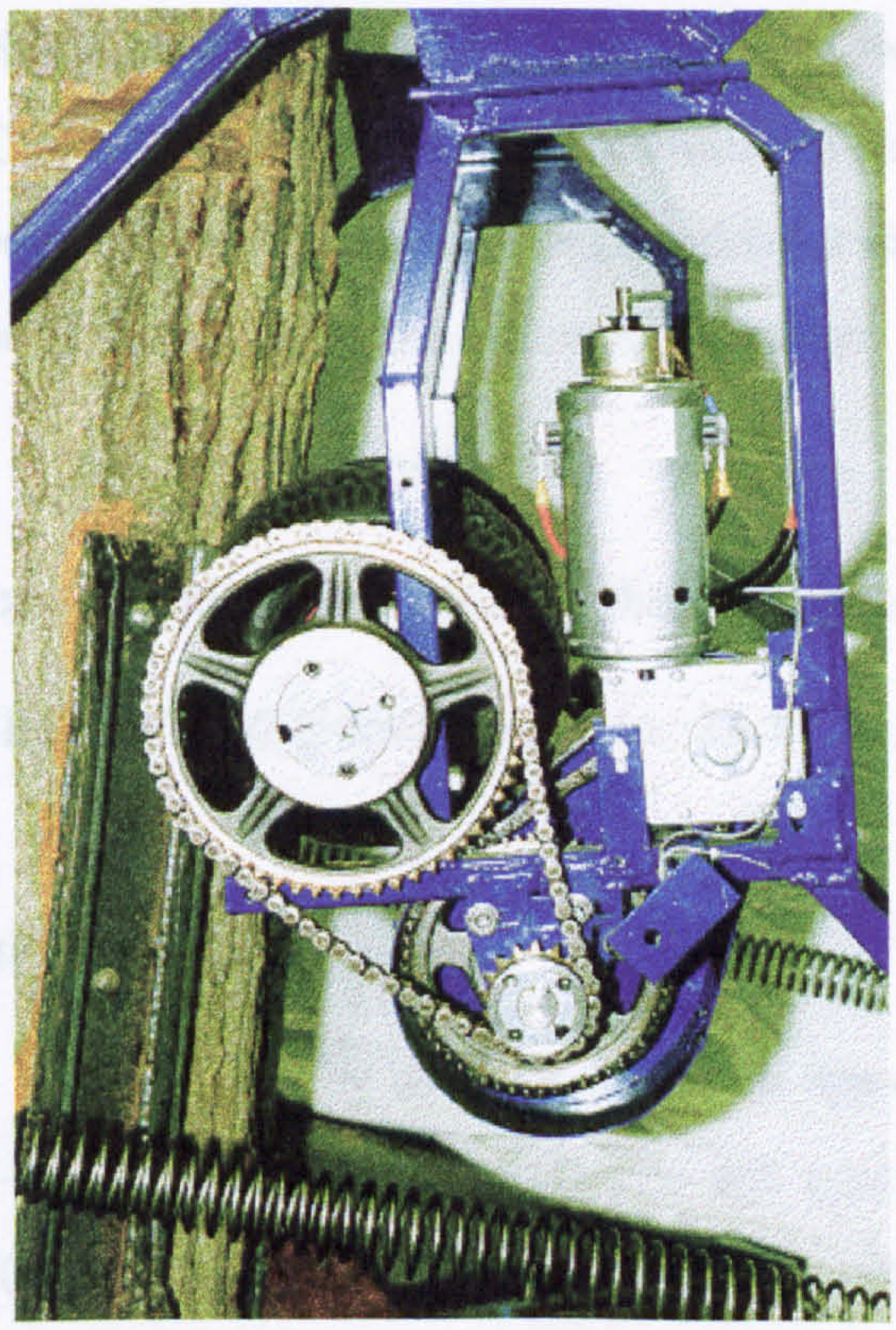


Figure 4-4 Motor, worm gear, electric brake and sprocket chain transmission system

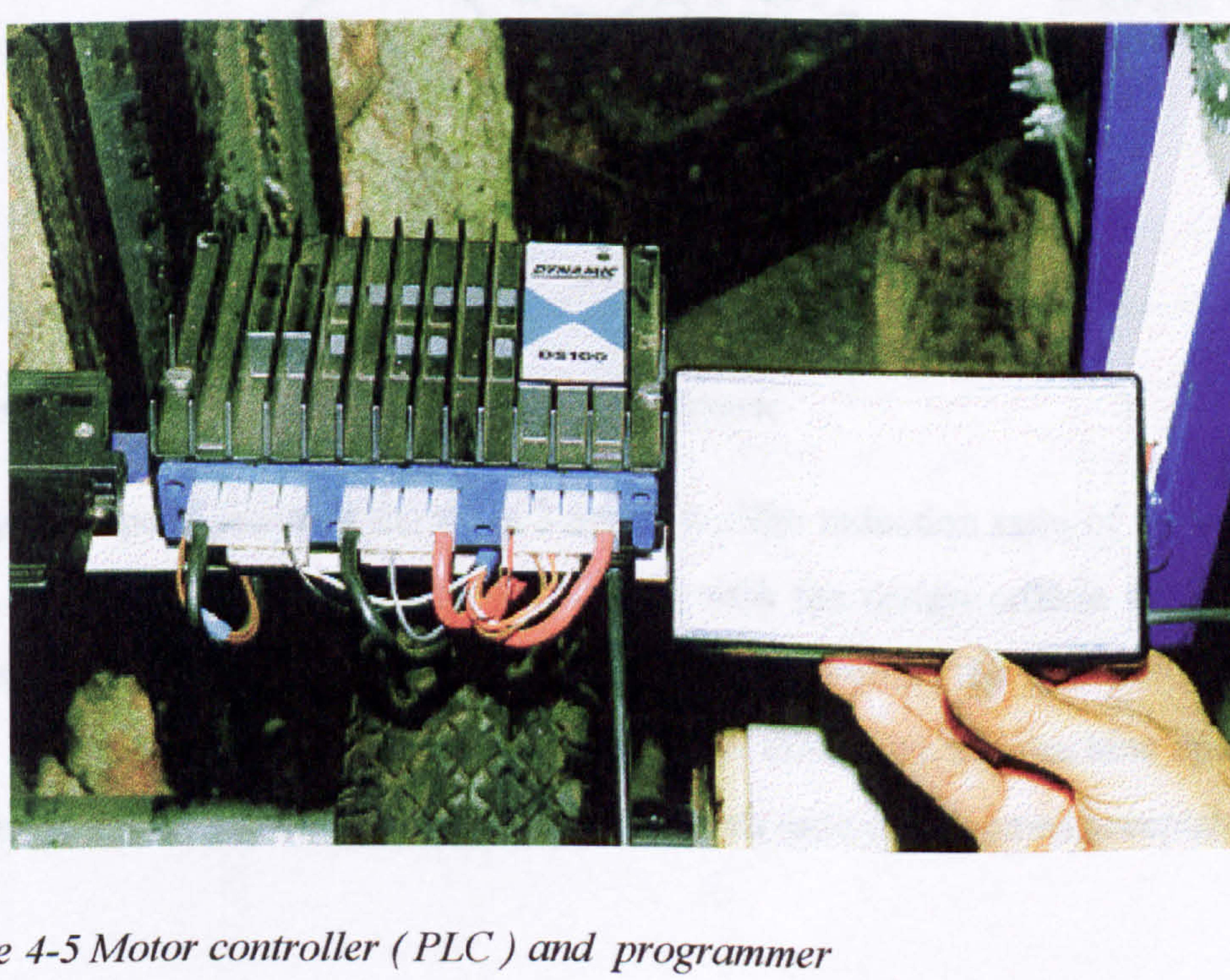


Figure 4-5 Motor controller (PLC) and programmer

From the motor specification curve (EMD 1994) at this power the motor output speed, N_m, in region C of the curves area is 300 rev/min and the torque is 18.3 N m. The required speed reduction ratio is equal to N_m divided by axle speed, N_a . Axle speed in rev/min can be calculated from the following Equation:

$$N_a = 60v / \pi D \qquad (4.6)$$

Where D is wheel diameter, 0.35 m; v is machine climbing speed, 0.41 m/sec.

Substituting values in Equation 4.6 gives N_a equal to 22.37 rev/min. Dividing N_m by N_a

gives the reduction ratio of 13.4.

This speed reduction was achieved by using two pairs of sprocket and chains, as is illustrated in Figure 4-6. The sprocket and chain size and types were designed from the Fenner catalogue (Fenner 1988) by employing the method explained in Appendix B.

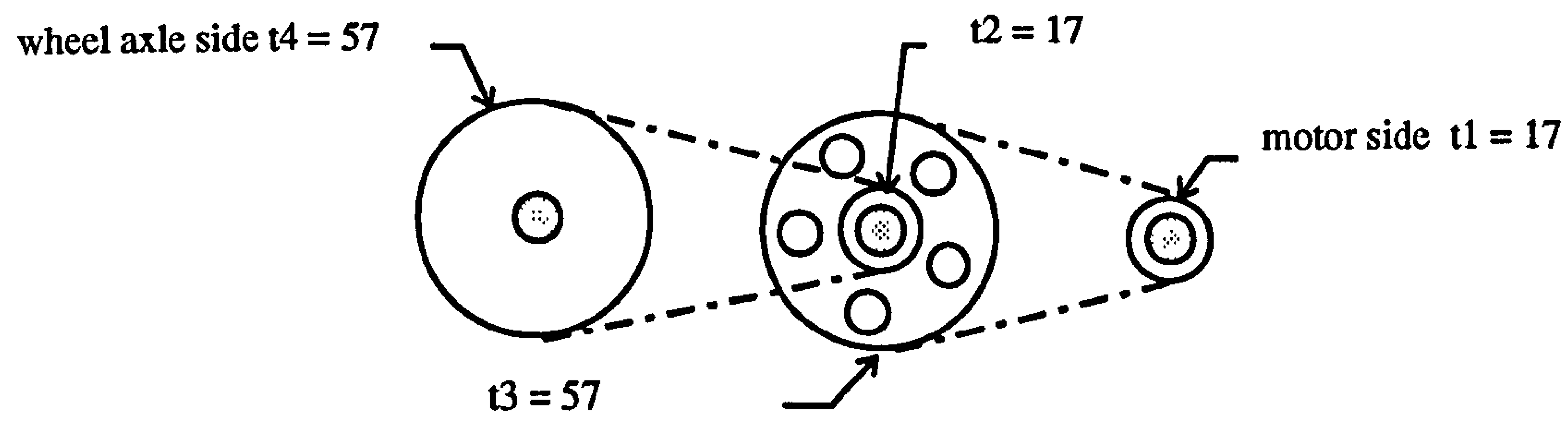


Figure 4-6 Sprocket and chain transmission system

The system specification is shown in Table 4-4. The reduction ratio of the system is 11.42. This ratio does not comply precisely with the design criteria of 13.4, but because of the standard number of teeth, or speed ratio in the selection table, the space limitation on the machine, and because at this stage the machine speed was more important than the payload that it can lift, this ratio was deemed acceptable.

The final drive sprocket and chain can be seen in Figure 4-4 (chain guard removed for clarity).

Table 4-4 Chain drive specification

No.	Number of teeth	Pitch mm	Quantity	Chain length mm
t_1	17	9.5	2	
t_2	57	9.5	2	788
t_3	17	12.7	2	
t_4	57	12.7	2	951

4.7.1. Wheel adjustment mechanism

The machine requires adjustable wheel spacing on each axle to be able to move on different diameter trees (concept G1 in Section 2.5.3) and use the advantage of self guiding system explained in Section 2.5.7 . A piece of tubing was welded to the rim hub of each wheel to extend the hub. The hub extension and the axle were drilled at 100 mm intervals. This mechanism provides the wheel spacing adjustment by fixing bolts through hub extension and the axle holes at desired 100 mm intervals. This can be seen in shaft drawings (Appendix E, attached floppy disks).

4.7.2. Driving shaft design

To design the driving shaft C the free body diagram is shown in Figure 4-7. C_1 and C_3 are bearing supports. C_4 represent the sprocket fixing place. Maximum bending stress ocures when two wheels on the shaft are placed on the middle of the shaft, Point C, at this position $C_1C_2 = C_2C_3$. C_{2x} and C_{2z} are the vertical and horizontal reaction forces of wheels on shaft C. C_{2x} is C_x and C_{2z} is C_y of Table 4-2 (for working on a 850 mm diameter tree, Cx = 1080 N, Cy = 918 N) multiplied by a factor of 1.5 for the leaf base climbing situation, as explained in section 2.5.6.3, therefore:

$$C_{2x} = 1620 \text{ N}$$

$$C_{2y} = 1377 \text{ N}$$

To facilitate working on the maximum diameter tree (850 mm) spacing between bearing supports of shaft, C_1C_3 , was selected 1050 mm. From the sprocket and bearing thickness measurments, C_3C_4 was selected 50 mm.

The required torque, T_m , for the driving shaft can be calculated from the following equation:

$$T_m = T_2 = C_{2y} \times \text{wheel radius}$$

Where C_{2y} is 1377 N and wheel radius is 175 mm which gives:

$$T_m = 240975 \text{ N.mm.}$$

The chain tension, F_m, created by motor which provides , T_m, was calculated from this equation:

$$F_m = T_m / r_4$$

Where r_4 is radius of sprocket t_4 which is 115 mm (sprocket specification, Table 4-4). Substituting values of T_m and r_4 gives F_m equal to 2095 N. Direction of, F_m , depends on the position of the motor on the chassis which is shown in Figure 4-7.

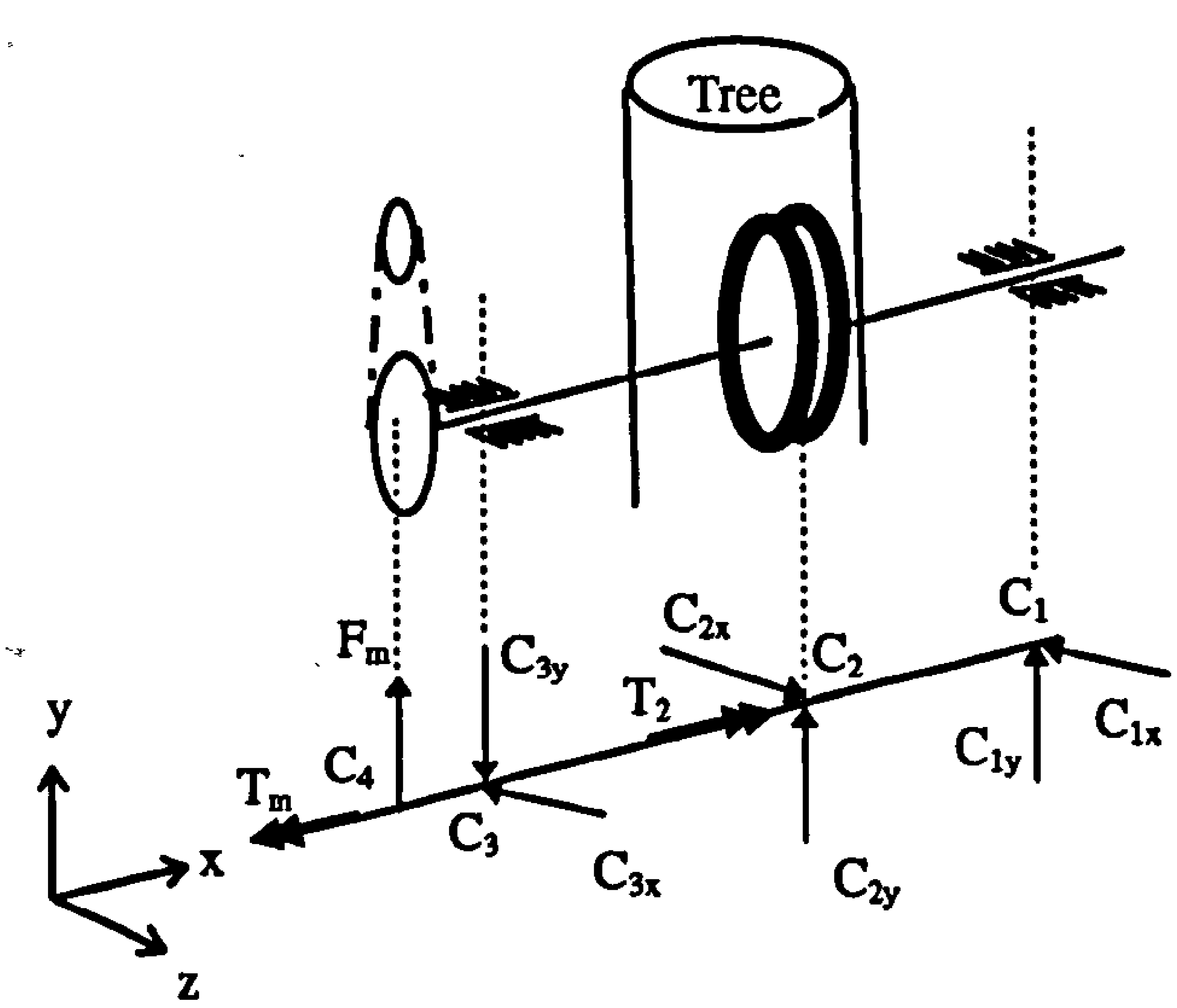


Figure 4-7 Driving shaft 3-dimensional free body diagram

Using above values the maximum bending moment in the shaft was calculated (details in Appendix B).and is:

$$M= 525792 \text{ N.mm}$$

This moment was used to select the driving shaft diameter.

4.7.2.1.Driving shaft diameter

A 35 mm outside diameter shaft was selected in order to fit the wheel hub, the inside diameter was selected 25 mm to fit in bearings and sprocket hubs. On the basis of these values the yield stress of the required steel to resist the forces applied to the shaft was calculated (Spotts 1978). The result of the calculation is 369 MPa (details in Appendix B). This level of stress is valid and lies in the range of commercial shafts strength, therefore the selected diameters of the shaft were acceptable for the machine.

4.7.3. Bearing selection

Each driving shaft is supported by two bearings. Bearing selection was based on the basic dynamic load rating, C (SKF 1994). For the operating hours of 3000 suggested for agricultural machinery a 35 mm inside diameter self aligning ball bearing with adapter sleeve and SHN-plummer block housing with designation number of 1208 EK was selected (details Appendix B). This bearing can tolerate the shaft deflections very well, does not need special casing on the frame therefore the frame can be easily manufactured and because of adapter sleeve, the driving shaft can be fitted quickly and simply.

4.8. Traction system

A pair of springs were selected in Section 2.5.3, Part H to force wheels on the tree to produce the required traction. The effort was concentrated to use the minimum spring tension by choosing the best possible action line for the springs, therefore light springs can provide the required forces. These forces are C_{2x} and C_{2y} which positions are illustrated in Figure 4-7. These are forces that the springs should indirectly resist. The force diagram and a graphic vector analysis to find the best spring action line can be seen in Figure 4-8. It can be seen that when vector K is parallel with vector C it can resist vector C with minimum value.

Line pc and link ij of the "one axle unit" intersect at point k. This was therefore selected as a point at which the spring will be suspended and the spring tension force, vector K, acts to press wheels on the tree. Vector K was located parallel to vector C, it was achieved by fitting a small pulley on the "two axle unit" at the right point on the chassis. The winch rope which pulls the spring passes over this pulley to pull the spring in the required direction as shown in the Figure, then:

$$pc \times C = pk \times K$$

$$K = \frac{pc \times C}{pk}$$

Point p, horizontal hinge, and point k, spring suspension pin point could be in different places along the dotted line pk. The selected and presented positions for p and k, were optimised using ACAD 13, making an animated model, and considering the constrains such as space limitation, interaction with chain drive system, "one axle unit" rotation limitation to work on small tree diameter and ease of manufacturing the "one axle unit" frame. The values used in following calculation were measured from the final dimensions presented in Drawing No.1 (Appendix E). Substituting the values in K Equation gives:

$$K = \frac{340 \times 2126}{688}$$

$$K = 1050 \text{ N}$$

This is the force that the pair of springs should exert, therefore each spring produces half of the total force, or 525 N.

Rotating the "one axle unit" around it's horizontal hinges, p, to find the spring elongation for different tree diameters using ACAD (Soan et al 1995) showed that a spring with initial length of 320 mm and maximum length of 910 mm is required. Another important criteria in spring selection was a small rate (N/mm) which helps the machine to move smoothly over leaf bases. This information plus the required force of 525 N resulted in a spring with the following specification (SPEC 1995) being chosen:

Catalogue No. = T33360, initial length = 377 mm, extended length = 921 mm,

spring diameter = 55 mm, wire diameter = 5 mm, force at extended length = 543 N

spring rate = 0.84 N/mm, initial tension = 82 N

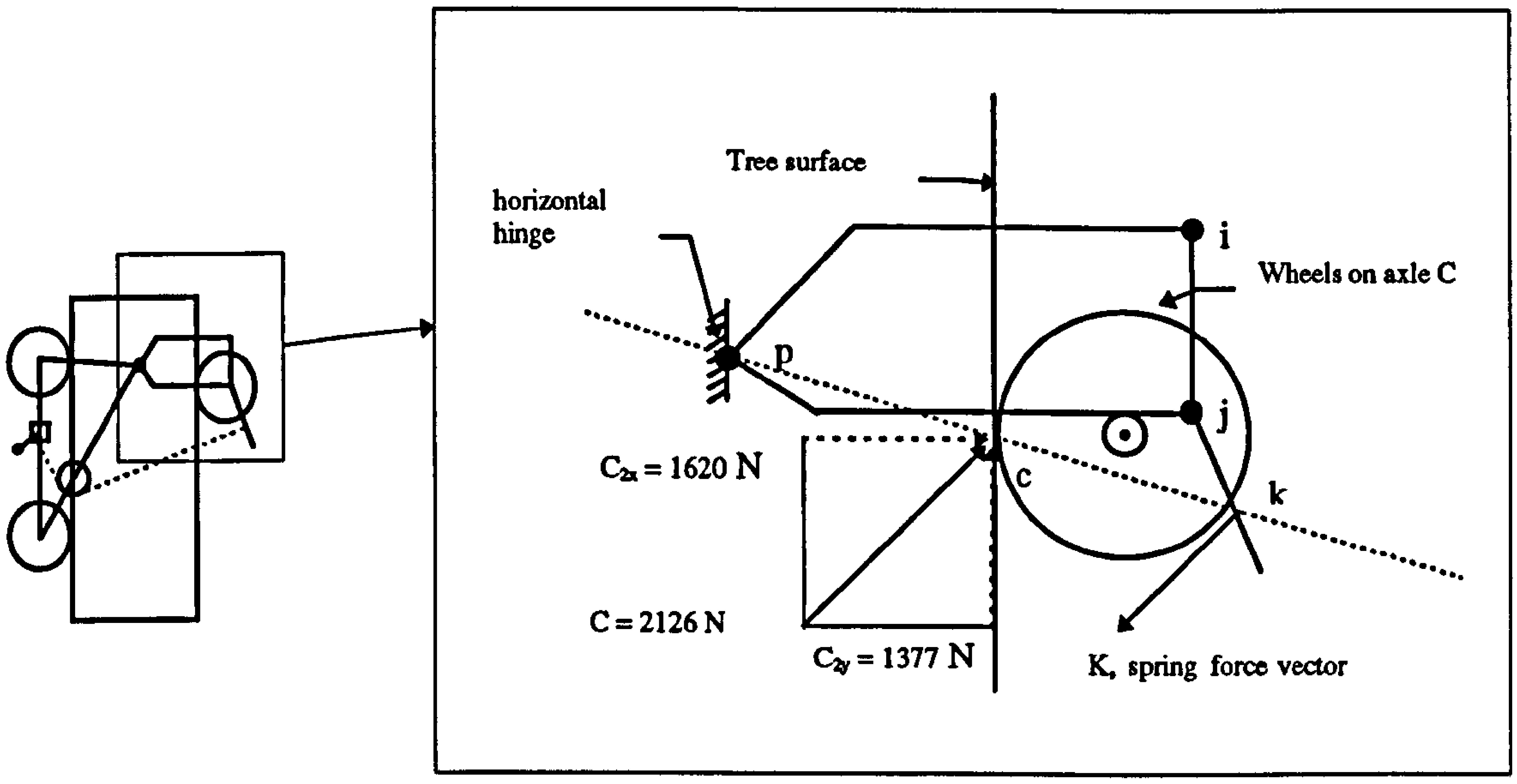


Figure 4-8 Detail of "one axle unit" and spring force analysis

4.8.1. Traction control mechanism

To load the springs for traction and to provide different extended lengths to control the traction and produce different traction levels, two alternatives were chosen in concept selection, Section 2.5.3. They are H1 and H3 or simple lever mechanism and hand winch. The spring specification in Section 4.8 shows that the spring force at extended length is 543 N which is much greater than the value that a worker can provide. The study of lever movement also showed that the lever displacement required to provide this force is also too much, therefore, the simple lever, H1 was rejected. The suitable mechanism was the hand winch or concept H3. Working with the hand winch is easy and safe for the worker because the ergonomics factors have been considered in it's design. A pair of hand operated winches, type DL600, that provide a maximum tension of 2750 N were selected being the nearest size available(KEY 1995).

4.9. Frame design

The design of the frame was based on the critical load locations, where the motor and gear boxes were mounted. The detail of the location can be seen in drawing of "one axle unit" presented in Appendix E on floppy disks.

"Method of joints" and "Method of sections" (Meriam and Craige 1993) were used to determine unknown forces in the frame. Sections were checked for the maximum bending moment and shear stress. The plane stress analysis was used in combined stress conditions. The shear stress caused by torsion and transverse loading were calculated and mentioned in the design.

The critical loading was on link hj in Figure 4-8, where the bearing C3 was mounted, and the maximum force from the wheels and transmission system were applied to the link. With the assumption that link pj is a straight beam, different hollow square sections were evaluated to establish the minimum suitable dimensions. Detail of the method is presented in Appendix B. Assuming 166 MPa allowable stress for mild steel, including a factor of safety of 1.5, the result is hollow square section of $25 \times 25 \times 2.5$ mm. Most frame elements were made from this section for considerations of ease of manufacture.

the climbing system was manufactured and pilot tests demonstrated that the machine could climb the tree with payload. The traction system could provide enough traction to lift batteries, so, they were fitted on the machine. To increase the payload capacity springs tension was improved by changing the suspension points on the "one axle unit" using two extension bars and the wheel spacing of the stabilisers were increased to enable their easier removal from the machine.

Figure 4-9 shows the manufactured experimental machine and main parts such as motor and transmission system, driving wheels, stabilisers, batteries and traction system including spring and hand winch can be seen from this figure.

Figure 4-9 Machine on the ground

CHAPTER 5

MACHINE EVALUATION PROCEDURE

5. Machine Evaluation Procedure

5.1. Introduction

The machine evaluation procedure includes the studies conducted on the experimental machine to evaluate the selected concepts, theory and the design guidelines for the future work. A measurement system was developed and then the following tests were carried out on the machine:

1. Machine weight for comparison with design value and calculation of the power and efficiencies.
2. Machine ground speed to test the speed and its ability for moving on smooth and uneven surfaces.
3. Climbing speed to determine the total harvesting time and comparison with other systems.
4. Pay load capacity to find the maximum weight that the machine can lift.
5. Stability of the machine with respect to the wheel spacing and payload positioning to establish the limits of wheel spacing for the safe and stable climbing.
6. Climbing ability with one driving axle to compare the results with theory.
7. Coefficient of friction between the tree and the tyre to relate it to the wheel slippage.
8. Machine preparation time and harvesting time, to find the most time consuming tasks and find a solution to improve them if they are slower than other harvesting systems.
9. Power consumption concerning the effect of different payloads, wheel spacing and wheel static load to establish the power requirement for different machine settings and machine sizes. These tests are named "ground" and "climbing" tests.
10. Efficiency tests to evaluate the machine power performance.
11. Slippage tests to confirm that the wheel slippage is within a reasonable range.

5.2. Measurement system

To test the machine a measurement system was designed as is seen in Figure 5-1. The system could measure the torque, speed, wheel slippage and the input electric power that motors consumed and had two main units:

- Sensors
- Data acquisition system

There were four main sensors in the measurement system:

1. torque sensor
2. speed sensor
3. wheel slippage sensor
4. input electric power sensor

These sensors in conjunction with the data acquisition system capture signals during the tests. Each box in Figure 5-1 shows an element of the system as follows:

The "torque meter" connected to axle A was used to measure the axle torque in static condition, when the machine was fixed to the ground, to draw relevant characteristic curves of the motor and transmission system.

The "slippage sensor" reads the number of turns of wheels on axle C and showed it on the "counter display", the number was then converted to the wheel slippage for the test height.

To calculate the output mechanical power, the speed of the driving axle C was required for each test. A "speed sensor" was connected to the transmission system of axle A to measure the instantaneous speed of the axle. It was a small dc generator and was attached on the motor out put shaft by a plastic coupling. The generator was bolted to the sprocket guard as can be seen in Figure 5-3. It rotates at the same speed as the motor output shaft but was calibrated to measure the axle speed. The calibration method is explained in Section 5.3.2. The product of this speed by the torque gives the axle power.

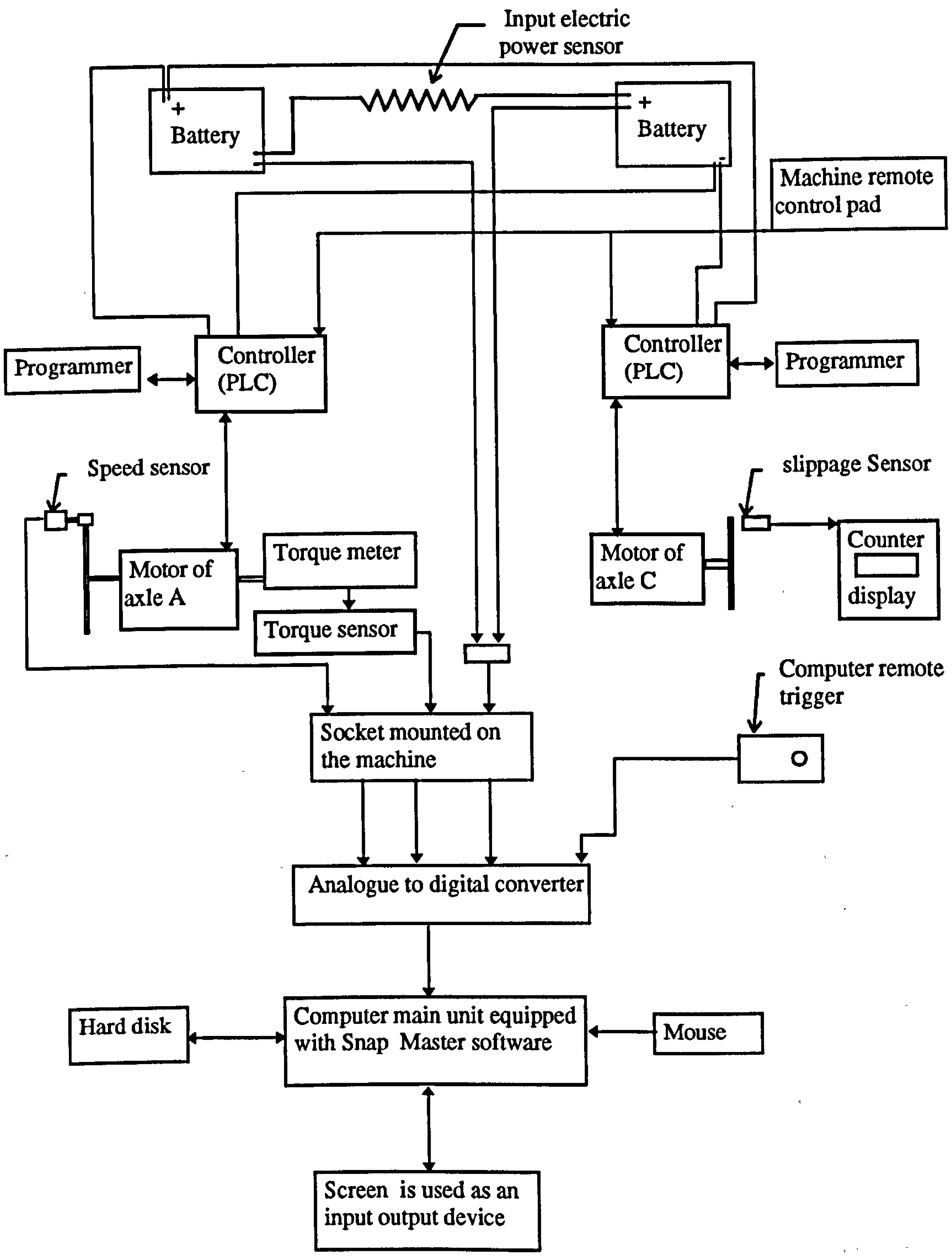


Figure 5-1 The diagram of the measurement system

The "input electric power sensor" as can be seen in Figure 5-1 is the connecting wire between the two 12 volt batteries. The voltage drop across this wire was measured and used to calculate the electric power consumption. The resistance of the wire was measured by an accurate meter and was 0.0052 ohms. The method of measurement is explained in Section 5.3.3.

The explained sensors were connected to a computer equipped with a data acquisition system through the "socket mounted on the machine" and the "analogue to digital converter".

The "computer remote trigger" was a push switch near the machine to start the computer in the control room at the beginning of each test.

The motors were controlled by the "controller (PLC)" and control parameter values such as maximum climbing speed were down loaded to the PLCs using the "programmer". Then the machine was controlled by the "control pad" during the tests.

5.2.1. Torque sensor

To measure the instantaneous torque required for the calculation of the output mechanical power in axle A, a tube tension dynamometer was used as a torque sensor (Figure 5-2). The main element of the sensor is a steel tube 10 mm outside diameter and 1 mm thickness with a full bridge of strain gauges . The tube tension dynamometer was connected to the arm of a friction type torque meter which was designed for this purpose using the method explained by Collett and Hope (1990). The torque meter is made from two wooden blocks with a semi-cylinder hole at the middle of one face. These two blocks squeeze the axle when two designated nuts are tightened. The tube tension dynamometer is connected to the torque meter arm on one end and to the ground on the other. When the shaft is rotating and the torque meter nuts are released, the torque is zero and the axle rotates at maximum speed. Tightening the nuts creates torque and reduces the speed. Different levels of torque could be applied to the axle by tightening the nuts.

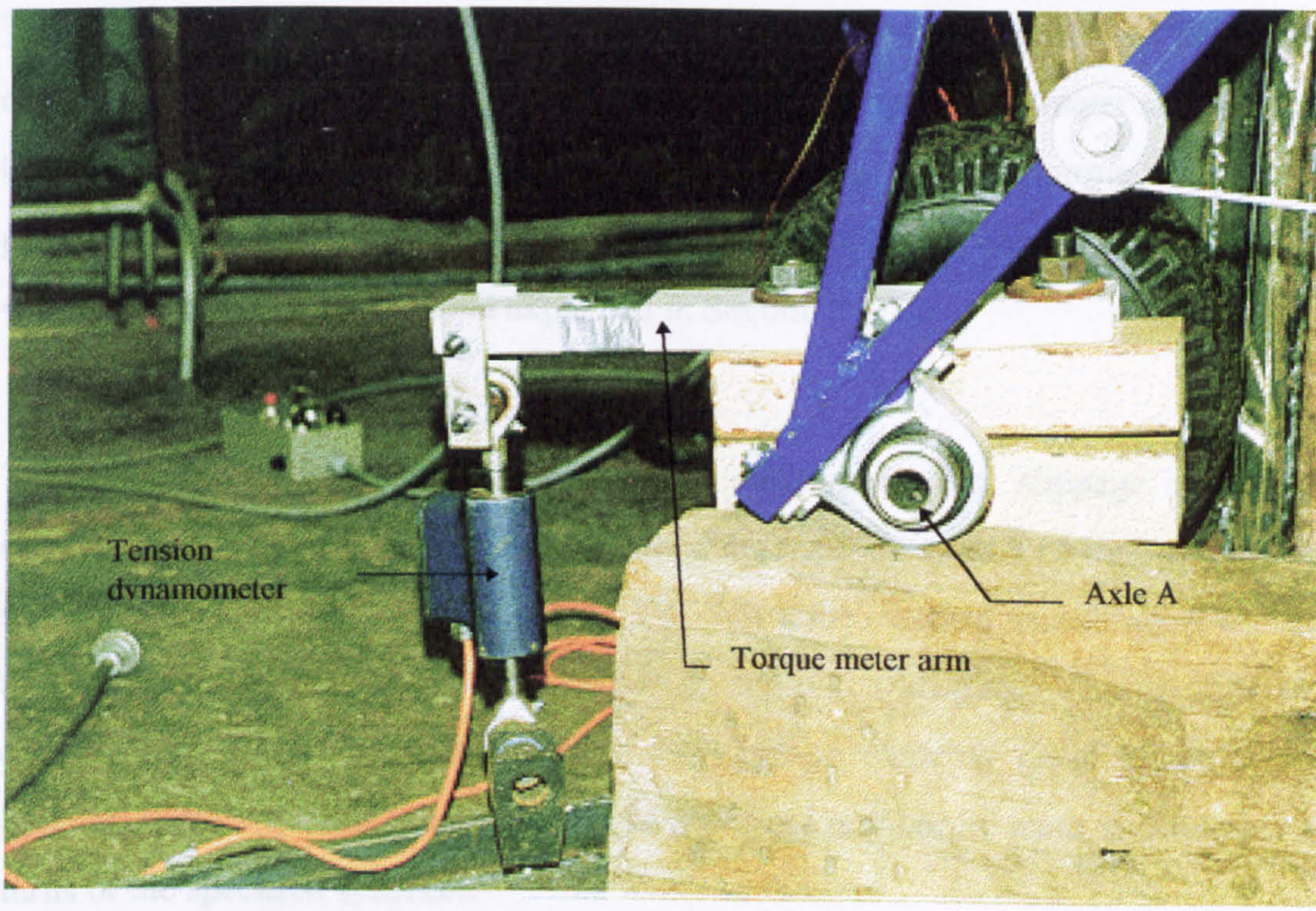


Figure 5-2 The friction torque meter mounted on the driving axle A

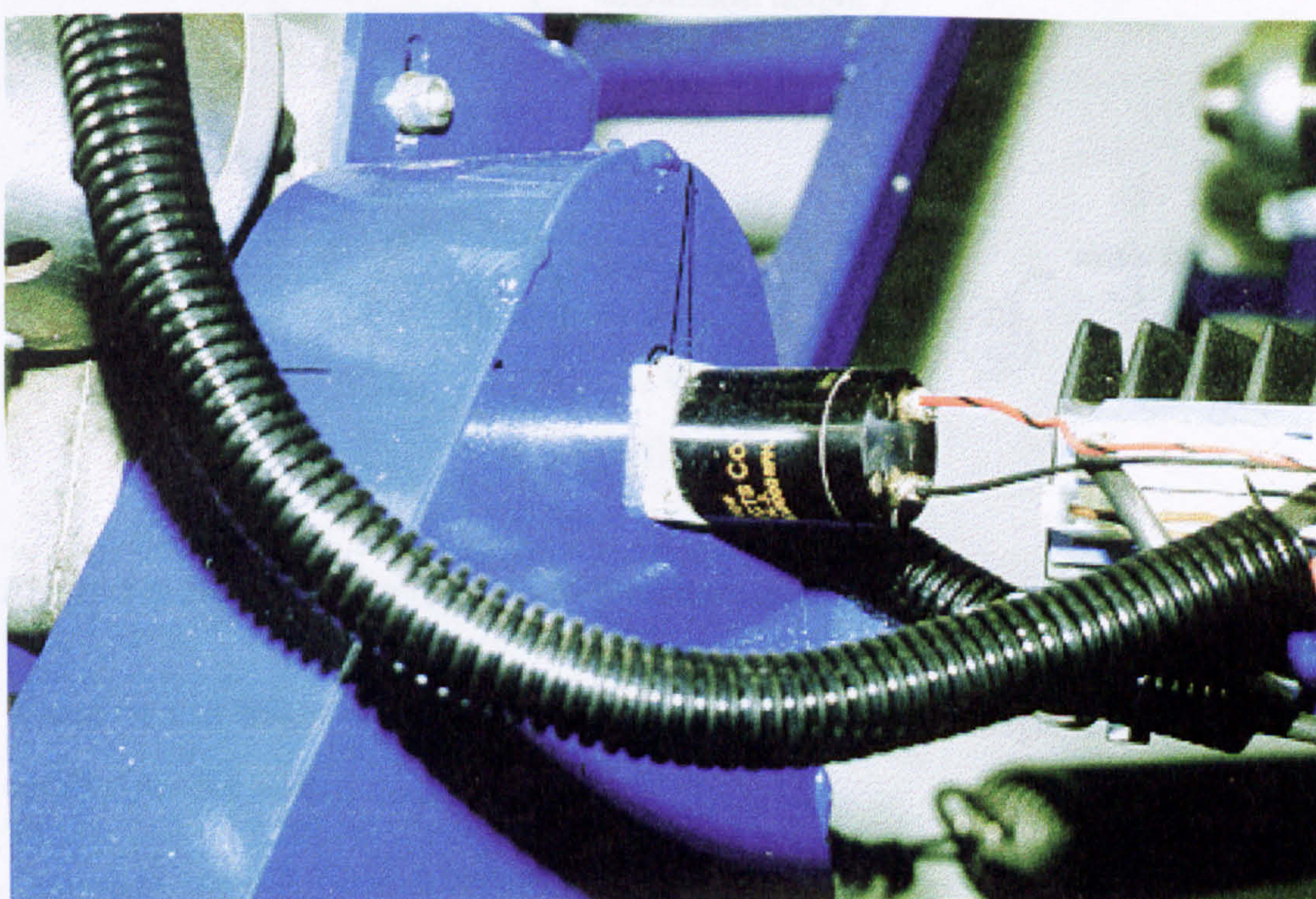


Figure 5-3 speed sensor fixed on the machine

The sensor is connected to the computer to record the force sensed by the sensor during the tests. Multiplying this force by the length of the torque meter arm which is 0.265 m gives the torque produced in the axle. The power and torque curves were produced for different values of motor speeds set through the controller. The sensor was calibrated using the method explained in Section 5.3.1.

5.2.2. Wheel slippage sensor

The objective of fitting the sensor was to determine wheel slippage during the climbing tests. A proximity sensor was fitted near sprocket no. 2 of the transmission system of the axle C as is seen in Figure 5-4. When the machine was climbing the tree an electronic counter showed the number on the digital display. This number was converted to the wheel slippage using the slippage equation given by Innes and Kilgour (1978). In this analysis t and N represent number of teeth and number of turns of the specified sprocket:

$$\text{Slippage} = \left(1 - \frac{\text{actual height}}{\text{theoretical height}}\right) \times 100 \qquad (5.1)$$

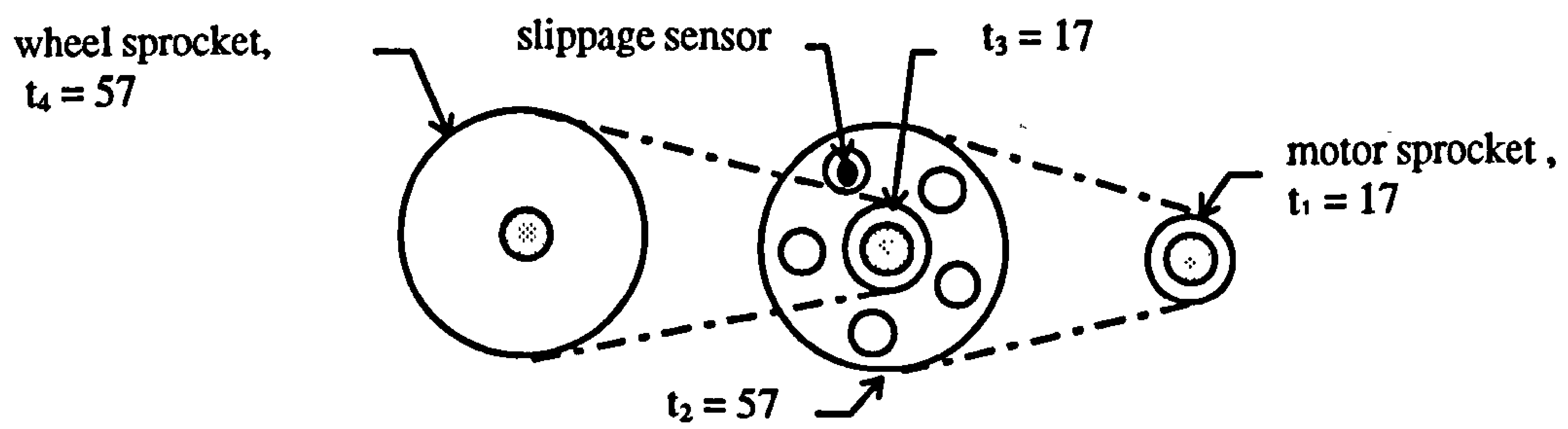


Figure 5-4 Slippage sensor fitted on the chassis near the sprocket 2

Where "actual height" is constant value of 2050 mm, the real height that the machine climbed at each test.

$$\text{Theoretical height} = N4 \times (2\pi r) \qquad (5.2)$$

where r is rolling radius of the tyre. It was measured when the machine was on the tree and was equal to 128 mm.

Substituting r and then Equation 5.2 in Equation 5.1 gives:

$$\text{Slippage} = \left(100 - \frac{254.9}{N_4}\right) \tag{5.3}$$

N4 can be calculated from the sprocket and chain equations (Shigley and Mitchell 1983) and illustration in Figure 5-4 as follows:

$$N_4 = N_3 \times \frac{t_3}{t_4} \tag{5.4}$$

Sprocket 2 and 3 are keyed on one shaft and the hub of the sprocket 2 has five holes. At each turn the sprocket cuts the magnetic field of the sensor five times and the digital counter screen increments five digits therefore:

$$N3 = N2 = \frac{\text{counter number}}{5} \tag{5.5}$$

Putting Equation 5.5 in Equation 5.4, substituting values for t_3 and t_4 and putting the result in Equation 5.3 gives:

$$\text{Slippage} = \left(100 - \frac{4273}{\text{counter number}}\right) \tag{5.6}$$

Equation 5.6 converts the number that the digital counter displayed after climbing each test height to the wheel slippage. It was used in Chapter 6 to calculate the wheel slippage.

5.2.3. Data acquisition system

A "Snap Master" data acquisition system which was installed on a personal computer system was used to record the sensors output. All the data were recorded with the sampling rate of 35 Hz. It was selected because at this frequency the noise from other experimental equipment working in the laboratory did not disturb the system. As each test took about 5 to 12 seconds this frequency gave a reasonable number of data points and because the signals were not oscillating with reference to Nyquist sampling theorem aliasing was not a problem (Bentley 1988).

Four differential analogue channels of the Snap Master kit were used during the tests:

Channel 1: connected to the remote control trigger to start the recording.

Channel 5: connected to the tube tension dynamometer (torque sensor) that recorded the force applied to the arm of the torque meter. These readings were converted to the torque.

Channel 6: connected to the batteries connecting wire which was used to calculate the input electric power (electric power sensor).

Channel 7: connected to the dc generator that recorded the motor speed (speed sensor).

All tests were conducted using the same measurement system. During climbing tests the machine climbed a constant height of the tree (2050 mm). The torque metre sensor and its recording channel were not used for climbing tests. The torque and power curves produced from ground tests were used to calculate the torque in climbing test. During the climbing tests, the measurement system with the help of the computer recorded the instantaneous wheel speed and the input electric power.

To start recording data the "computer remote trigger" on a table by the machine was pushed. The computer started to record the data for a defined number of seconds and stored it in a file in directory c:\sm\userdf with the extension of "smb". An initial number was allocated to the first file in a special window and whenever the trigger was pressed the new file was made and recorded with one increment in the file name. Subsequently, the data are imported to a spread sheet to be analysed.

5.3. Calibrations

5.3.1. Torque sensor calibration

To calibrate the "torque sensor" the tension dynamometer shown in Figure 5-2 was suspended from a crane hook in the laboratory. A weight holder was suspended to the other end of the sensor. The weight of the holder was 1.5 kg. The weight holder was loaded in 20 kg increments to 100 kg.

For each loading the data were recorded for 5 seconds. The average value of the voltage recorded during 5 seconds was taken as the voltage output of the sensor at that loading. The axle torque is the load on the sensor multiply by the torque meter arm which is 0.265 m. Figure 5-5 shows the calibration curve.

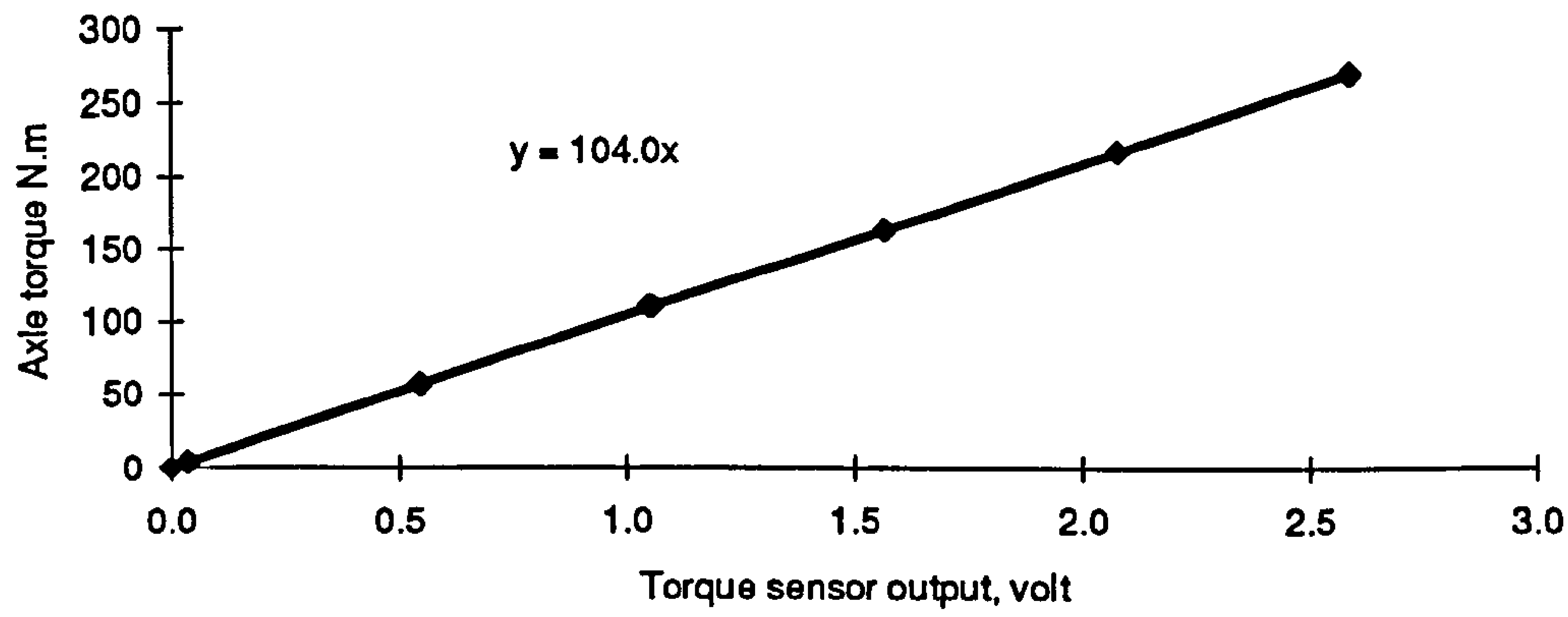


Figure 5-5 Torque meter calibration curve

5.3.2. Speed sensor calibration for axle A

The aim of this calibration was to enable determination of the axle speed (axle A) through the voltage produced by the speed sensor.

An electronic tachometer was used to calibrate the sensor. A narrow strip of white fluorescent tape was attached to shaft of sprocket 1 in Figure 5-4. When holding the record button the tachometer was aimed toward the shaft. It printed the speed of the shaft on its screen after a few seconds. At the same time the computer was set to record signals from the speed sensor for 5 seconds.

The data for the calibration curve were collected during the ground tests period when the machine was in a static position on the ground and the driving axle A was set to rotate freely under no load. The calibration test was conducted with three replications for five level of axle speed named n_2, n_4, n_6, n_8 and n_{10} in the range of zero to maximum down loaded to the motor controller by the programmer. The physical value of these names (n_2, ..., n_{10}), were recorded by the electronic tachometer and computer and are listed in Table 5-1. To draw the calibration curve for axle A average tachometer reading of Table 5-1 were divided by 11.24 which is the reduction ratio of the chain transmission system. The result is calibration curve shown in Figure 5-6.

Table 5-1 Speed of shaft 1 recorded by the electronic tachometer, rev/min

Speed	n_2	n_4	n_6	n_8	n_{10}
Replication 1	95	181	279	367	464
Replication 2	89	190	277	368	430
Replication 3	90	185	285	369	447
Average	91	185	280	368	447

Table 5-2 Speed of shaft 1 recorded from the speed sensor by the computer, volt

Speed	n_2	n_4	n_6	n_8	n_{10}
Replication 1	1.007	1.924	2.932	3.871	4.869
Replication 2	0.946	2.009	2.900	3.872	4.530
Replication 3	0.968	1.947	2.748	3.884	4.684
Average	0.974	1.951	2.860	3.876	4.694

5.3.3. The input electric power sensor calibration

The voltage drop across the wire that connects the batteries was used to measure the input electric power as is shown in Figure 5-1. The input electric power, P, can be calculated from the following equation (Bell & Whitehead 1993):

$$P = IV_m \tag{5.7}$$

where I is the circuit current and V_m is the motor terminal voltage.

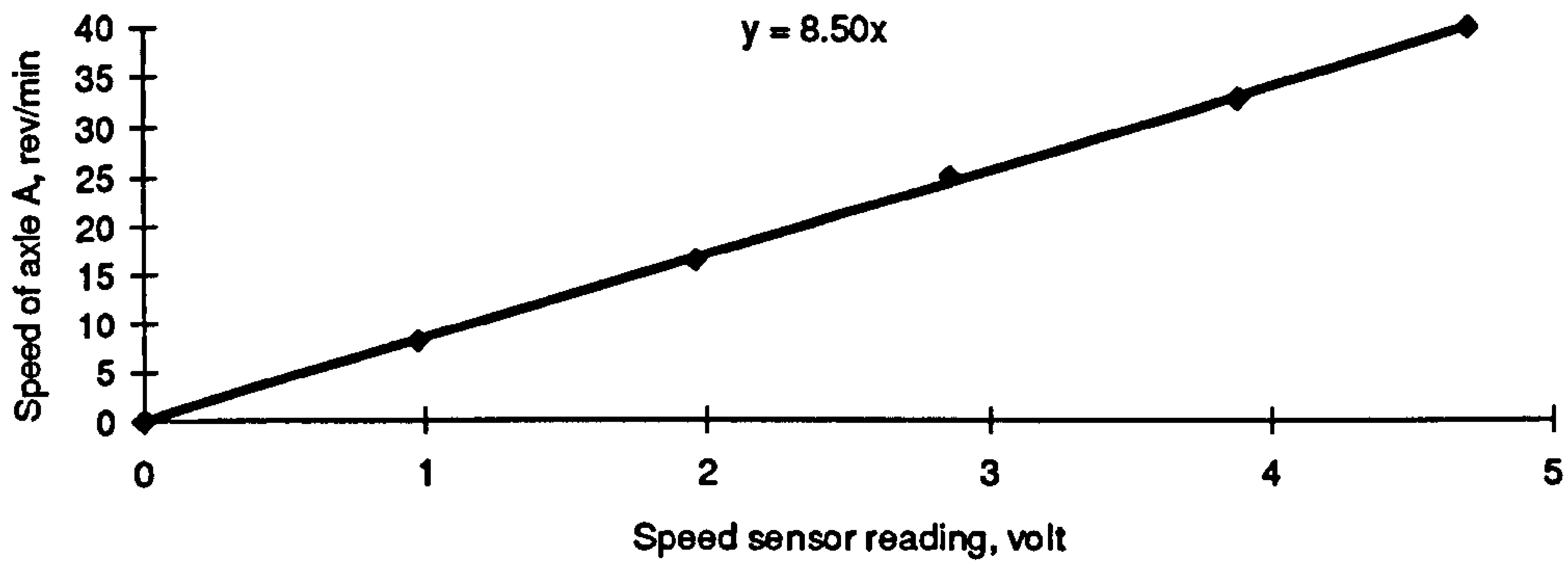


Figure 5-6 Calibration curve for speed sensor

I, the main circuit current in Equation 5.7 can be calculated from the voltage drop across the batteries connecting wire using the following equation:

$$I = \Delta V / R \qquad (5.8)$$

To calculate V_m the method of achieving a controllable motor speed for a permanent magnet motor from a dc source using a simple PLC is shown in Figure 5-7. To control the motor speed the controller (PLC) switches the power ON and OFF through a controlled thyristor with a constant period. Changing the duty-cycle, ratio of ON-time to (ON+OFF)-time, changes the motor terminal voltage, V_m, and as a result motor speed also changes proportional to this ratio (Hindmarsh 1985), therefore, the following equation can be written:

$$V_m / V = n_m / n \qquad (5.9)$$

Where V is the dc source voltage and n is motor speed at this voltage. When duty-cycle is equal to one motor terminal voltage is equal to dc source voltage, V, and motor turn with maximum speed (n). V_m is any voltage lower than the dc source and n_m is the corresponding motor speed.

It is explained in Chapter 6 that climbing tests were conducted with n_6 from Table 5-1, therefore, n_m is 283 rev/min. The dc source voltage, V, is 24 and motor speed at this voltage, n, is n_{10} from Table 5-1 which is 447 rev/min. Substituting these values in Equation 5.8 gives V_m equal to 15 volt.

where ΔV is voltage drop across the wire; R is the wire resistance, 0.0052 ohm.

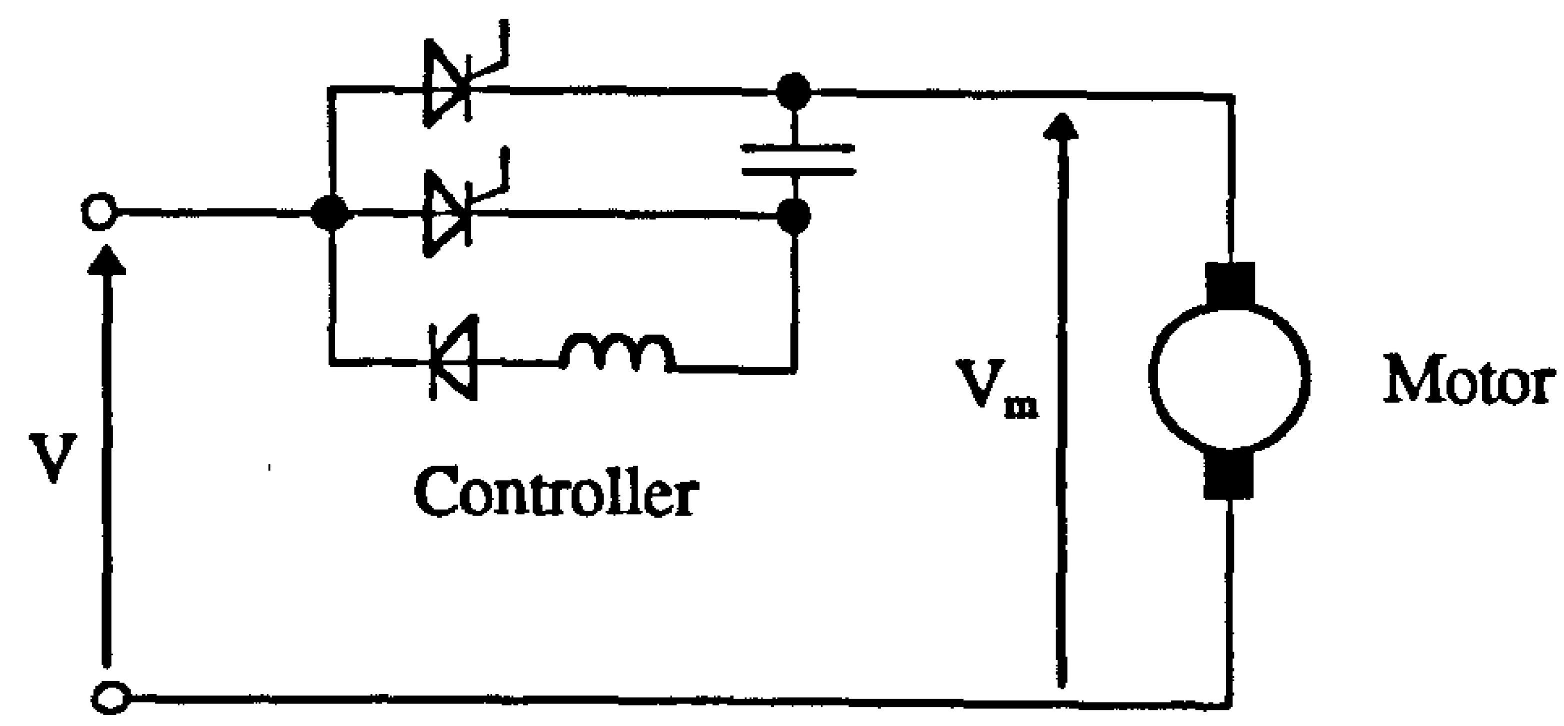


Figure 5-7 Simple motor control circuit

Substituting values of Equation 5.8 and 5.9 in Equation 5.7 gives:

$$P = 2884.6\Delta V \qquad (5.10)$$

Equation 5.10 converts the voltage drop across the two batteries connecting wire (electric power sensor) to the equivalent electric power in watt.

5.4. Programme

The test programme includes all relevant studies which were conducted on the experimental machine.

5.4.1. Machine weight

The test was conducted to determine the machine weight. The machine attached to a balance was lifted by a crane in the laboratory and the balance displayed the machine weight. The total machine weight including the cutter arm, stabilisers and batteries is 150 kg.

The stabilisers which weigh 12.8 kg were removed from the machine during the climbing tests. Therefore, the total weight of the machine during the tests was 137.2 kg.

5.4.2. The machine ground speed

The aim of the test was to determine the maximum ground speed that the machine can achieve on different surfaces. The test was carried out on two different surfaces:

- concrete
- uneven grassy and surface similar to that found in the date palm groves.

A length of 15 meters was marked on the ground and the time that the machine took to travel 15 meters was recorded. The motor controller was set on maximum speed (n_{10}) and measurements were conducted with the maximum speed. Each test was replicated three times. The results in Table 5-3 show little difference in performance.

Table 5-3 Machine ground speed measurement

	t_1 sec	t_2 sec	t_3 sec	t ave. sec	L m	v ave. m/sec
Concrete	26	26.3	26.1	26.1	15	0.57
Grass surface	26.5	27	27.1	26.9	15	0.56

5.4.3. The machine climbing speed

The aim of the test was to determine the maximum climbing speed that the machine achieves without any payload.

The height that the machine climbed during each test, was established by attaching a surveying tape to the machine. The climbed height is the difference between the machine elevation before a run when the machine is near to the ground (h_1) and after the run where the machine is at the top of the tree (h_2). The reference point for these measurements was the ground. Then the climbed height is $h=(h_2-h_1)$.

The time taken by the machine to climb the height h was recorded using a stop watch. Table 5-4 shows the results recorded for the test, which was replicated three times.

5.4.4. The machine payload

The aim of this test was to measure the machine pay load capacity by addition weights to the machine by 20 kg increments and attempting to climb the test tree. The machine was able to carry 100 kg or five blocks of 20 kg weights. To protect motors from overload and extra heat the long period climbing tests were carried out with maximum of 80 kg payload.

Table 5-4 climbing speed measurement

Replication	h_1 m	h_2 m	$h = (h_1 - h_2)$ m	t sec	v = h/t m/sec
1	1.20	3.85	2.65	9.5	0.28
2	1.20	3.68	2.48	9.0	0.28
3	1.21	3.80	2.59	9.8	0.26
Average					0.27

5.4.5. Machine stability and wheel spacing

The aim of the test was to find the safe values of wheel spacing that machine could climb the tree without moving to one side of the tree and becoming immobilised.

Small wheel spacing and high payload tended to pull the machine to one side and affect its stability when it is climbing. Conversely if the wheel spacing was more than a maximum, the axle touched the tree. The test was designed to establish the limits of wheel spacing, with two types of loading:

- Central payload of 100 kg.
- Eccentric payload of 85 kg.

The range of wheel spacing suitable for the machine was determined by experiments, which involved testing all combination of the wheel spacing for pair of wheels on axles A, B and C. There were 4 possible wheel spacing : 200, 300, 400 and 500 mm for each axles and considering three axles there were : $4 \times 4 \times 4=64$ possible combinations of settings. All these setting were tested for central and eccentric payload, therefore, a total number of $64 \times 2= 128$ settings were tested.

The settings that were considered to be safe as the machine did not move to one side of the tree or the axle touch the tree are shown by grey cells in Table 5-5. Wheel spacing less than values shown in grey cells of the table made the machine unstable. It moved to the side and eventually stopped. At wheel spacing more than grey cells the machine axles touch the tree.

Table 5-5 Central payload, 0 to 100 kg and eccentric payload, o to 85 kg

Load type	Axle	Wheel spacing in mm			
Central payload	A	200	300	400	500
	B	200	300	400	500
	C	200	300	400	500
Eccentric payload	A	200	300	400	500
	B	200	300	400	500
	C	200	300	400	500

These results will be discussed more in section 6-2.

5.4.6. Climbing leaf bases with two and one driving axle

The aim of this test was to compare the machine performance employing "two driving axle" and "one driving axle". The theory developed in Section 2.5.4 predicted that the machine with two driving axles is more powerful than a one driving axle machine to climb the steps (leaf bases) and based on that theory the machine was designed with two driving axles. This Section compares the differences between the climbing ability of the two choices and the validity of the theory.

Axle C was selected for one driving axle test because it carry greatest horizontal load between driving axles, therefore, it can produce more traction force. If the machine could be operated with one driving axle it would be:

- simpler to use
- cheaper to run
- easier to make

The objectives of the test were:

1. To determine the machine pay load capacity and performance on a smooth tree surface.
2. To study the machine load capacity and performance on an uneven tree surface simulated by wooden steps attached to the tree.
3. To compare the results with the original theory that suggests that the dual driving axle would improve the machine's ability to climb leaf bases.

To ensure that the motor generated the maximum power the batteries were charged before the test.

The controller (PLC) was set to give the maximum current to the motor and the speed was set on n_6, the speed value which was used for all other tests.

To study objective one, the machine was gradually loaded by increments of 200 N.

To study objective 2, wooden blocks with 20 mm and 41 mm height (the maximum height which the wheel could climb) attached to the tree by nails just in front of the wheels of axle A and C. The ability of the machine to climb then was watched and recorded. The results will be explained in Chapter 6.

5.4.7. Measurement of the coefficient of friction

To compare the climbing efficiency in the laboratory with real condition the coefficient of friction between the wheel and the test tree trunk surface was measured by the method explained for the real tree in Section 3.5. The results can be seen in Table 5-6. Substituting values from the Table in Equation 3.6 gives $\mu = 0.63$.

Table 5-6 Horizontal force needed to pull the rubber on the sample

Sample	A			B			C			Average
Replication	1	2	3	1	2	3	1	2	3	
F_h , N	13	13	14	15	15	14	14	14	13	13.9

This value is smaller than the coefficient of friction between the tyre and real tree, which was determined to be 0.85 in Section 3.5. The machine therefore, should climb the real tree with better performance because of greater gripping capabilities.

5.4.8. Machine harvesting time

The machine harvesting time was measured and the data were used to :

- evaluate the machine harvesting speed
- compare the machine with other harvesting systems
- use the data in economic evaluation of the machine

The operations were recorded on a video tape and the time to perform each task was measured by reviewing the tape.

Preparation time is one of the time consuming steps in mechanised harvesting systems thus for better evaluation steps of this stage were grouped separately. It helped to recognise those actions which needed more time to be improved for the future modification of the machine.

The preparation time concludes two main parts:

- Attachment of the machine to the tree
- Detachment of the machine from the tree

With reference to Figures 2-7 and 2-8 the steps which should be followed to attach the machine to the tree are:

1. separate the "one axle unit" from the stabilisers by loosening the bolts
2. turn the "one axle unit" about the vertical hinge e in the horizontal plane after undoing the horizontal hinge (p) bolt thereby making the machine ready to fit round the tree.

Table 5-7 Times required to attach the machine to the tree

Step or Activity	Time sec.			
	Replication			Average
	1	2	3	
1	17	18	20	18
2	15	14	17	15
3	19	20	18	19
4	15	14	17	15
5	40	38	45	41
6	45	40	48	44
7	10	12	9	10
8	60	50	55	55
Total time sec.	221	206	229	218

3. move towards the tree to surround it by the machine
4. return the "one axle unit" to its position and tighten the hinge bolt
5. connect the springs to the one axle unit
6. load the springs by turning the winch handles
7. move the machine a few cm up to lift the stabilisers from the ground
8. remove the stabilisers

The time recorded for the above steps are shown in Table 5-7 with the "total time" of 3 minutes and 38 seconds.

The following steps should be followed to detach the machine from the tree:

1. fit the stabilisers to the machine. The machine should be in a suitable height so that the stabiliser wheels do not touch the ground when they are being fitted. Otherwise it is more difficult and time consuming to fit them.
2. lower the machine to the ground till wheels of axle A touch the ground
3. unload the springs by rotating the winch handles
4. detach the springs from the one axle unit arms

5. turn the one axle unit when the horizontal hinge p is unbolted
6. move back and leave the tree
7. return the one axle unit to its original position and bolt it
8. Bolt the one axle units to the stabiliser wheels frame. This connection insures a good rigidity for the machine during the transport mode.

Table 5-8 Time required to detach the machine from the tree

Step or Activity	Time sec			
	Replication			
	I	II	III	Average
1	70	78	75	74
2	15	14	17	15
3	24	20	30	24
4	34	30	37	33
5	12	18	14	14
6	15	17	20	17
7	16	17	14	15
8	24	30	27	27
Total time sec	210	224	234	223

The machine harvesting time speed can be calculated by adding up the time that machine requires to complete the sequence of harvesting actions. The harvesting sequence of actions and the recorded time for each action can be seen in Table 5-9. The time to move to the tree from the previous tree and climbing the tree were calculated assuming 5.8 m average tree spacing and 10.3 m average tree height from Table 3-8. The machine ground speed was assumed 0.56 m/sec from section 5.4.2 and the machine climbing speed 0.27 m/sec from section 5.4.3. It was assumed that the climbing time is equal to the time to come back to the ground.

Row 11 of the Table is the time to cut a date bunch. Because the harvester arm moving system and cutting device mechanisms were not manufactured for the test rig, this time was calculated based on the selected concepts, L1, M2 and N3 from the morphological chart (Figure 2-4). The time was calculated using the speed of an electric actuator made by LINAK Company which has a speed of 50 mm/sec and the force of 200 N enough to move the harvester arm system and cut the date stalks. The harvester arm and the cutting device (concepts L1 and M2) length of travel to move from one date bunch to another and cut it was measured from the test machine and the drawings.

Table 5-9 Sequence of harvesting steps and the consumed time to do them

No.	Action	Definition	Time sec	% of the total time
1	t_1	Move to the tree from the previous tree	10	0.7
2	t_2	Attach the machine to the tree	218	15
3	t_3	Climb the tree	38	2.7
4	t_4	Harvest 4 bunches of dates	4×37	10.4
5	t_5	Come back to the ground	38	2.7
6	t_6	Unload 4 bunches of dates	4×7	1.9
7	t_7	Detach the machine from the tree	227	16
8	t_8	Move to the other side of the tree	30	2
9	t_9	Attach the machine to the tree	218	15
10	t_{10}	Climb the tree	38	2.7
11	t_{11}	Harvest 4 bunches of dates	4×37	10.4
12	t_{12}	Come back to the ground	38	2.7
13	t_{13}	Unload 4 bunches from the machine	4×7	1.9
14	t_{14}	Detach the machine from the tree	227	16
15	t_t	Total harvesting time	1438	100

The results show that the attachment and detachment of the machine from the tree consumes a significant part of the total harvesting time. To discuss the result of Table 5-9 in Chapter 6 and find methods of improving the attachment, detachment and climbing times results of the table were put in main groups as shown in Table 5-10.

Table 5-10 Total time required for completion of main harvesting steps

No.	Action	Definition	Time sec	% of the total time
1	t_t	Total harvesting time	1438	100
2	t_{ad}	Total attachment and detachment time = $(t_2 + t_7 + t_9 + t_{14})$	891	62
3	t_b	Total bunch harvesting time = $(t_{11} + t_4)$	302	21
4	t_c	Total climbing time = $(t_3 + t_5 + t_{10} + t_{12})$	152	10.6

The figure shows that the total harvesting time is 1438 seconds or 24 minutes and the attachment to the tree and the detachment from the tree (preparation time) occupy more than 62 % of the total harvesting time. Preparation time improvement is discussed in Chapter 6.

5.4.9. Ground tests

"Ground tests" means carrying out the tests when the machine was stationary and was not climbing the tree.

The primary aim of the ground tests was to generate the motor and transmission system characteristic curves, a series of curves that relate the voltage drop across a wire (electric power sensor) to the torque and speed of driving axle A. These curves were used to predict the torque in axle A during the power tests. This method was easier than direct instrumentation and measurement of the axle torque during the time that the machine was climbing the tree.

The secondary aim was to find the optimum climbing speed that the motor could produce required torque to carry maximum payloads to provide a safe climbing test situation without bouncing or diverting to one side of the tree.

During these tests the machine was clamped to the test tree at ground level. Only motor A and its transmission system were used during the tests, motor C was disconnected form the power source. The objective was to apply different torque to axle A and record the voltage drop of the wire and revolution of the axle A simultaneously.

By placing a wooden block between the main chassis and the tree and two wooden blocks on the ground under the axle bearings wheels of axle A had no contact with either the tree or the ground. As a result it could rotate freely , whilst the whole machine was kept stationary. The torque meter explained in Section 5.2.1 was connected to the axle A to apply torque to it. The measurement system including the data acquisition system, speed sensor and electric power sensor also were used for these tests.

To minimise the test errors such as heat produced in the torque meter and level of charge in the battery the ground tests had a completely randomised design (Basiri 1993). The test was conducted with five level of motor speeds n_2, n_4, n_6, n_8 and n_{10} down loaded to the controller where n_2 is the minimum and n_{10} is the maximum speed. For each speed level up to 9 level of torque were applied by tightening the torque meter nuts. Each test was carried out with three random replications. Table 5-11 shows the test design and randomised table.

Table 5-11 Test sequence and the randomised table for the ground tests

Test no.	1	2	3	4	5	6	7	8
Speed setting	n_{10}	n_4	n_8	n_2	n_6	n_{10}	n_2	n_4
Replication	1	1	3	3	1	3	2	3
Test no.	9	10	11	12	13	14	15	
Speed setting	n_8	n_8	n_{10}	n_6	n_2	n_4	n_6	
Replication	1	2	2	2	1	2	3	

5.4.9.1. Ground test method

The applied method is defined as a sequence of steps as follows:

1. Switch the controller on from the remote control pad.
2. Set the speed of the motor A at the required level. To set the speed, plug the programmer to the controller enter the set-up menu and press the change button to achieve the desired speed (this should be done when the controller is switch on). The sequence of selecting the speed must be followed from the Table 5-11.
3. Switch the computer on and run the MS windows

4. Run the Snap Master
5. Open the "tree.ins" file which was created for the ground tests
6. Click the start button on the computer screen. Now the Snap Master is ready to start recording data whenever the trigger near the machine is pressed.
7. Undo the torque metre nuts to release the torque completely. It is zero torque or maximum speed condition.
8. Turn the speed potentiometer on the control pad to the maximum (should be on maximum for all of the tests). The motor starts to rotate with the maximum speed
9. Press the "computer remote trigger" (refer to Figure 5-1) to record the data for maximum speed.
10. Tighten the nuts a few turns to create a new level of the torque.
11. Press the trigger to record the data for this level
12. Repeat steps 10 and 11 until the shaft stops under the load.

To have reasonable amount of data points for each curve the torque meter nuts should be tightened gradually to produce about nine level of torque from zero to the shaft stopping value.

For each torque level the data were automatically recorded for 5 seconds on the hard disk in the control for further analysis of the results. The ground test data analysis method is explained in Appendix C and the results are presented and discussed in Chapter 6.

5.4.10. Slippage tests

One of the main features of a good climbing system is climbing the tree within a reasonable range of driving wheels slippage. Two systems selected from morphological chart (Figure 2-4) control the wheel slippage which are horizontal plane stability system (concept C1) and traction control system (concept L1). The payload also affects the wheel slippage. The slippage therefore was selected to evaluate the selected concepts C1 and L1.

Two wheels on each axle of horizontal plane stability system are not vertical to the tree which might cause extra slippage. The concept was checked for extra slippage by the slippage test. The wheel spacing $D_1 = 300$ mm and $D_2 = 350$ mm were selected for the test. D_1 was calculated from Equation 2.23:

$$\text{Wheel spacing } \geq \sin \tan^{-1}\mu \times \text{tree diameter} \qquad (2.23)$$

The coefficient of friction between the test tree and the tyre was determined 0.63 in Section 5.4.8 and the test tree diameter was 520 mm. Substituting values in equation gives $D_1 \geq 277$ mm and the nearest wheel spacing setting on the axle shaft was 300 mm, therefore D_1 was selected 300 mm and D_2 was set on 350 mm which was the next spacing on the shaft.

Two spring tensions T_1 and T_2 were selected to evaluate the traction control system by slippage test. To select T_1 the wheel spacing D_2 was set and the machine was loaded 80 kg, the maximum payload for the test, then the spring was gradually loaded to the value that machine did not have obvious slippage when was climbing the tree. To see the significant effect of spring tension on the slippage, T_2 was set on the maximum value which was possible to create on the machine because minimum slippage will occur with this high tension. The spring length was measured at T_1 and T_2 and then using the followings equation given by the spring manufacturer the spring tension was calculated.

$$T = (\text{ spring rating} \times \text{deflection }) + \text{initial loading} \qquad (5.11)$$

$$\text{deflection} = \text{spring length - initial length} \qquad (5.12)$$

Where T is the spring tension.

The spring length was determined 940 mm at T_1 and 1040 mm at T_2. From the spring specification, Section 4.8, spring rating is 0.84 N/mm , spring initial length is 377 mm and spring initial loading is 82.3 N. Substituting these values in Equations 5.11 and 5.12 give:

$$T_1 = 555 \text{ N}$$

$$T_2 = 639 \text{ N}$$

The slippage for above settings of wheel spacing and spring tension was measured at different payloads because the payload has also affects on the slippage. The study was

completed by drawing the slippage and power curves at different payloads. To have enough data points for the curves, the tests were conducted with 5 different payloads $L_1 = 0$, $L_2 = 20$ kg, $L_3 = 40$ kg, $L_4 = 60$ kg and $L_5 = 80$ kg. For low tension, T_1 , because of the safety matter 80 kg payload was not applied, as the uneven surface of the tree at the top diverted the machine to one side of the tree at this loading.
To discuss the slippage and power curves, or the machine performance including the horizontal plane stability system at wheel spacing D1 and D2, the traction system at spring tensions T1 and T2 and both systems at payloads L1 to L5 a "randomised complete block design" statistical analysis model with factor T (spring tension) split on D (wheel spacing) and Factor L (payload) split on T was used (Basiri 1993). The advantage of the model is that it includes a comprehensive test which the results show the effect of each factor separately and also it also show the interaction effect of factors with each other. The MSTATC statistical analysis software was used to analyse the test data. The randomised test design can be seen in Table 5-12.

The randomisation was started with wheel spacing then for each wheel spacing the spring tension was randomised in each replication and then the payload was randomised for each tension. The test began with the first replication of wheel spacing D_1. Wheel spacing D_2 began when the third replication of D_1 finished. This is standard method to randomise the model (Basiri 1993).
This test design model was also used for the power tests. At the same time that the slippage was measured the computer was switched on and recorded the signals from the speed sensor and the electric power sensor. The method of conducting the test is explained in the next Section.

Table 5-12 Randomised table for slippage and power tests; D is wheel spacing in mm; T is spring tension in N; Numbers are payload, L, in kg

	Replication					
	1		2		3	
D_1	T_1	60	T_2	80	T_2	0
		20		60		40
		40		0		20
		0		20		60
	T_2	80		40		80
		20	T_1	20	T_1	40
		60		60		0
		0		0		60
		40		40		20
	Replication					
	1		2		3	
D_2	T_1	0	T_1	20	T_2	80
		40		60		0
		60		0		40
		20		40		60
	T_2	20	T_2	40		20
		60		80	T_1	40
		0		20		60
		40		0		0
		80		60		20

5.4.10.1. Slippage and power test method

The method of conducting slippage and power tests is explained as a series of steps. The test was carried out according to the sequence in Table 5-12, explained in previous Section, therefore the first setting from the Table are D_1 , T_1 , and 60 kg payload. Setting of D_2 were applied after finishing tests of wheel spacing D_1. To run the test for the first setting:

1. Set the wheel spacing of axle C on D_1 to 300 mm.
2. Set the spring tension on T_1.
3. Switch on the controllers on the remote control pad.
4. Set both controllers on n_6 for forward speed and n_3 for reverse speed. Keep this setting for all of the slippage and power tests.
5. Switch on the computer
6. Run MS windows
7. Run the Snap Master
8. Open the file "Tree1.ins" which has been created for slippage and power test.
9. Click on the start button on the screen. The Snap Master is now ready to record the data from the machine sensors, once the trigger on the table near the machine is pressed.
10. Put the payload on the machine.
11. Switch on the slippage sensor counter of wheels C and write the counter number.
12. Push the computer remote trigger to start recording signals of input electric power sensor and speed sensor by the computer (the computer records the data for the climbing period and then automatically stops).
13. Start climbing the tree by turning the speed potentiometer on the control pad to the maximum. The machine should be correctly positioned, axle A being level with the starting point which is marked as a white line on the tree
14. stop the machine when axle B reaches the white mark on the top of the tree. The height of travel is 2050 mm.
15. Switch off the slippage sensor counter that has recorded the number of the revolutions of wheel C during the climbing period.
16. Put the machine into reverse by switches on the remote control pad.
17. Land the machine
18. Stop the machine at the starting line by returning the potentiometers to zero.
19. Write down the slippage sensor counter number.
20. Repeat the above steps 10 to 19 for other payloads and settings.

The method of data analysis of the slippage and power test is explained in appendix B and results are explained and discussed in Chapter 6.

CHAPTER 6

MACHINE PERFORMANCE

6. Machine Performance

In this Chapter the results of the test programme detailed in Section 5.4 are discussed and developed. Figure 6-1 shows the experimental machine on top of the tree. The operator controls the machine from the ground by a wire attached to the remote control pad.

6.1. Machine weight, modifications and speed

The experimental machine weight test was explained in Section 5.4.1. The weight excluding the batteries (20.2 kg) is 132.8 kg which is 15% more than 115.2 kg, the value calculated in Section 4.3. This extra weight is because of using a tube 2 mm thicker than the one in the design for the axles as the designed size was not available at the time of manufacture. The sprocket and chain system which was used was heavier than the design because it came from a different manufacturer. The rest of the weight variation is because of chain and sprocket guards, bolts, measurement equipment and other small modifications such as battery frames and stabiliser caster wheel frames.

The machine lifting capability was not known at the early stage of the design, therefore, light weight was a design criterion. In order to reduce the machine weight concept W2 or batteries "on the ground" was selected for the machine (Figure 2-4). The climbing tests showed that the machine has a considerable potential to lift weights, so, the batteries were fitted on the machine, in so doing concept W2 was replaced by W1 which is batteries "on the machine".

Tests were carried out with batteries "on the ground" to investigate the feasibility of using long wires attached to a DC power sources placed on the ground (row V of morphological chart)

Figure 6-1 The experimental machine on top of the tree carrying a 20 kg pay load

The results showed that battery wires do not cause problems and it is feasible to use a generator or other power source on the ground to power the machine but, the experimental machine or real machine with batteries on board is easier to use, especially if it is used for periods of less than one hour, which is the battery running time.

For stabilisers "fixed wheels" , concept Q1 of the morphological chart (Figure 4-4) were replaced with "caster wheels", concept Q2, because it was difficult to steer the machine on the ground. Two eccentric pivot frames were made for the stabiliser wheels to perform this task. To increase the payload capacity the modified stabilisers (12.8 kg weight) were removed during all the climbing tests, so that, the total machine weight during the climbing tests was reduced to 137.2 kg which was 19 % heavier than the design value. The climbing tests showed that the machine can lift a considerable payload and the 19% extra weight did not affect the machine's climbing performance.

The measured machine ground speed (Section 5.4.2) was 0.56 m/sec. In hand harvesting method the worker has an average speed of 0.73 m/sec (this is calculated using an average spacing of 5.2 between rows and trees in rows, from Appendix A, divided by 7.1 sec which is the average time to move from one tree to the next from Table 3-7). This is a reasonable speed for travelling between the trees during harvesting and considering the total harvesting time of 20.5 per tree (Table 3-7) ground speed alone is not a major factor in harvesting speed (0.6 % of the total time). The machine wheels had very strong torque and could easily overcome obstacles on uneven ground surfaces, so, they could perform satisfactorily in the date groves. The sprocket on the driving axle A was a cause of concern as it occasionally touched the ground when it was uneven. To increase the clearance and overcome the limitation a smaller sprocket should be used for the future design, but the reduction ratio should be kept constant.

The maximum climbing speed of the experimental machine (Section 5.4.3) was 0.27 m/sec which compares poorly with 0.41 m/sec, the original design speed (Section 4.6). This reduction can be explained by extra weight, wheel slippage, and a reduction in the design value of wheel diameter due to the tyre deflection under load and movement on the curved tree surface. The speed, however, is close to the average climbing speed of a worker during manual harvesting which is 0.31 m/sec (Appendix A). The machine climbs more slowly than the worker but the climbing speed is adequate and slower speeds were set for the climbing tests because of instability, bouncing and safety considerations. The measured climbing time constitutes only 11 % of the total harvesting time (Table 5-9), so, it is not a major factor in total harvesting speed.

6.2. Machine payload and stability

Machine payload tests showed that the machine can lift 100 kg. The design lifting payload was 50 kg but as indicated in Section 4.6, the power was designed for climbing large leaf bases and the motors selected in Section 4.6.1 could produce a power higher than the required power to lift 50 kg, so, the machine could lift 100 kg payload. The important element of this machine performance is the ratio of machine weight (137.2 kg) to the payload (100 kg) which is 1.4. Table 2-2 shows that this ratio is very low compare to ratios of 20 to 80 for existing harvesting systems. It also shows that the machine weight is 2 to 4% of the existing systems. Comparison shows that the proposed machine can offer a significant saving on the energy consumption and construction costs.

The minimum wheel spacing theory (Equation 2.23) which provides the machine stability in the horizontal plane (concept C1 of the morphological chart) can be validated by stability test results presented in Table 5-5. The experimental tree diameter was 520 mm and the coefficient of friction between the trunk and tyre was 0.63 (Section 5.4.8). Rewriting Equation 2.23 and substituting these values in it gives:

$$\text{Wheel spacing} \geq \sin \tan^{-1}\mu \times \text{tree diameter}$$

or

$$\text{Wheel spacing} > 277 \text{ mm}$$

The equation suggests that wheel spacing should be more than 277 mm for self guidance, which was defined as following the tree trunk without moving to one side. The grey cells in Table 5-5 show the stable setting for concentric and eccentric loading. It shows that for central loading the machine is stable and self guided when axles A and B have a spacing more than 277 mm. In central loading axles A and B alone are enough to control the machine and axle C can have a 200 mm settings which is smaller than 277 mm. This is because the tendency for the machine to deviate is at a minimum in the central loading condition.

For eccentric loading, which is the simulation of reality when one side of the basket is loaded with dates, all three axles were required to have wheel spacing of more than 277 mm.

The test results prove the theory of minimum wheel spacing. The minimum wheel spacing can therefore be predicted using Equation 2.23. The maximum wheel spacing limits can be predicted by clearance requirements between the axle and the tree. For the driving wheels the maximum wheel spacing must also be checked for extra wheel slippage as explained in Section 5.4.10.

The machine was unstable and moved to one side of the tree when it climbed a bend or big knot on the test tree, especially when the wheel spacing was smaller than the stable settings (Table 5-5). To improve the system in these conditions the addition of simple shoes like the ones explained for concept C3 of the morphological chart is suggested. The wheel spacing system draws the machine back to the right path when it moves past a bend or uneven section of the tree.

Another result from the stability test is the evaluation of concept C2 of the morphological chart (Figure 2-4) where one wheel is used on axle C to reduce the machine complexity. The stability settings of Table 5-5 also show that this concept is rejected because the axle requires two wheels with a considerable wheel spacing, a minimum of 200 mm for central loading and 400 mm for eccentric loading. The machine cannot be controlled with one wheel on axle C (zero wheel spacing).

In summary the selected concept C1 with its wheel spacing adjustment system can effectively control machine stability. It will be checked for acceptable wheel slippage value in Section 5.4.10.

6.2.1. Tree safety

The machine applies loads to the tree surface through its wheels and also applies its total weight as an eccentric load to the tree. These loads produce stress levels in different parts of the tree. To check the tree safety against these stresses the maximum value of the stresses were calculated for the original design by the method explained in Section 4.5. The same calculations were repeated for the new experimental machine weight and centre of gravity. The experimental machine's new weight for the climbing tests, including 100 kg payload, was 237.2 kg. The position of the centre of gravity at this loading was found by suspending the machine from the crane and changing the suspension point to establish the point at which the machine stays in a level condition. This was found to be along a vertical line that passed through the centre of axle A, where the length of the centre of gravity, x , is approximately 300 mm (refer to Figure 2-9). Table 6-1 shows the result of the calculations.

Table 6-1 Safety factors of stresses applied to the tree from the experimental machine

Working stress			Failure stress			Safety factor (S)
Sign	Calculation source	Value, MPa	Sign	Calculation source	Value, MPa	
σ_r	Equation 3.1	0.17	$\sigma_{r\,max}$	Table 3-9	2.96	17
τ_L	Equation 3.2	0.14	$\tau_{L\,max}$	Table 3-9	1.00	7
σ_c	Equation 3.3	0.30	$\sigma_{c\,max}$	Table 3-9	5.33	18
σ_t	Equation 3.4	0.23	$\sigma_{t\,max}$	Table 3-9	60.10	261

The main conclusion is that the tree trunk can tolerate the machine stresses with a very high safety factor, so, the machine will not cause any physical damage to the tree during harvesting operations.

6.3. Driving axles and climbing leaf bases

The theory developed in Section 2.5.4. predicted that the machine requires two driving axles to climb the large leaf bases on the tree surface. Based on this theory the experimental machine was made with two driving axles, A and C. Figure 6-2 shows the driving wheel of axle C climbing the test tree leaf base simulated by a wooden step. The driving axle test explained in Section 5.4.6 was conducted to validated this theory. The summary of the results is shown in Table 6-2.

Table 6-2 Maximum payload capacity of the machine in kg for one and two driving axles

Surface condition	spring tension	one driving axle (C) payload, kg	two driving axles (A & C) payload, kg
Smooth	T_1 and T_2	80	100
20 mm block in front of wheels C	T_1 and T_2	40	100
20 mm block in front of wheels A	T_2	40	100
41 mm block in front of wheels C	T_2	40	100
41 mm block in front of wheels A	T_2	20	100
20 mm block in front of wheels A and C	T_2	20	100

Figure 6-2 Wheel of axle C climbing a leaf base simulated by a wooden step

Table 6-2 shows that the climber which has enough torque in each driving axle to climb the leaf bases, simulated by wooden steps, cannot use its full power to climb big steps when just one axle is driving so the wheels start to spin excessively and the machine fails to climb.

Table 6-2 shows that the machine can climb over a wooden step with a maximum height of 41 mm (6 mm more than the maximum value caused by leaf bases on the tree) in front of axle A or C with a payload of 100 kg. For one driving axle the following results can be seen:

- The payload capacity of the machine decreases from 100 kg to 80 kg on the smooth tree surface.

- The payload capacity decreases from 100 kg to 40 kg when there is a 20 mm height step in front of axle A or C.

- The payload capacity decreases from 100 kg to 40 kg when a 41 mm height step is in front of wheels C, and to 20 kg when it is in front of wheels A. It shows that the driving axle can deal with steps in front of its wheel better than other axle wheels which are not driven.

In summary the machine can lift 100 kg over a 41 mm wooden step with two driving axles, but, with one driving axle the pay load capacity reduces to 20 kg and the machine only climbs steps with a maximum height of 20 mm. Therefore utilising one driving axle for climbing on leaf bases decreases the payload capacity by 80% and the step height climbing by 50 %.

It is important to mention that, with all of the one driving axle tests, at the point where the height of the wooden step was greater than the maximum values of 20 mm or 41 mm according to the test, the machine failed to climb and wheels started to spin excessively. This suggests that the machine has the power to climb over the step but, as was predicted in the theory "one driving axle" cannot develop the required traction conditions to use all of the available power of the machine and on steps the climbing ability decreases with one driving axle. It was also noted that the machine can climb over larger steps when only one wheel of an axle faced a wooden step. This suggests that the machine can climb over steps bigger than 41 mm in real conditions because it seldom happens that both wheels face a leaf base simultaneously.

The other important results are that the selected traction control system (concept H3 of the morphological chart in Figure 2-4) has a significant effect on improving the step climbing ability and, as is seen in Table 6-2 increasing the spring tension from T1 (555 N) to T2 (639 N), which pushes the wheels harder on the tree surface, increased the step climbing ability of the machine from 20 mm to 41 mm.

The tyre section height was found to be important for the smooth work of the machine on the steps. Therefore the low pressure, high section height tyres are recommended for the climbing machine.

The main conclusion is that the theory of two driving axles and the relevant equations developed in Section 2.5.4 are valid for the design. Therefore, to climb the leaf bases successfully the design power should be divided between at least two driving axles. The experiment machine with two driving axles could climb over 41 mm height leaf bases simulated by wooden steps, where the maximum absolute value on the actual tree is 34 mm from Section 4.1 paragraph 4, therefore, the machine should climb the leaf bases of the real tree successfully.

6.4. Harvesting efficiency

The total average machine harvesting time was measured (Section 5.4.6) and determined to be 1438 seconds (24 minutes). Given a field efficiency of 75% which is the average value for a farm machine (Witney 1988), the real time is 1917 seconds (32 minutes). To evaluate the harvesting efficiency the harvesting time of the proposed machine was compared with the target time which was set as the average hand harvesting time in Iran (explained in Section 3.4). It was also compared with mechanised harvesting time in Saudi Arabia in the years 1989 and 1990 reported by Al-Suhaibani et al (1992) and the mechanised method in the USA (Perkins and Brown 1964). The comparison can be seen in Figure 6-3. It is obvious from the charts that the total average harvesting time for the experimental machine developed during this study is longer than that required for hand harvesting in Iran, which is the main criteria, and for mechanised harvesting. It should be explained that the deviation between harvesting time in 1989 and 1990 in Saudi Arabia where the cultivation methods are similar to Iran is because the machine become stuck in the irrigation channels and did not have enough room to manoeuvre in the traditional cultivated date grove in 1990. It should be also mentioned that the high harvesting speed in the USA is because of the wide tree spacing and new irrigation methods which allow the machine to travel fast between tree rows, and moreover, the harvesting system employed there has a double action mechanism with two operators: when one operator is cutting date bunches the other one lowers bunches and shakes them in a special container to detach the dates, which improves the harvesting time.

The next step is to establish the potential areas for improvement in the design to reduce the total harvesting time. To identify the most time consuming processes contributing to the total harvesting time achieved by the machine the data in Table 5-9 were used to produce Figure 6-4 where the harvesting steps are divided into main groups. The figure shows that the main area for improvement is the mode change, from transport to climbing, and from climbing to transport, or transformation time. Table 5-6 and Table 5-7 show different steps for attachment and detachment of the machine to and from the tree. In Table 5-6 the attachment time is 218 seconds. The total for steps 1, 2, 4, 5 and 8 is 154 seconds. Most of the time is consumed fitting fastening and loosening bolts to join or separate various mechanisms. This time could be reduced to 10 seconds for each step if the selected concept of bolt and nuts (concept J1 in morphological chart, Figure 2-4) is replaced with a quick fitting latch, J2, which was suggested for the one axle unit locking system. 10 seconds for each step is a reasonable over-estimate because these actions are like opening and closing a car or house door which does not take more than 10 seconds. This modification will reduce the 154 seconds to 50 and the attachment time will reduce by 104 seconds, so, the new attachment time will be 115 seconds. Figure 6-5 and Figure 6-6 show the process of attaching the experimental machine to the tree.

The same improvement can be applied to the detachment time which is 223 seconds (Table 5-7). The time currently consumed for steps 1, 4, 5, 7, and 8 of the detachment procedure is mostly spent on tightening and undoing bolt and nuts. Using the quick fitting latches the time for each step could potentially be reduced to 10 seconds. The total time of 165 seconds for the five step could be reduced to 50 seconds. This would reduce the detachment time by 115 seconds. The improved detachment time would then be 108 seconds. Putting the new times for attachment and detachment in Table 5-8 the new total harvesting time would be 989 seconds (16.5 minutes) and given 75% field efficiency the real time would be 1319 seconds (22 minutes).

Figure 6-3 shows that the improved harvesting time is 5 minutes (18%) shorter than the hand harvesting method in Iran and 1.4 minutes (6%) shorter than mechanised method in Saudi Arabia (23.4 minutes, the average of two years) and from the worker point of view it would be more efficient than the mechanised harvesting method employed in the USA because two men are required for the harvesting operations whilst the proposed machine requires only one operator.

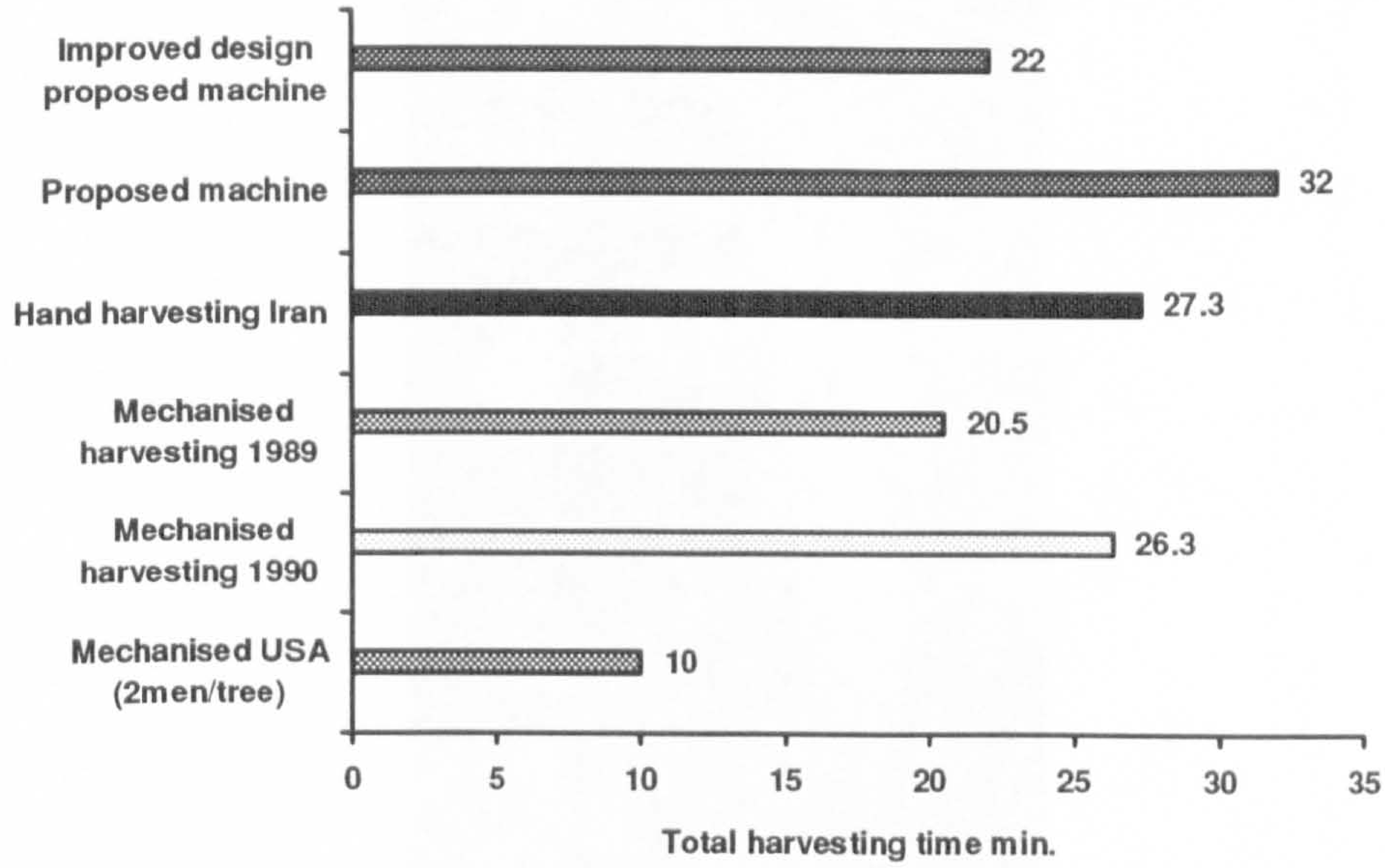


Figure 6-3 Comparison between proposed machine harvesting time and other available hand harvesting and mechanised methods

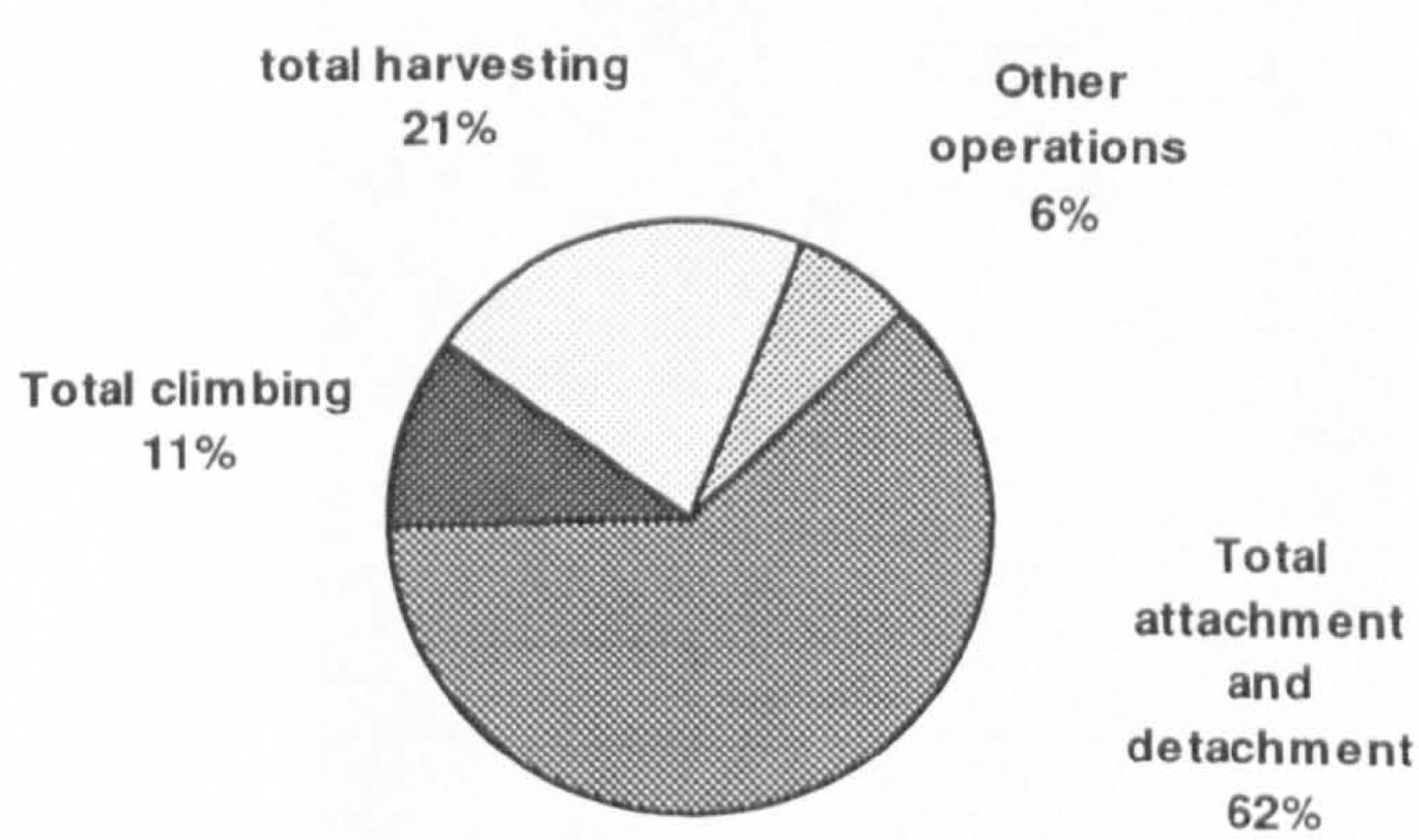


Figure 6-4 Important harvesting step times as a percentage of the total harvesting time.

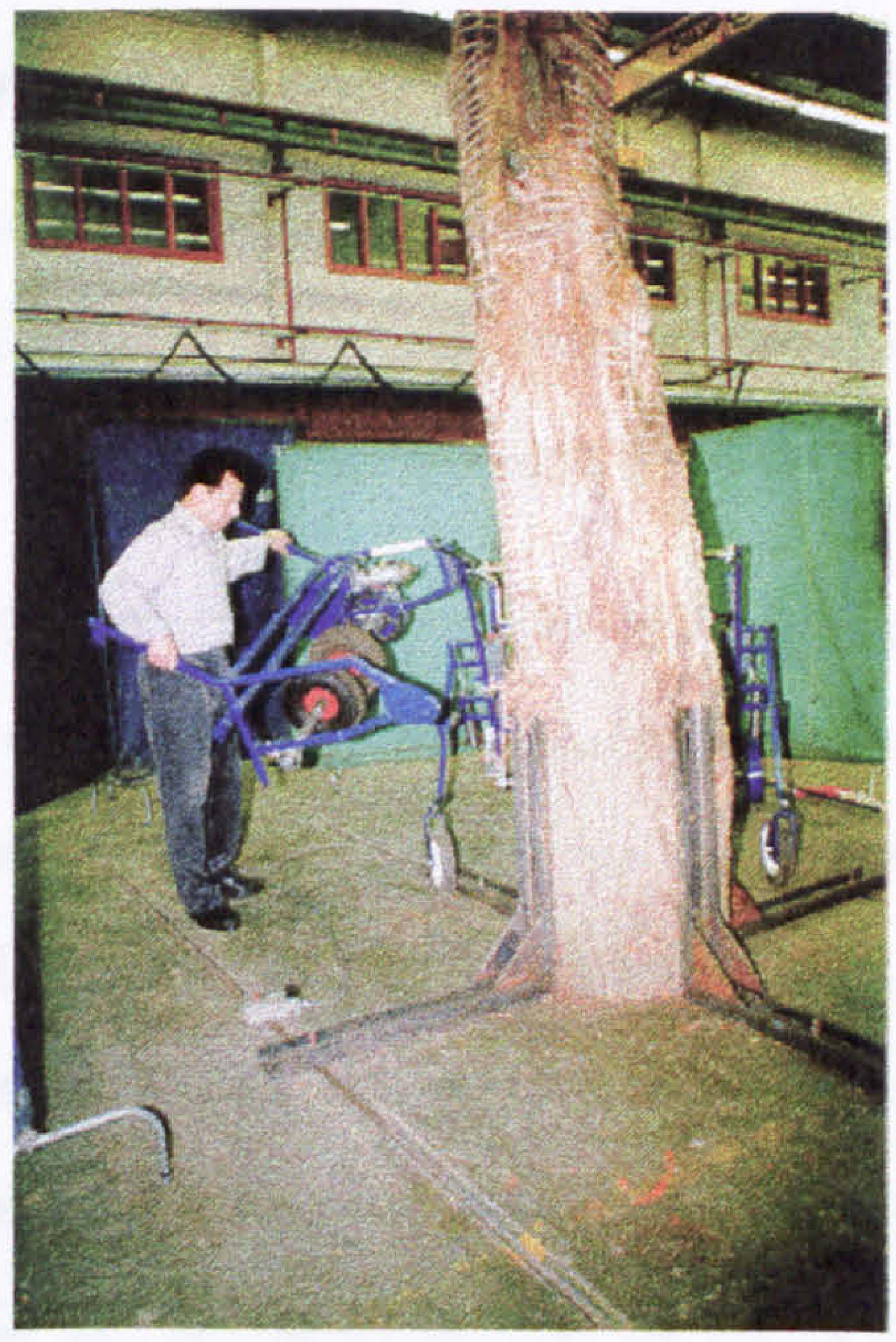


Figure 6-5 Opened "one axle unit" and approaching the tree

Figure 6-6 Closing the "one axle unit" to complete the attachment task

6.5. Ground test results

Ground tests were conducted to establish torque calibration curve for axle A in order to measure the power during the climbing tests. The optimum speed for climbing tests was also established by these tests (Section 5.4.9).

Data recorded by the computer during these tests are presented in Appendix C, Tables C-1 to C-5. The data were analysed to draw the torque curves for five levels of constant motor speeds, n_2, n_4, n_6, n_8 and n_{10} using the method explained in the Appendix. The torque calibration curve (Figure 5-5) was used to convert the voltage readings of the torque sensor to the real torque existing in the axle. Then the torque sensor readings were calibrated against the power sensor. Using this technique without a sensor on axle A, the axle torque during the climbing test was easily measured from power sensor records. The resultant curves, shown in Figure 6-7, are the best trend line fitted to the data points of each speed and show the relationship between voltage drop across the power sensor (the wire which connects two batteries in series) and the torque in axle A. The following results were seen from the curves:

1. the coefficient of determination , R^2, is more than 0.95 for all curves (n_2 to n_{10}). It proves that the voltage drop across the power sensor is a stable and accurate device to find the torque in axle A and to estimate the consumed electric power.
2. The Programmable Logic Controller (PLC) produces the same maximum torque in constant pre-set speeds of n_4, n_6, n_8 and n_{10}.

As a result of the ground tests, motor speed of n_6 was selected as the speed of the climbing tests.

The curve for speed n_6 is presented separately in Figure 6-8 and was used as a reference curve for the torque and power measurements during the tree climbing tests. Preliminary climbing tests were conducted to evaluate this speed. It was established as 0.17 m/sec. The test results showed that the controller provides enough power for the motor to lift 80 kg, the maximum payload of the climbing tests. Speed n_6 was slower but better than n_8 and n_{10} because it was a safe climbing speed at which the machine did not bounce or move to one side of the tree. From the torque point of view Figure 6-7 shows that at this speed the motor can almost produce the same maximum power as that produced at higher speeds like n_8 or n_{10}. Therefore, the full power performance will be achieved at this speed.

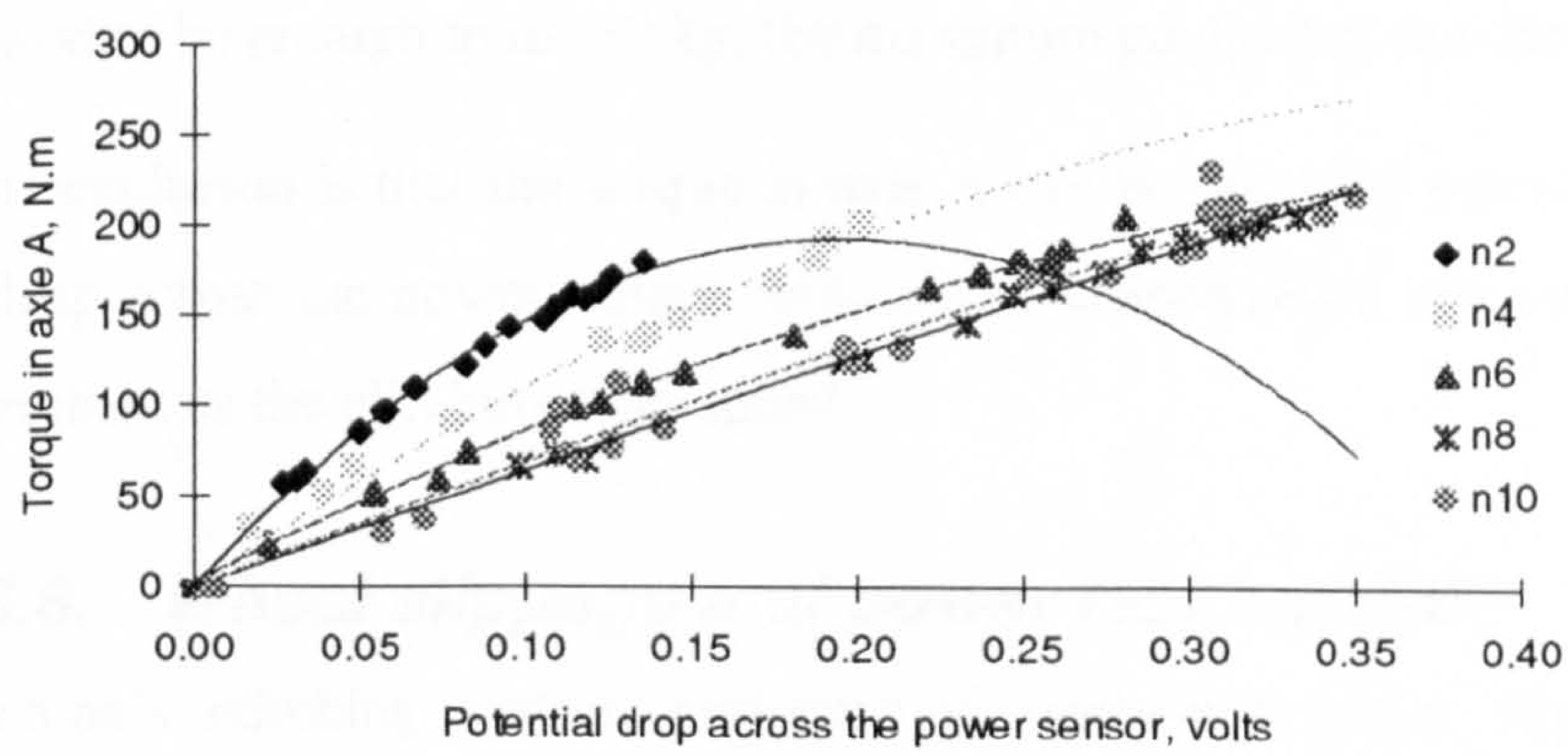


Figure 6-7 Axle torque versus voltage drop across power sensor for five level of motor speeds (n_2 to n_{10})

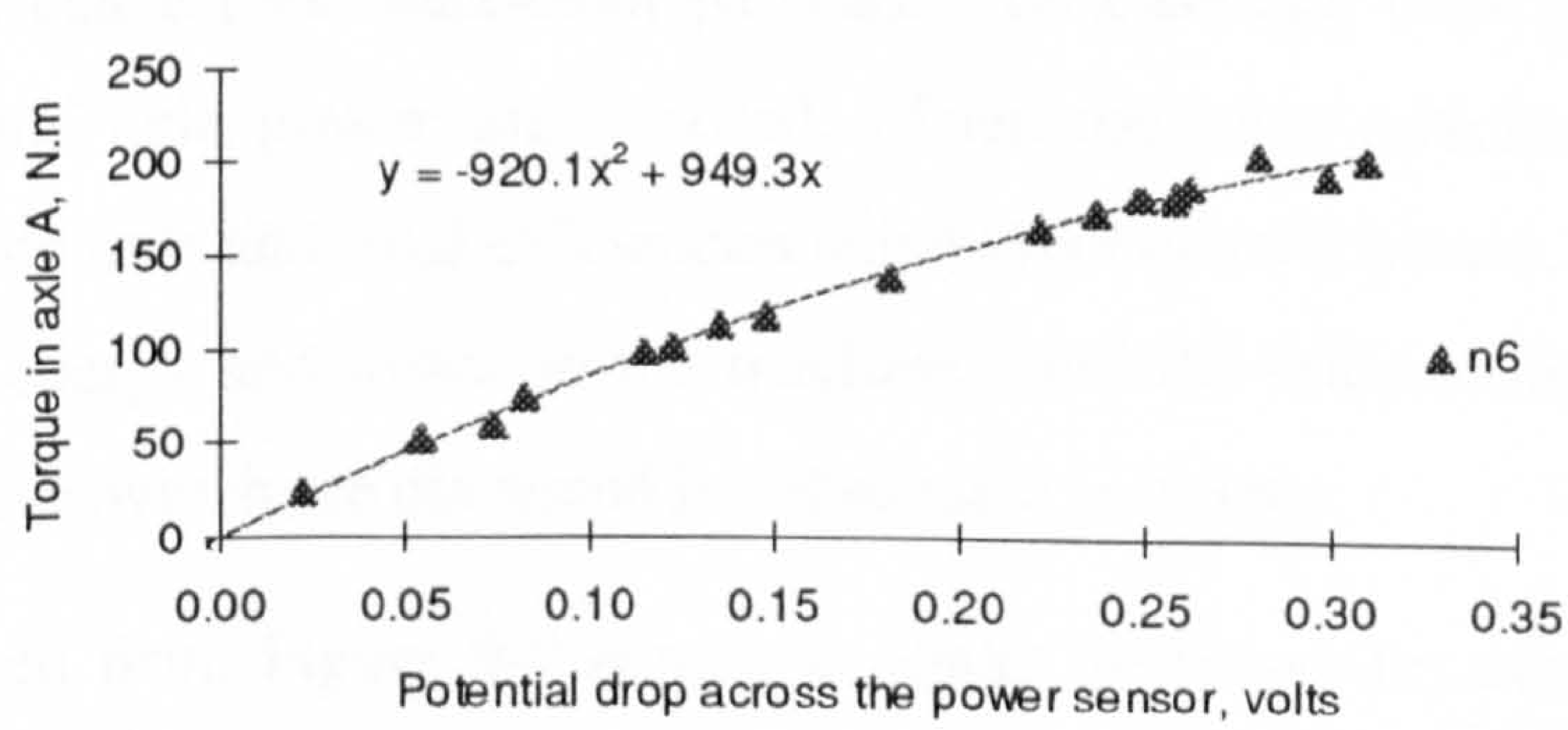


Figure 6-8 Axle torque versus the voltage drop across the power sensor for motor speed n_6

The power absorption at speed n_6 was evaluated from the torque curves. Voltage drop across the wire (ΔV), which are the values of horizontal axis in torque curves, are proportional to the power consumed by the motor (Equation 5.10). The maximum data point of curve n6 represents 1.35 volts on horizontal axis and the maximum value achieved is 1.62 volts which is on curve n_{10}, so, speed n_6 can use about 83 % of the maximum power achievable by motors which preliminary climbing tests showed to be enough to lift 80 kg, the maximum payload of the climbing tests.

The main conclusion is that the torque in axle A can be precisely calculated from the voltage drop across the power sensor, and, from the analysis of axle curves speed n_6 was determined as the climbing tests speed.

6.6. Wheel slippage and power test results

The machine's climbing system performance, including wheel slippage, power consumption and efficiency are evaluated in this section. Firstly the wheel slippage graphs are presented for different wheel spacing, spring tensions and pay loads to establish the stability and traction control systems' performance at each setting (concepts C1 and L1 in Figure 2-4). The best value of wheel slippage cannot be determined from the curves until the tractive efficiency at each setting is established because optimum wheel slippage occurs at maximum tractive efficiency when the machine can lift the maximum payload. To calculate tractive efficiency, climbing power and axle power are required. Then the total performance is evaluated by calculating axle and total efficiencies using input electric power. Figure 6-9 shows the flow of energy and losses in the machine, and the relationship between power and efficiencies which are discussed in subsequent sections.

As is seen from Figure 6-9 electrical power decreases because of heat and friction losses in the motor and moving components of the transmission system until it is converted into the mechanical power in driving axle. The axle power decreases due to rolling resistance and slippage. The main element considered here is wheel slippage because it is used to evaluate the machines' horizontal stability and traction control systems. The slippage, power and speed were measured simultaneously with one test design including two levels of wheel spacing D1=300 mm and D2 = 350 mm , two levels of spring tension T1= 555 N and T2 = 639 N, and five levels of payloads L1=0 kg , L2 = 20 kg, L3 = 40 kg, L4 = 60 kg and L5 = 80 kg. The machine was set for these levels with the sequence determined by a "randomised complete block design" explained in Section 5.4.10.

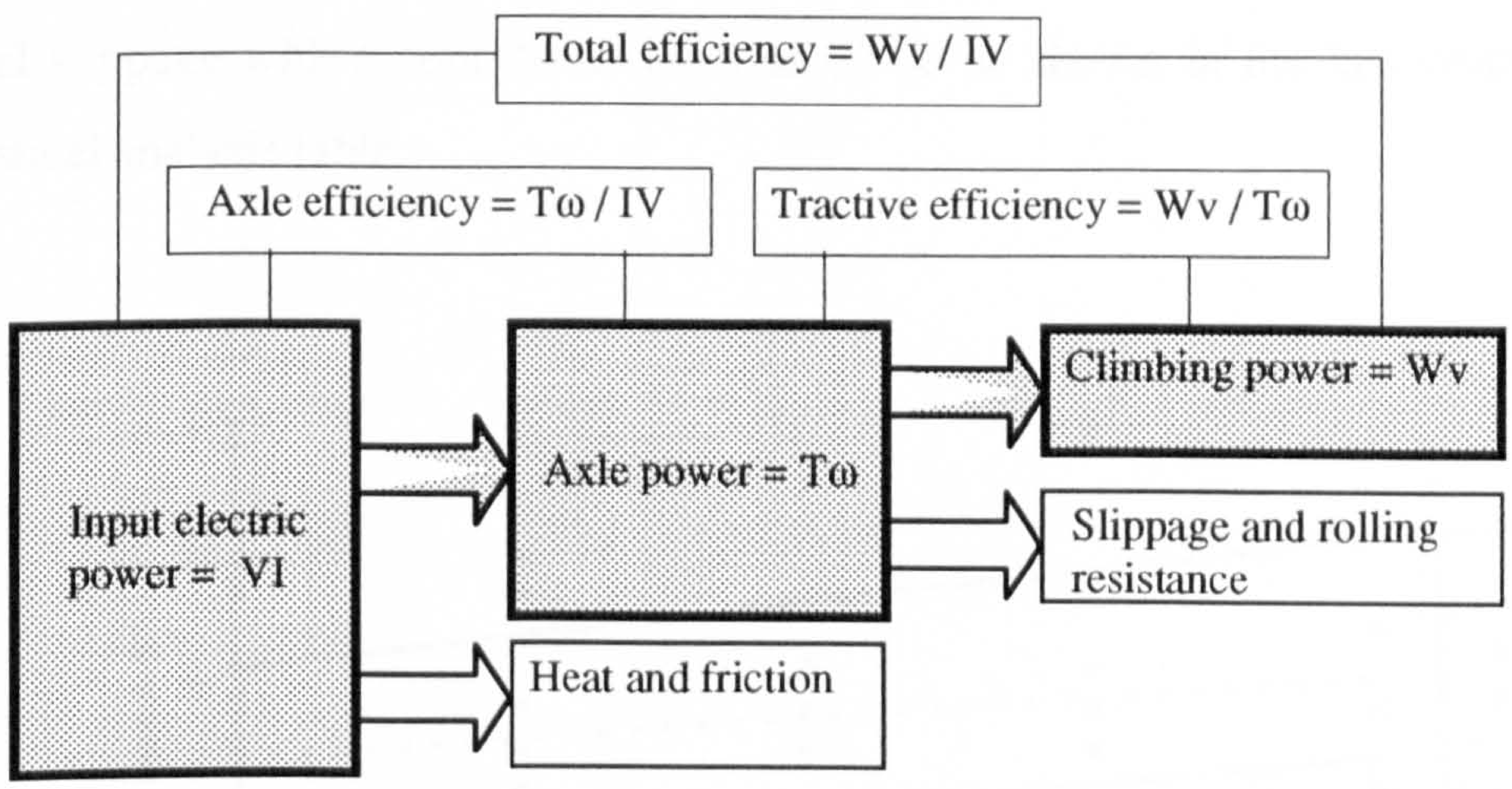


Figure 6-9 Flow of energy and losses in the climbing system

Using the measurement system the slippage and power tests were carried out and results were analysed using a statistical analysis package as explained in Section 5.4.10.1. In subsequent sections the results of slippage, power and efficiency tests are presented graphically. Appendix C contains the test data followed by statistical analysis for each test. In statistical analysis tables factor A is wheel spacing; factor B is spring tension and factor C is pay load. The last column in each table with the heading "prob" shows the probability of the factor not being significant for the measuring parameter. Values less than 0.05 mean that the factor has a significant effect on the measuring parameter and vice versa.

6.6.1. Slippage test results

Wheel slippage was selected as the control criterion to determine the climbing system performance and also to establish the best wheel spacing and spring tension.

Depending on the tyre and surface type each wheel has the maximum tractive efficiency or lifting capacity at a specific slippage value. To calculate the slippage value Equation 5.6 was used to convert the slippage sensor readings (Section 5.2.2) to slippage values. The test result data are presented in Table C-6. Figure 6-10 is a graphical presentation of the results and the best linear curve fitted to them. Statistical analysis of the results is presented in Table C-7 and shows that the wheel spacing (factor A) and the spring tension (factor B) both have significant effect on the wheel slippage with a confidence level of 95 % as shown in the last column of the statistical analysis table.

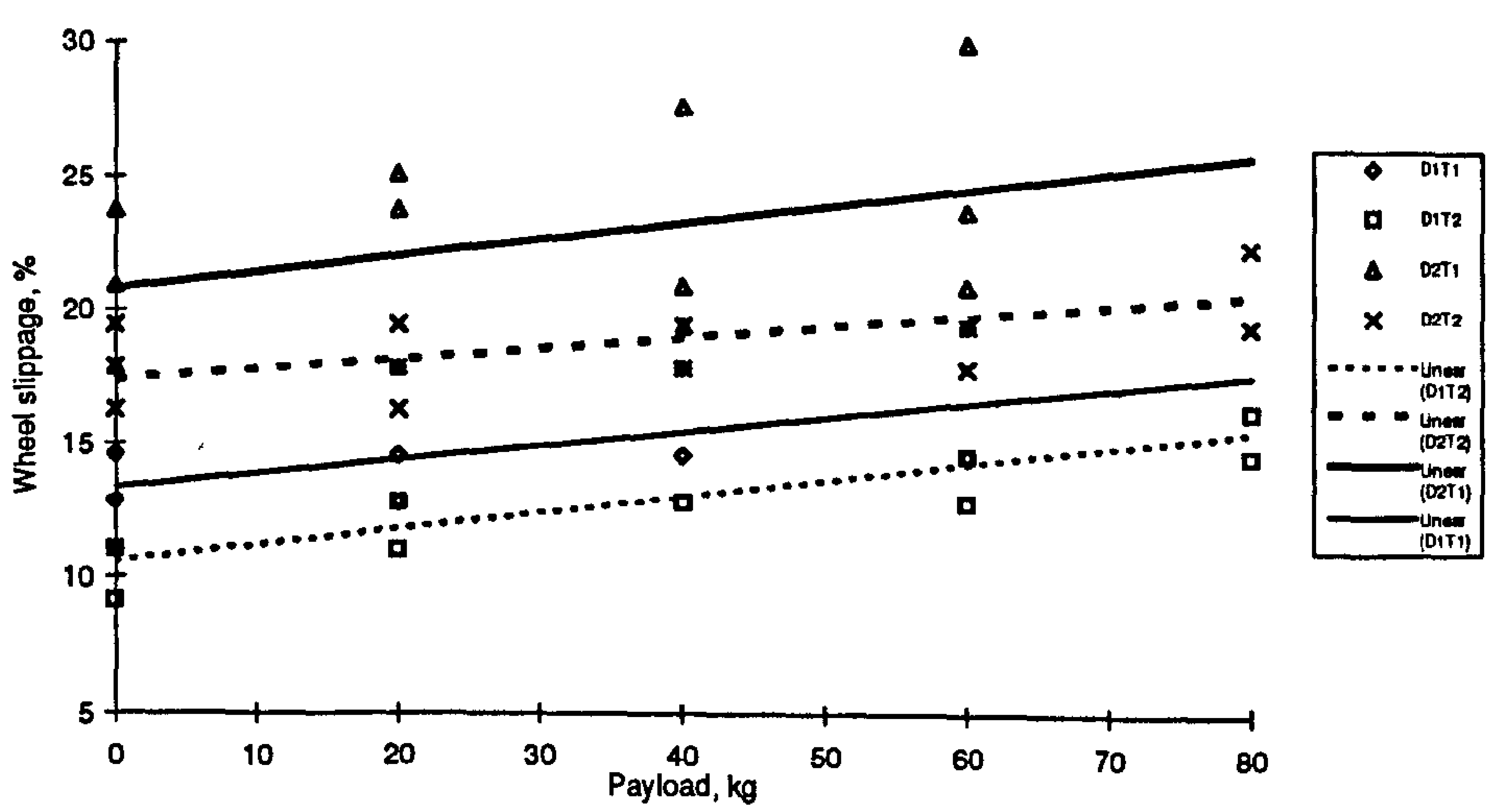


Figure 6-10 Relationship between the payload and wheel slippage at wheel spacing: D1= 300 mm , D2=350 mm and spring tensions: T1=555 N , T2= 639 N.

The main conclusion is that wheel spacing and spring tension have an effect on slippage and can reduce or exceed it. This is a positive result and shows that the control stability and traction control system can effectively control the slippage and increase the traction force to improve the payload capacity of the machine. Figure 6-10 shows that wheel slippage generally increases with payload. It also shows that to decrease the slippage, wheel spacing, D, can be reduced or spring tension, T, can be increased. The minimum slippage occurs with D1T2 which is low wheel spacing and high spring tension and the maximum slippage curve occurs with D2T1. According to the curves the slippage values are between 10 - 23% for all the tests. To establish the optimum slippage value the axle and climbing powers are required, which will be discussed in the tractive efficiency section.

6.6.2. Input electric power

The test was conducted according to the same method described for slippage tests. Equation 5.10 was used to convert the power sensor readings to actual input electric power. Figure 6-11 shows the graphic results.

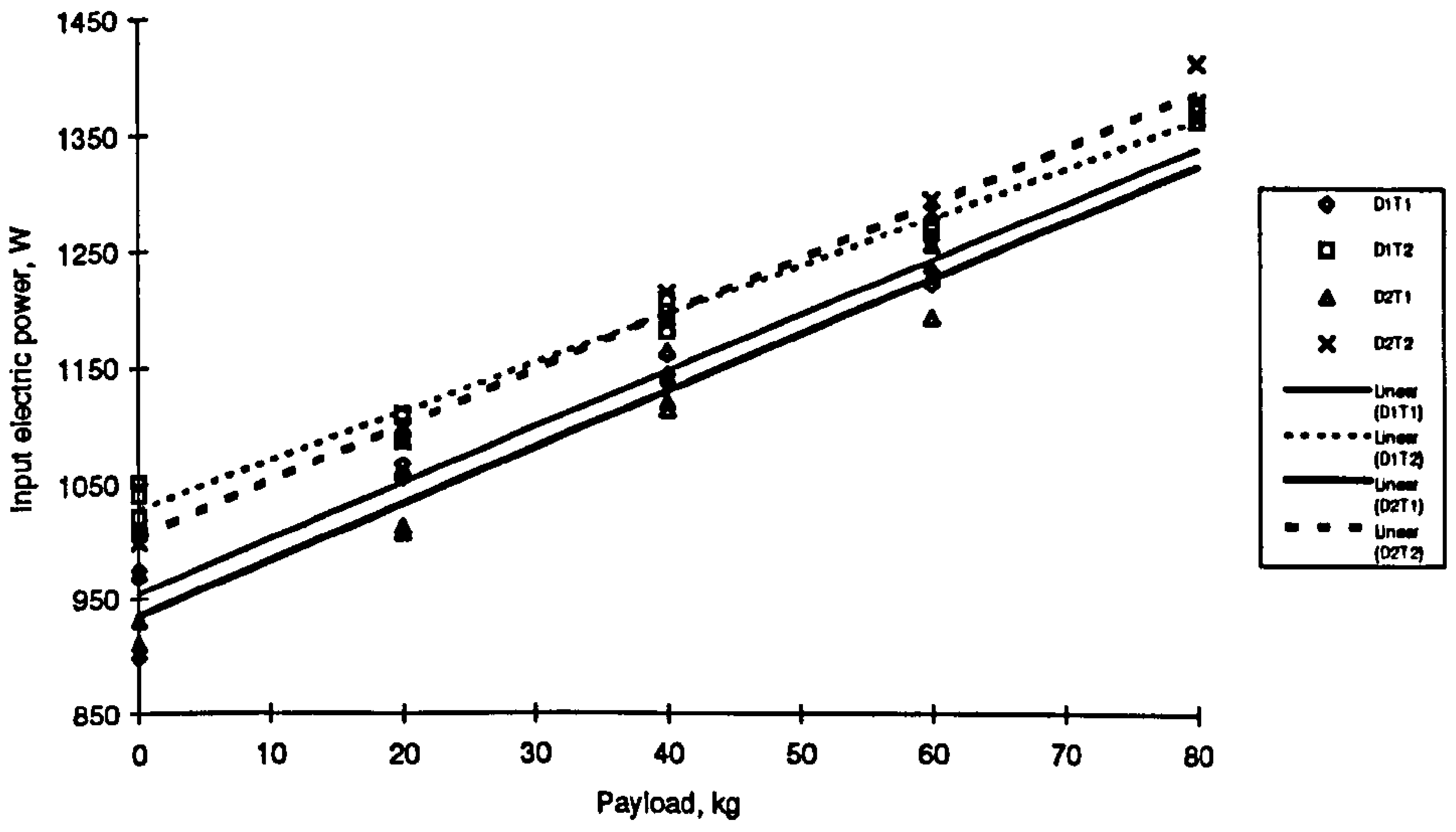


Figure 6-11 Relationship between payload and input electric power at wheel spacing: D1 = 300 mm, D2 = 350 mm and tensions: T1= 555 N , T2 = 632N.

Statistical analysis was undertaken to study the effect of wheel spacing, D, spring tension, T, and payload, L, on the power consumption. The statistical analysis (Table C-9) shows that changing the wheel spacing, D, from 300 mm to 350 mm has no significant effect on the input power consumption. Increasing the spring tension , T, from T1 = 555 N to T2 = 639 N has a significant effect on the power consumption, increasing it about 100 watts. Increasing the payload from 0 to 80 kg increases the power consumption from 950 to 1350 watts. Given that the experimental machine weight at zero payload is 137.5 kg and assuming a linear relationship between weight and power, the following equation can be established between weight and power:

$$\text{(Electric Power)} = 5 \times \text{(total machine weight)} + 262.5 \qquad (6.1)$$

Equation 6.1 can be used to calculate the required input electric power (watt) to design a climbing machine (for a climbing speed of 0.17 m/sec) when the machine weight (including payload) in kg is known. This equation calculate the machine maximum power consumption of 1450 watts.

6.6.3. Axle power

The objective is to establish the relationship between the payload and the machine axle power and also to calculate the axle efficiency which is the ratio of axle power to the input electric power. Axle power, which is the power in a rotating shaft can be calculated from the following equation (Meriam and Kraige 1993):

$$P_a = T\,\omega \qquad (6.2)$$

Where ω is axle angular speed, radian/sec and T is the axle torque in N.m which was calibrated against the power sensor in ground tests.

T is determined using the following equation taken from Figure 6-8 :

$$T = -920.1(\text{power sensor reading, volts})^2 + 949.3(\text{power sensor reading, volts}) \qquad (6.3)$$

The angular speed, ω, was calculated from the following equation:

$$\omega = 2\pi n / 60 \qquad (6.4)$$

Where n is axle speed in revolution per minute, calculated from the speed sensor calibration curve (Figure 5-6) as:

$$n = 8.5(\text{speed sensor reading, volts}) \qquad (6.5)$$

Substituting Equation (6.5) in Equation (6.4) gives:

$$\omega = 0.890 \text{ (speed sensor, volts)} \qquad (6.6)$$

Substituting Equations (6.6) and (6.3) in equation (7.1) gives:

(6.7)

$$P_a = [\,-920.1 \text{ (power sensor reading, volts)}^2 + 949.3 \text{ (power sensor reading, volts)}] \times [0.890\text{(speed sensor reading, volts)}]$$

The axle power for each test run was directly calculated by substituting the average value of power sensor and axle speed sensor readings in Equation (6.7).

Figure 6-12 shows P_a for different machine settings calculated from equation (6.7).

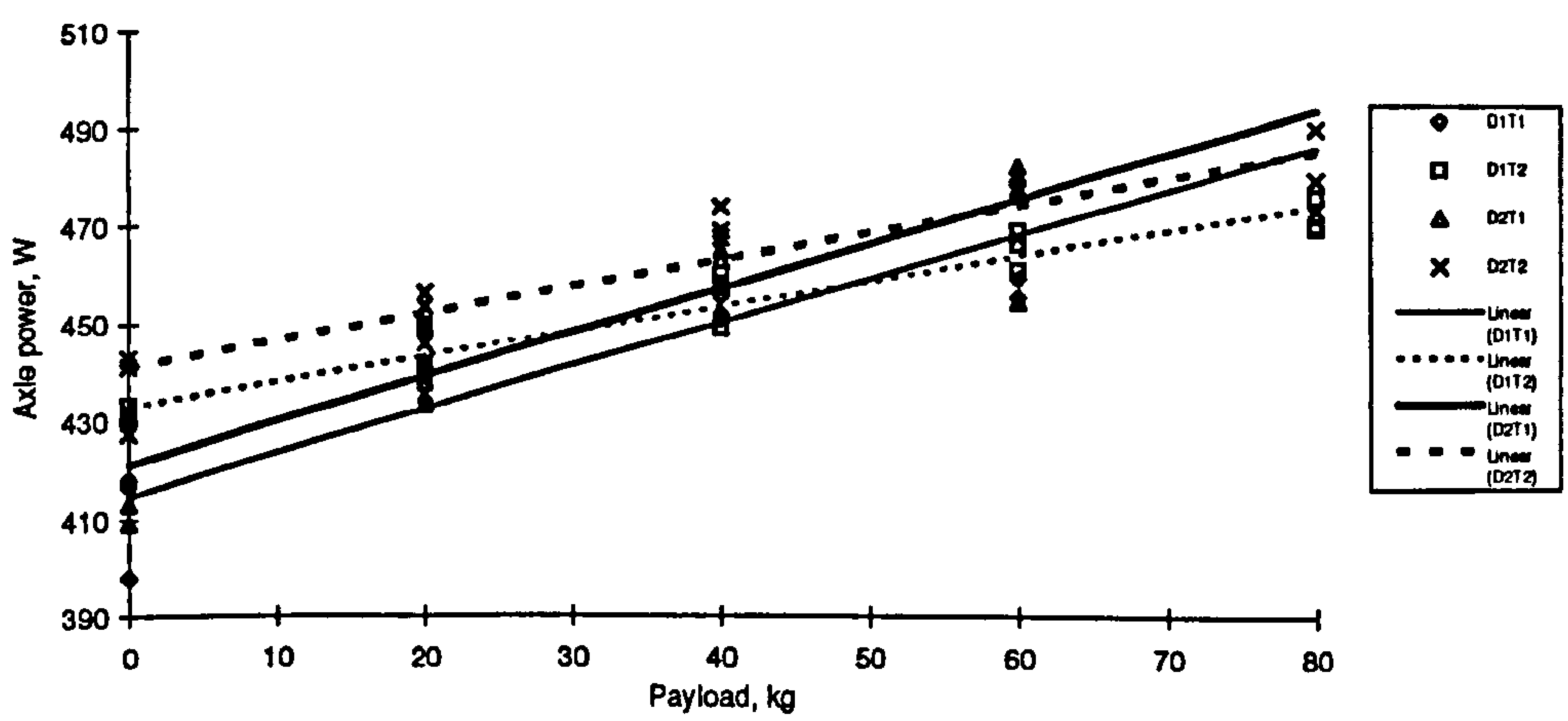


Figure 6-12 Relationship between the payload and axle mechanical power at wheel spacing: D1= 300 mm , D2=350 mm and spring tension: T1=555 N , T2= 639 N.

The statistical analysis for the axle power (Table C-11) shows that wheel spacing and spring tension do not have a significant effect on the axle power with in the set limits.

6.6.4. Climbing power

The climbing power is the power required to lift the machine total weight and is calculated from the following equation:

$$P_c = Wv \qquad (6.8)$$

Where P_c is climbing power in watts. W is the total weight including the payload for each run in N. v is the actual climbing speed, m/sec, which was calculated from the following equation:

$$v = x/t \qquad (6.9)$$

Where distance x is the test path height which was 2050 mm for all the slippage and power tests. t is the time taken by the machine to climb the test path height, calculated by counting the number of data points recorded during the climbing period by the speed sensor channel. The sampling rate of the data recording was 35 Hz. Dividing the number of data points by 35 gave time, t.

Values of climbing power are given in Table C-12 and shown graphically in Figure 6-13. The statistical analysis (Table C-13) showed that the spring tension does not have a significant effect on the climbing power but the wheel spacing does have a significant effect, in that decreasing the wheel spacing increases the climbing power. It can be concluded that the machine climbed the tree faster when the wheel spacing was smaller because wheel grip was better and slippage decreased. Curves show that the climbing power changed in the range of 265 to 425 watts.

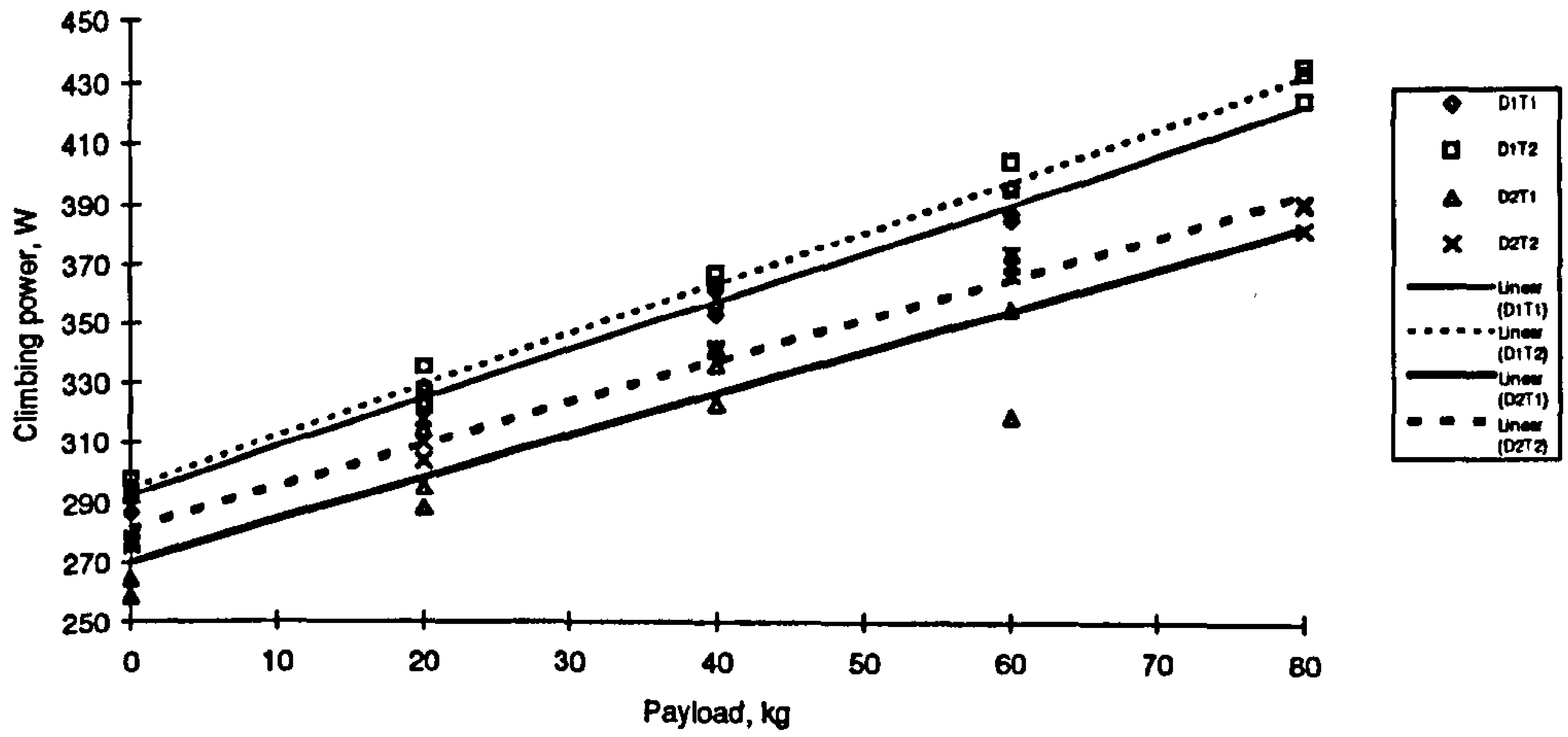


Figure 6-13 Relationship between the payload and climbing power at wheel spacing: D1= 300 mm , D2=350 mm and spring tensions: T1=555 N , T2= 639 N.

6.6.5. Tractive efficiency

Tractive efficiency is determined as the ratio of climbing power to axle power (Wv/Tω in Figure 6-9). The optimal values for the slippage, wheel spacing, spring tension, and payload can be determined from the tractive efficiency curves presented in Figure 6-14 (data in Table C-14). A maximum efficiency of 90 % is achieved for D1T2 (narrow wheel spacing and high spring tension) for an 80 kg payload. Statistical analysis in Table C-15 suggests that wheel spacing has a significant effect on the tractive efficiency in that narrow wheel spacing increases the tractive efficiency.

6.6.6. Axle efficiency

Axle efficiency is the ratio of the axle power to the input electric power (Tω/VI in Figure 6-9) and shows the efficiency of conversion of electrical power to mechanical power, including friction losses in the worm gear box and sprocket-chain transmission system .Figure 6-15 shows the axle efficiencies (data in Table C-16). The statistical analysis (Table C-17) suggests that the wheel spacing and spring tension both have a significant effect on the axle efficiency.

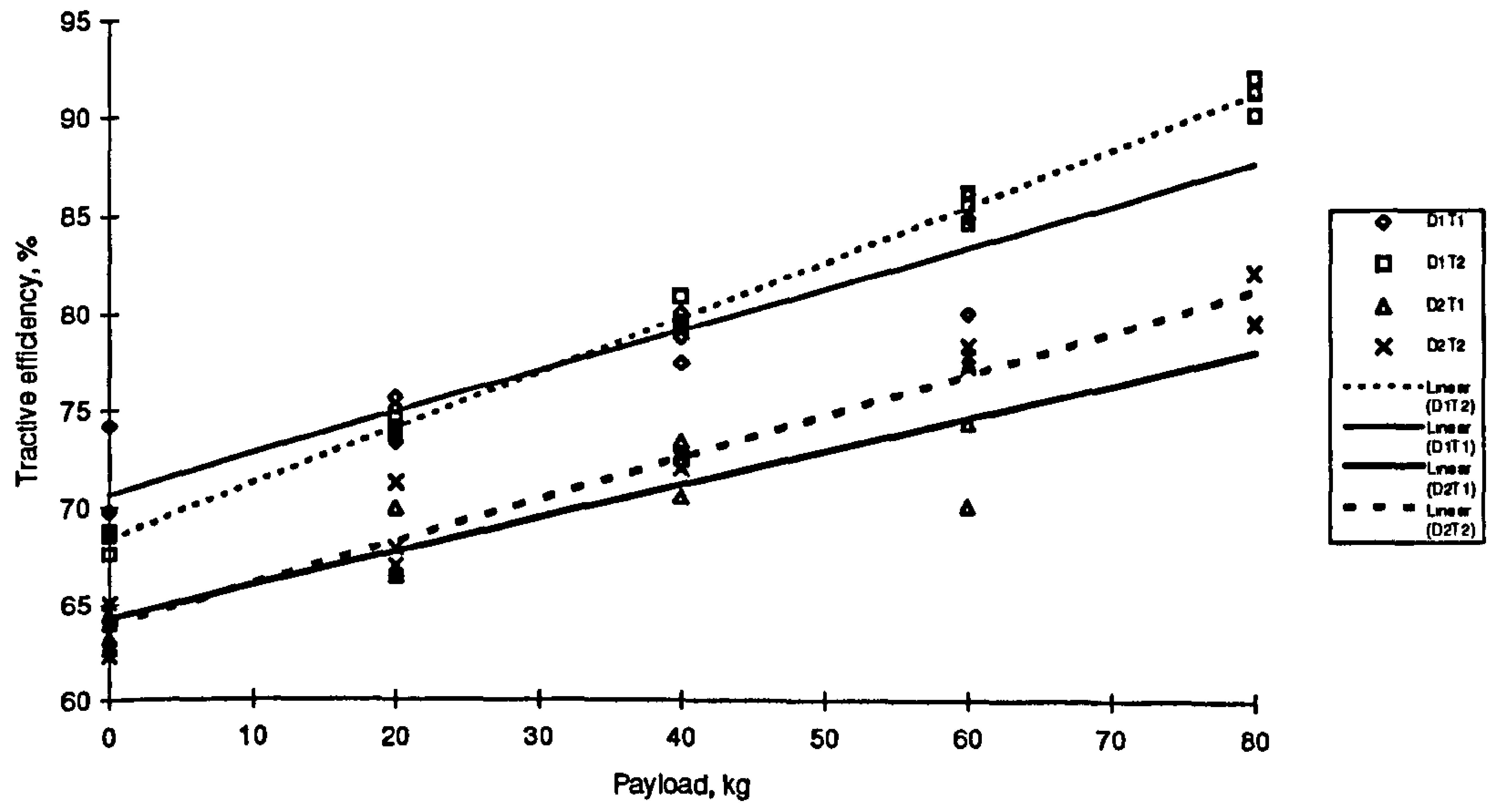


Figure 6-14 Relationship between the payload and tractive efficiency at wheel spacing: D1= 300 mm , D2=350 mm and spring tensions: T1=555 N , T2= 632N.

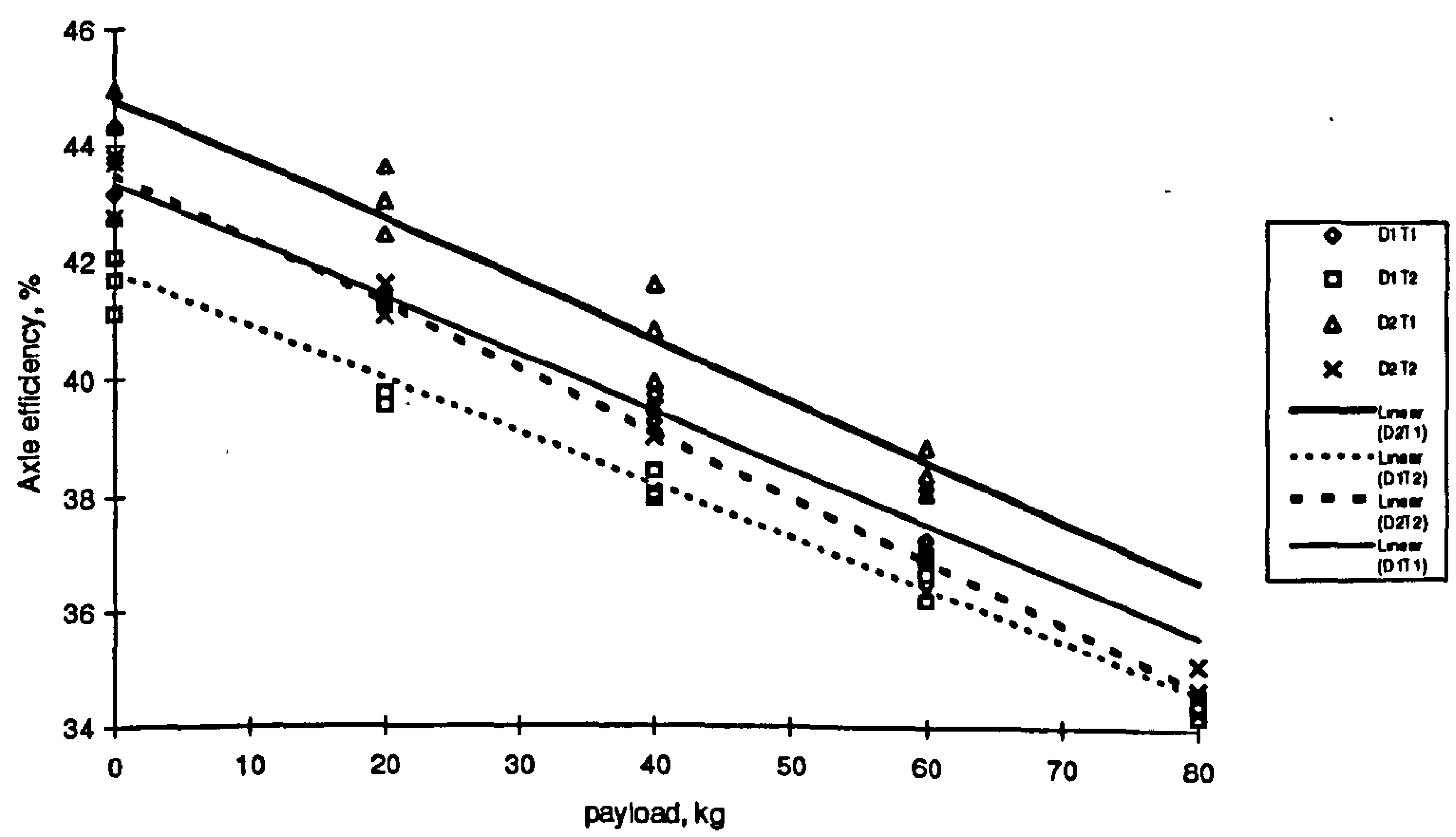


Figure 6-15 relationship between the payload and axle efficiency at wheel spacing: D1= 300 mm , D2=350 mm and spring tension: T1=555 N , T2= 639 N.

Axle efficiencies increase with lower spring tensions and wider wheel spacing but decrease with increasing payload on the machine. This is because friction losses and tyre deformation increase, whilst the motor efficiency decreases at higher loads

(Hindmarch 1985; Kenjo and Nagamori 1985) . The maximum axle efficiency was 45% for zero payload, with the wider wheel spacing and the lower spring tension (D2T1).

6.6.7. Total efficiency

Total efficiency is the ratio between the climbing power and input electric power and is the overall efficiency of the system (Wv / VI in Figure 6-9). Tables C-18 and C-19 show the data and the statistical analysis of the test results. Figure 6-16 shows that the total efficiency is in the range of 27 to 33 %. It is also clear from this Figure that the total efficiency for the wider wheel spacing, has a maximum value of 29 % at about 30 kg payload for low spring tension (curve D2T1) and a maximum value 28.5 % at 40 kg payload for high spring tension (curve D2T2) and that efficiency starts to decrease which is attributed to extra wheel slippage.

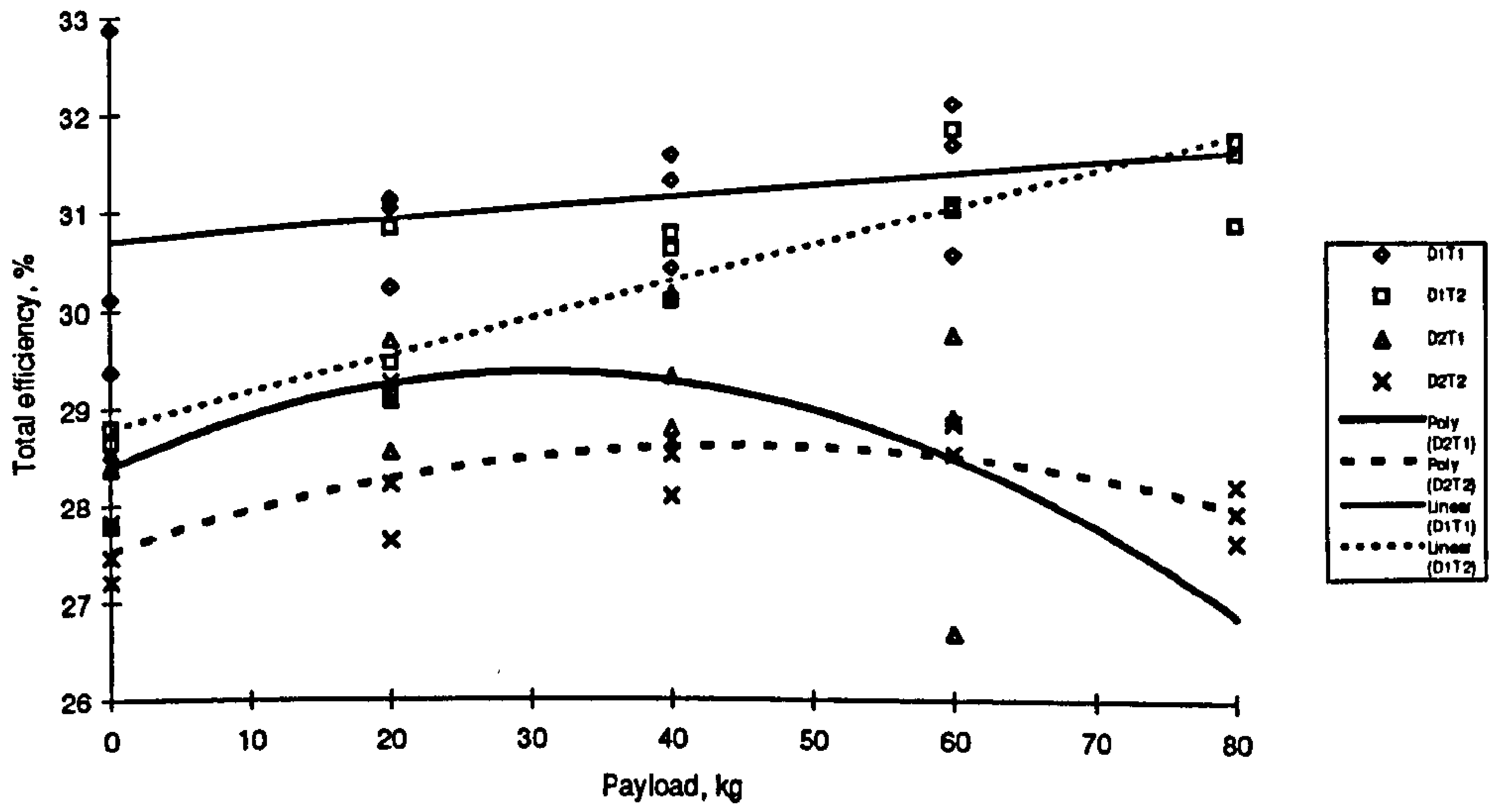


Figure 6-16. Relationship between the payload and Total efficiency at wheel spacing: D1= 300 mm , D2=350 mm and spring tension: T1=555 N , T2= 639 N.

6.6.8. Conclusions of slippage and power tests

From the tractive efficiency curves (Figure 6-14) it can be seen that the machine can achieve a maximum tractive efficiency of 90% which is a good performance and shows that the climbing system can use from 90% of the available axle power to lift payloads. It shows that selected concepts of the horizontal plane stability system and traction control system (concepts C1 and L1 in Figure 2-4) are suitable for the machine. Tractive efficiency curves show that narrow wheel spacing (curves D1T1 and D1T2) can achieve better efficiencies. The best wheel slippage occurs at these settings and can be derived from slippage curves D1T1 and D1T2 of Figure 6-10 which is between 10 to 15%. This analysis shows that the self guidance system introduced by Equation 2.23 which led to the selection of wheel spacing of 300 and 350 mm is valid and that the climbing system can perform satisfactorily with these wheel spacing. The test results suggest 300 mm wheel spacing (8% greater that 277 mm, the minimum value suggested by self guidance system, Equation 2.23) for driving axles in order to achieve a better tractive efficiency but the results of stability experiments in Table 5-5 suggest a minimum of 400 mm to prevent the machine from moving to one side of the tree on points where the tree cross section is not completely circular or there is a bump on the tree surface. To overcome this problem and to use the maximum tractive efficiency benefits it is recommended to adjust the wheel spacing on 300 mm (8 % greater than minimum value calculated from self guidance Equation) and to add a pair of safety shoes as suggested in concept L3 (Figure 2-4) which prevent the machine from deviating from a straight line at critical points on the tree.

Increasing the tractive efficiency by payload suggests that the machine benefits from a positive weight transfer system (Figure 2-5) which pushes the driving wheels harder onto the tree surface when the payload increases and increases the tractive efficiency automatically.

Input electric power measurements show that the machine consumes a maximum power of 1.45 kW.

CHAPTER 7

ECONOMIC ANALYSIS OF THE MACHINE

7. Economic Analysis of the Machine

A simple economic analysis was conducted to establish the situation where the machine deployment can be justified economically under Iranian conditions. This situation was defined by the number of trees which should be harvested annually. Using this definition the machine is economically justified when:

annual machine cost ≤ annual hand harvesting cost

or

annual machine cost ≤ hand harvesting cost per tree × NT (7.1)

Therefore the minimum number of trees, NT, to be harvested annually is when the machine harvesting cost is equal to hand harvesting cost and can be defined by rewriting Equation 7.1 for NT as follows:

NT = annual machine cost / hand harvesting cost per tree (7.2)

Equation 7.2 shows that to find the minimum number of trees, NT, the annual machine cost and the hand harvesting cost per tree are required.

The annual machine costs are divided into two categories, fixed costs and running costs. Fixed costs are dependent on the duration of ownership of the machine and its main elements are depreciation and interest. Running costs vary in proportion to the utilisation of the machine and includes fuel and oil, repair and maintenance and labour.

To calculate fixed costs an estimation of machine price was required. This price was established based on the production price in a local factory in Iran. Five factory managers were asked to quote a price for a batch of 10 machines including the material and part prices supplied in UK (Table D-1) from drawings, photographs and explanation of the experimental machine. The average price of 11 million Rials was established. An exchange rate of 4 750 Iranian Rials equal to one pound was used to convert the parts price (£ 1225) to Rials. The price of a 3 kW electricity generator popular in Iran, 2.5 million Rials, was added to the price, thus, a total price of 13.5 million Rials was used as machine price (PP) in cost calculations.

This method of price estimation is justified by Ullman (1992) which explains that in many companies cost estimating is accomplished by a professional who specialises in determining the cost of a component whether it is made in-house or purchased from a vendor. This method was also used by Amjad (1994) to predict the price of an improved or new machine that would be charged to the farmer by a local factory in a research to select an appropriate mechanisation system in Pakistan.

The straight-line depreciation was used for this study because it is most suitable for estimating costs for the entire life of the machine. It is, however, a simplified method and ignores the more rapid depreciation which occurs in the early life of the machine. It is more important when a machine is traded after a short period of ownership but it does not apply to the Iranian situation because farmers usually get a loan from the Agricultural Bank with an interest rate of 14 % and return period of 10 years to buy a farm machine and are not allowed to sell it after a short period.

To calculate running costs 10 000 Rials per day was allocated for the machine operator i.e. the salary paid to a full time worker. The accumulated repair and maintenance was established 50 % of the purchase price. This is a statistical value for combine harvesters and forage blowers (Wintey 1988), the type of agricultural machines which do not engage with soil (similar to date harvester), therefore, have a medium repair cost. The hand harvesting cost was established from the date grower information. Two experienced part time workers usually make a contract to harvest a tree for 3 000 Rials. Considering the above explanation and discussions, detailed calculations are presented in Appendix D. Equation D.14 was developed which calculates minimum number of trees, NT, based on purchase price, PP, in million Rials and machine harvesting time, MHT, in minutes. The equation is presented graphically in Figure 7-1 for current experimental machine (MHT = 32 minutes) and for the improved design using quick latches (MHT = 22 minutes) explained in Section 6.4. Solid bars show that 1083 trees should be harvested for the current design at 13.5 million Rials purchase price and the continuous line curve shows that 72 days is required to harvest this number of trees. The improved design (MHT = 22 min) shown by grey bars and dotted line curve. The grey bars show that 978 trees should be harvested annually and the dotted line curve shows that 45 days is required to harvest this number of trees.

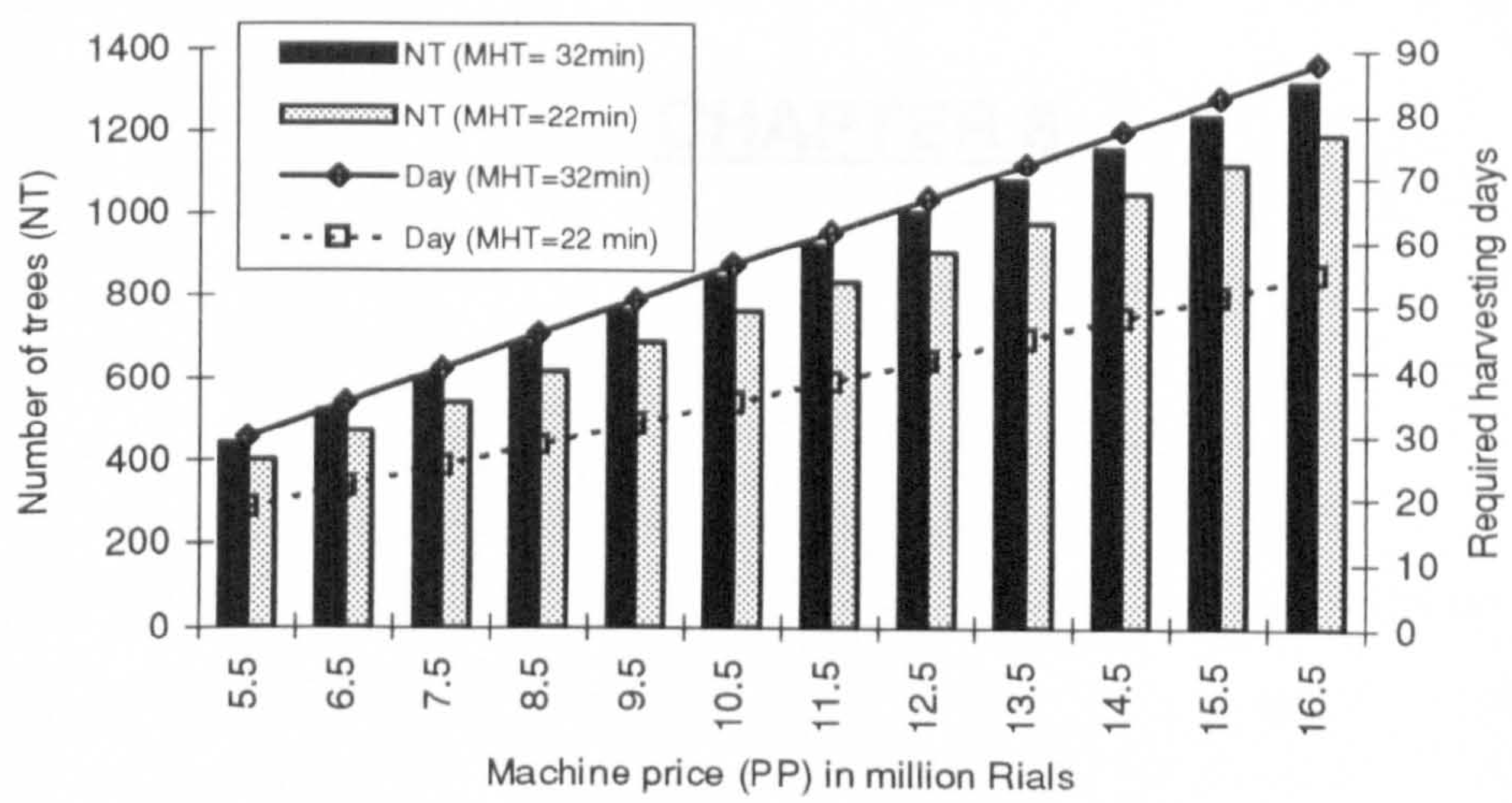


Figure 7-1 Number of trees and required harvesting days as a function of machine price

The machine price was estimated based on parts and materials supplied in UK, and as supplying them from internal market may reduce the purchase price the curve can be used to established the number of trees for a potentially cheaper purchase price.

From 45 days, required period to harvest the minimum number of trees for the improved design it is concluded that the machine is economically justified because in some areas, depending on the date variety and climate, the harvesting period may take up to 3 months (Kashani 1992), therefore, one machine has enough time to harvest more than the required minimum number of trees which will improve its profitability.

A comparison between the machine price of 13.5 million Rials (£ 2850) and an available machine which costs £ 16 000 (refer to target cost page 2-15) shows that the proposed machine costs 20% of the available technology which is a significant improvement in capital investment on date harvesting mechanisation.

CHAPTER 8

CONCLUSIONS

8. Conclusions

The main conclusion that may be drawn from this study is that it is possible to design a tree climbing machine using the principles of vertical wheeled traction to harvest and service date palm trees. This system would provide a significant improvement in the present techniques used for date harvesting. The machine has the potential to be manufactured at 20% of the cost, between 2 - 4% of the weight (4-8 tonnes) and 3% of the power requirements (49 kW) of the existing mechanised harvesting systems while still maintaining equal performance.

The design criteria and performance of the machine can be predicted from the use of mechanical analysis using vertical traction theory. The theory would enable the design of a machine for a wide range of tree size, surface roughness and machine weight. The traction control system controlled the load capacity of the machine from zero to 100 kg. It was also able to establish an optimum wheel slippage of 10 to 15% and achieve a maximum tractive efficiency of 90%.

Additional conclusions are:

1. The existing harvesting systems suffer from a high level of capital investment and lack of manoeuvrability in closely planted date groves. Traditional hand harvesting methods suffer from many problems such as worker safety, lack of experienced labour and poor date quality caused by the harvesting method.

2. The optimum solution for the tree climbing machine is a vertical climbing system which has two driving axles and can easily climb the uneven vertical surface of the tree. It is powered electrically by on board batteries, a generator or another DC power source on the ground.

3. Double axle drive mechanisms have a significant advantage over single axle systems when climbing rough date palm surfaces. The experimental machine's lifting capacity over a 41 mm leaf base decreases from 100 kg to 20 kg with one driving axle.

4. The tree should be able to resist the stresses applied by the machine without significant damage, because the minimum safety factor (i.e., the ratio of maximum applied shear stress ($\tau = 0.14$ MPa) to failure stress) was established as 7.

5. The experimental machine could climb the tree with a maximum speed of 0.27 m/sec. The optimum climbing speed was 0.17m/sec for best control.

6. The machine can lift the 50 kg target payload which corresponds to the maximum yield of a popular date tree (Ghasb) in Iran. However, the machine can lift a maximum payload of 100 kg which gives a practical margin of performance.

7. The machine can climb an uneven tree surface caused by leaf bases of up to 41 mm high.

8. Self guidance is assessed by the range of sideways adjustment of the wheels, enabling both concentric and eccentric loading.

9. Given a field efficiency of 75 % the proposed harvester can potentially harvest a tree in 22 minutes which is 18% faster than the traditional hand harvesting method in Iran and 6 % faster than the average time of a mechanised method used in Saudi-Arabia.

10. The harvester can be justified economically for Iranian farmers if it harvests more than 978 trees annually which represents 45 days' use with a field efficiency of 75%.

CHAPTER 9

RECOMMENDATIONS FOR FUTURE WORK

9. Recommendations for Future Work

As the machine looks promising more work on its development could be undertaken as follows:

1. Improving the machine performance:

   a) Decreasing the transformation time from moving on the ground to climbing mode. Using a quick attach mechanism for the "one axle unit" gate and spring suspension mechanism can reduce the transformation time (see concept J2 of Figure 2-4).

   b) Add an alternative safety system to prevent the machine from moving to one side of the tree when it faces an unexpected bump on the tree surface or the tree cross-section is not completely round. One alternative is mounting simple guide shoes to the sides of the machine to prevent it from moving to one side of the tree in critical moments (such as the ones suggested for concept C4 in Figure 2-4). A second alternative solution is changing the rigid idle axle B to a self steering axis that turns to the opposite side when the machine is diverted to one side of the tree and only one wheel of the axle is touching the tree.

   c) Select a more suitable transmission for the drive axle to increase the ground clearance.

1. Design and develop a bunch cutter and the basket (bunch holder) to the machine.
2. Test the complete prototype under representative farm conditions in date groves.
3. Consider the detailed production costs under Iranian commercial conditions.
4. Develop a pollinator and pruning unit to be fitted to the harvesting arm instead of the bunch cutter. These units would change the harvester to a date service machine. The machine then can be used to do two expensive cultural operations

after harvesting which take place in different seasons and do not overlap with harvesting operations.

5. Consider methods to utilise the potential energy of the dates and the machine as they descend the tree.
6. Consider the machine application to service other palm trees especially oil palms.
7. Consider other applications where climbing a smooth or uneven surface of a column of any shape is required.
8. Develop the design method further to encompass a wide range of trees and machines and to develop it into a more comprehensive design tool.

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APPENDIX A

DATA OF THE DATE TREE MEASUREMENTS

Appendix A : Data of the Date Tree Measurements

Table A-1 The tree and fruit sizes measurements explained in section 3.4.

No.	1	2	3	4	5	6
City	Bam	Bam	Bam	Bam	Bam	Bam
Grove name	Dehghan	Dehghan	Dehghan	Dehghan	Istghah	Istghah
Variety	Mazafati	Mazafati	Mazafati	Mazafati	Mazafati	Mazafati
Tree trunk height, m	8.20	7.60	9.40	7.70	10	7
Diameter at ground, cm	76	70	83	72	85	79
Diameter at crown, cm	57	54	60	49	58	52
Row spacing, m	3.5	5	4.2	4.5	6.1	5.2
Across row spacing, m	4.2	7.1	5.3	5.5	6.2	7.2
Bunch & stalk weight, kg	14.5	12.5	13	14.5	12	12.5
Bunch diameter, cm	53	49	39	60	65	40
bunch length, cm	97	87	95	111	102	81
Number of bunches	11	10	12	8	9	9
Leaf base pitch, mm	101	115	93	129	70	127
Leaf base height, mm	60	55	70	61	75	49
Leaf base width, mm	237	276	290	301	282	252
Yield, kg	127.6	100	124.8	92.8	86.4	90
Harvesting method	B. Sh.	B. Sh.	B. Sh.	B. Sh.	B. Sh.	B. Sh.
Distance to nearest tree, m	4.1	3.2	4.5	3.5	5	5.2
Climbing speed, m/sec	0.4	0.35	0.36	0.41	0.25	0.3

Data of the tree and fruit sizes measurements explained in section 3.4.(Continued)

No.	7	8	9	10	11	12
City	Bam	Bam	Bam	Bam	Bam	Shahdad
Grove name	Sabzvary	Shahrdari	Shahrdari	Shahrdari	Shahrdari	Baghe rig
Variety	Karoot	Karoot	Karoot	Karoot	Karoot	Ghasb
Tree trunk height, m	6.50	14	16.40	17	15.20	14.1
Diameter at ground, cm	81	58	60	67	62	59
Diameter at crown, cm	64	40	41	42	48	33
Row spacing, m	3.8	3.5	4.3	4.1	5.7	4.9
Across row spacing, m	4.1	5.4	6.3	6.7	6.9	5.2
Bunch & stalk weight, kg	13.5	10.1	7.2	6.9	10.1	7.5
Bunch diameter, cm	51	42	48	39	57	54
bunch length, cm	98	84	80	69	58	42
Number of bunches	10	8	6	8	5	8
Leaf base pitch, mm	112	75	81	115	83	75
Leaf base height, mm	34	62	57	74	80	60
Leaf base width, mm	247	262	261	300	256	245
Yield, kg	108	64.5	34.6	44.1	64.6	48
Harvesting method	B. Sh.	B. Sh.	B. Sh.	B. Sh.	B. Sh.	B. C.
Distance to nearest tree, m	4	2.5	3.1	3.9	3.6	4.9
Climbing speed, m/sec	0.28	0.2	0.28	0.21	0.18	0.31

Data of the tree and fruit sizes measurements explained in section 3.4.(Continued)

No.	13	14	15	16	17	18
City	Shahdad	Shahdad	Shahdad	Shahdad	Shahdad	Shahdad
Grove name	Kohstani	Kohstani	Rahmani	Rahmani	Baghe rig	Baghe rig
Variety	Mazafati	Mazafati	Mazafati	Mazafati	Porkoo	Porkoo
Tree height, m	15.3	12.1	16.2	11.2	6.5	7.2
Diameter at ground, cm	60	54	53	66	50	52
Diameter at crown, cm	38	30	33	40	37	30
Row spacing, m	4.8	5.7	5.1	4.8	5.1	3.9
Across row spacing, m	6.1	7.1	6.3	5.9	5.5	4.5
Bunch & stalk weight, kg	8.1	7.1	7	8.5	6	10.2
Bunch diameter, cm	49	41	51	59	61	55
bunch length, cm	61	49	51	57	72	85
Number of bunches	7	6	7	8	8	6
Leaf base pitch, mm	111	68	100	60	111	91
Leaf base height, mm	69	62	55	43	63	71
Leaf base width, mm	300	290	235	280	250	281
Yield, kg	45.4	34.1	39.2	54.4	38.4	49
Harvesting method	B. C.	B. C.	B. C.	B. C.	F.P. &B. C.	F.P. &B. C.
Distance to nearest tree, m	3	3.7	3.1	2.8	3.2	3.5
Climbing speed, m/sec	0.21	0.18	0.31	0.27	0.2	0.22

Data of the tree and fruit sizes measurements explained in section 3.4.(Continued)

No.	19	20	21	22	23
City	Shahdad	Shahdad	Shahdad	Shahdad	Shahdad
Grove name	Baghe rig	Baghe rig	Baghe rig	Char farsahk	Char farsahk
Variety	Porkoo	Porkoo	Porkoo	Abdolahi	Abdolahi
Tree height, m	6.5	7.5	8	10.1	9.5
Diameter at ground, cm	59	51	54	65	63
Diameter at crown, cm	40	41	44	43	39
Row spacing, m	4.4	5.1	4.8	3.8	4.5
Across row spacing, m	7.3	6.7	6	5.8	5.9
Bunch & stalk weight, kg	9.8	6.5	8.5	8	6.7
Bunch diameter, cm	48	47	49	47	61
bunch length, cm	81	75	80	75	61
Number of bunches	7	8	9	9	8
Leaf base pitch, mm	10.5	7.8	11.5	6.7	9
Leaf base height, mm	65	55	51	45	55
Leaf base width, mm	234	235	263	238	254
Yield, kg	38.4	49	54.9	41.6	61.2
Harvesting method	F.P.&B. C.	F.P.&B. C.	F.P.&B. C.	F. P.	F. P.
Distance to nearest tree, m	3.2	3.5	2.9	3.6	2.5
Climbing speed, m/sec	0.38	0.41	0.29	0.37	0.31

Data of the tree and fruit sizes measurements explained in section 3.4.(Continued)

No.	24	25	Ave.	Std.	Max.	Min.
City	Shahdad	Shahdad				
Grove name	Sahb dadi	Sahb dadi				
Variety	Bazmani	Bazmani				
Tree height, m	7.3	6.9	10.3	3.6	17	6.5
Diameter at ground, cm	58	61	63.9	10.4	85	50
Diameter at crown, cm	47	42	44.1	9.3	64	30
Row spacing, m	4	4.9	4.6	0.69	5.7	3.5
Across row spacing, m	4.5	5.1	5.8	0.96	7.3	4.1
Bunch & stalk weight, kg	10	11.1	9.67	2.66	14.5	6
Bunch diameter, cm	55	57	51.1	7.4	65	40
bunch length, cm	89	91	77.2	17.8	111	42
Number of bunches	10	10	8	2	11	7
Leaf base pitch, mm	105	80	94.3	19.9	115	60
Leaf base height, mm	60	62	60	10	75	41
Leaf base width, mm	237	235	262	30	301	234
Yield, kg	80	88.8	66.9	28.2	127.6	34.1
Harvesting method	F.P.	F.P.				
Distance to nearest tree, m	4.3	3.8	3.66	0.74	5.2	2.5
Climbing speed, m/sec	0.41	0.33	0.31	0.08	0.41	0.18

APPENDIX B

DESIGN CALCULATIONS

Appendix B : Design Calculations

B.1 Machine frame elements weight

The machine frame weight was calculated using Figure 4-2 and assumptions explained in Section 4.3 as follows:

"Two axle unit" frame beams length = (650 + 1000 + 1193) $\times$ 2 = 5686 mm

"Two axle unit" lateral beams = 1050 $\times$ 2 = 2100 mm

"Two axle unit" frame weight = ((5686 + 2100) / 1000) $\times$ 2 kg/m = 15.6 kg

"One axle unit" frame beams length = (550 + 300 + 627) $\times$ 2 = 2953 mm

"One axle unit" frame lateral beams = 1050 $\times$ 3 = 3150 mm

"One axle unit" frame weight = ((2953 + 3150) /1000) $\times$ 2 kg/m = 12.2 kg

The arm beam length = 700 + 800 + 1110 + 200 = 2810 mm

The arm weight = (2810/1000) $\times$ 2 kg/m =5.6 kg

Stabiliser frame length = 2$\times$ 1000 2000 mm

Stabiliser frame weight = (2000 / 1000) $\times$ 2 kg/m = 4 kg

Stabiliser wheels weight = 1 kg $\times$ 2 = 2 kg

Total stabiliser weight = 4 + 2 = 6 kg

B.2 Sprocket and chain design

The sprocket and chain transmission system of the machine (Section 4.7) was designed from the Fenner catalogue (Fenner 1988) by employing the following design steps:

a) Service factor of 1.1 from Table 1 of the Fenner catalogue

b) Design power = motor maximum power $\times$ service factor :

700 $\times$ 1.1 = 770 Watt

c) Referring to Table 2 of catalogue, intersection of motor running speed, 300 rev/min; and design power, 770 watt, a 9.5 mm pitch chain was indicated.

d) From table 4 of the Fenner catalogue, the size and number of teeth, t, was selected for the sprocket system shown in Figure 4-4. This system of two pairs of sprockets was selected on the basis of space limitations and available sprockets in the catalogue.

e) Sprocket reduction ratio = $(t_3 / t_1) \times (t_4 / t_2)$

sprockets reduction ratio = $(57 / 17) \times (57 / 17) = 11.42$

f) Reference to the power rating tables of the Fenner catalogue, show that the power rating of a single chain for a 17 teeth sprocket at 450 rev/min is 0.86 kW. Multiplication a factor of 0.9 converts the power rating from that of a 19 teeth sprocket to 17 teeth sprocket. This gives a power rating of 774 watt, which is greater than the design power requirement 770 W. Therefore the 9.5 mm pitch, single chain was strong enough and selected for sprockets t_1 and t_3.

g) The process was repeated for sprockets t_2 and t_4. The only difference was the input shaft speed which is now reduced from 300 rev/min by the ratio of t_1 / t_3 to 89.47 rev/min. This input speed suggests that a single sprocket with 12.7 mm pitch is acceptable for the sprockets t_2 and t_4 .

The summary of the results was presented in Table 4-4.

B.3 Driving shaft design

To design the driving shaft C the maximum bending moment in the shaft was calculated. To calculate this the 3-dimensional force diagram of Figure 4-7 was divided in to xy and xz plans (Beer and Johnston 1981). Following equations can be written in xy Plan, (Figure B-1).

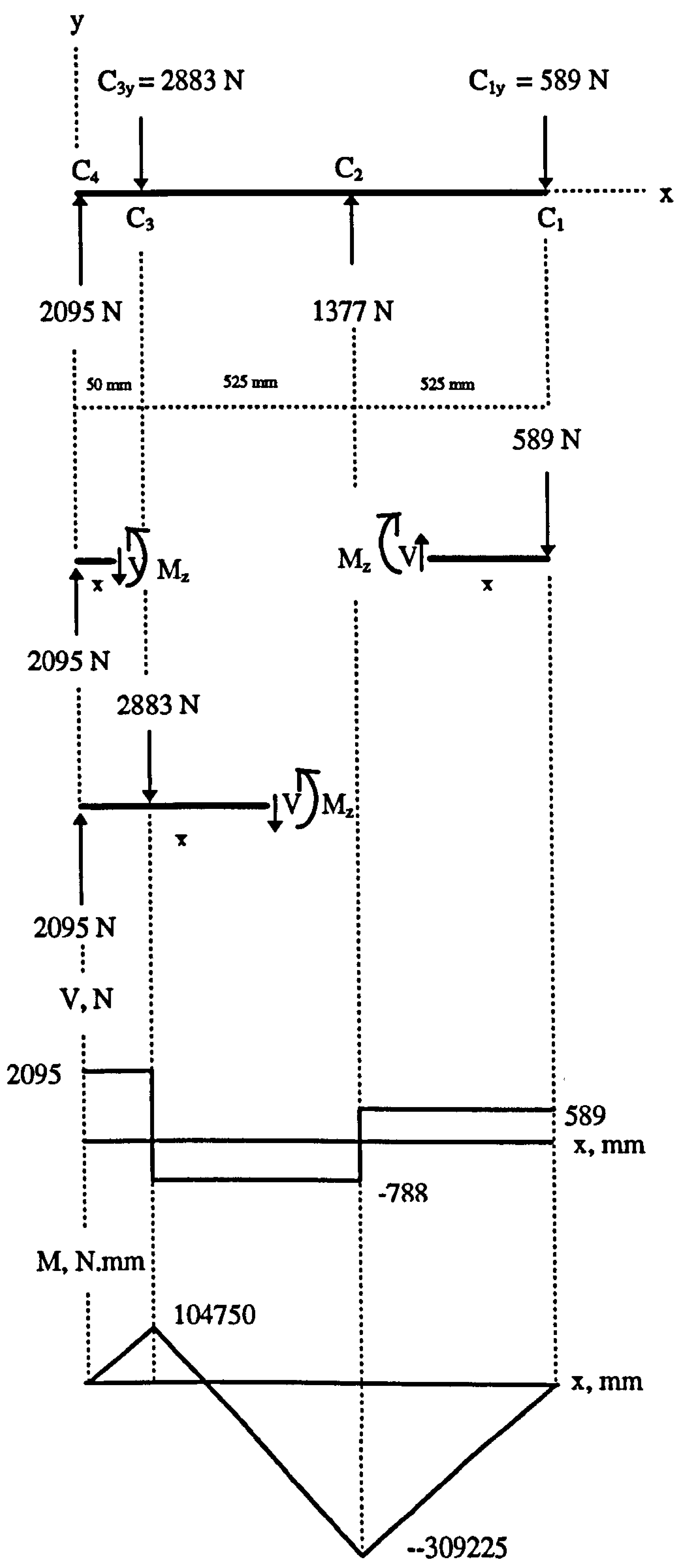

Figure B-1 Driving shaft free body, shear force and moment diagrams in zy plan

$$\sum M_{C3} = 0$$

$$-2095 \times C_4C_3 + 1377 \times C_3C_2 - C_{1y} \times C_3C_1 = 0$$

$$C_{1y} = (-2095 \times 50 + 1377 \times 525) / 1050 = 589 \text{ N}$$

$$\sum F_y = 0$$

$$2095 - C_{3y} + 1377 - 589 = 0$$

$$C_{3y} = 2883 \text{ N}$$

To find the maximum moment in plan xy the first interval of the shaft is analysed from the free body diagram of the section for $0 < x < 50$ mm. A vertical summation of forces and moment about the cut section yield

$$\sum F_y = 0 \qquad V = 2095 \text{ N}$$

$$\sum M = 0 \qquad M_z = 2095x$$

These values of V and M hold for $0 < x < 50$ mm are plotted for that interval in the shear and moment diagram shown. From the free body diagram of the section for which $50 < x < 575$ mm, equilibrium in the vertical direction and a moment sum about the cut section give

$$\sum F_y = 0 \qquad V = 2095 - 2883 = -732 \text{ N}$$

$$\sum M = 0 \qquad M_z = 2095x - 2883(x-50) = -732x + 144150$$

The last interval may be analysed by inspection. The shear is constant at 589 N and the moment follow a straight -line relation beginning with zero at the right end of the shaft.

the maximum moment, M_z ,occurs at point C2 with the value of :

$$M_z = -309225 \text{ N.mm}$$

The maximum moment in xz plan was derived from the force and moment diagram in Figure B-2.

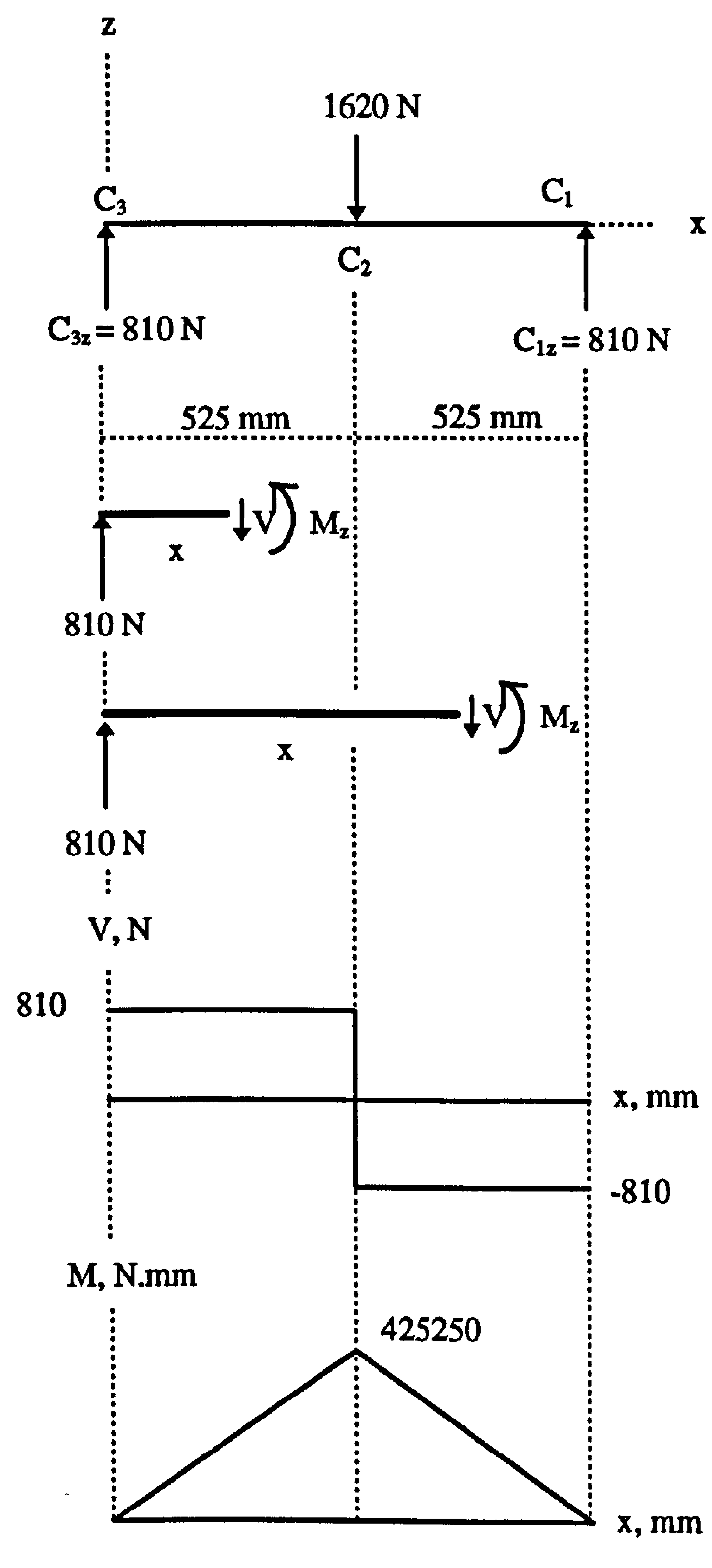

Figure B-2 Shaft free body, shear force and moment diagrams in xz Plan

Point C_2 is in the middle of C_1C_3 therefore, from the symmetry of force diagram:

$$C_{3x} = C_{1x} = 1620 / 2 = 810 \text{ N}$$

To plot shear force and moment diagrams for plan xz similar analysis of plan xy was used. Maximum moment, M_y, occurs at point C_2 with the value of:

$$M_y = 425250 \text{ N.mm}$$

The total maximum moment at point C2 is the summation of its two components:

$$M = \sqrt{M_z^2 + M_y^2}$$

$$M = \sqrt{309225^2 + 425250^2}$$

$$M = 525792 \text{ N.mm}$$

this moment was used to calculate the shaft diameter.

Driving shaft diameter

To find the yield stress of the 35 mm outside and 25 mm inside diameter shaft to fit the wheel hub, The following equations by Spotts (1978) where used:

$$\sigma_{yp} = 2 f_s \tau_{max} \qquad \text{(B.1)}$$

Where σ_{yp} is the shaft yield stress, MPa and f_s the safety factor of 1.5.

$$\tau_{max} = \frac{r_o}{J} \sqrt{(C_M M)^2 + (C_T T)^2} \qquad \text{(B.2)}$$

Where τ_{max} is maximum shear stress in the shaft in MPa; r_o is shaft outside diameter in mm; C_M is moment coefficient, 1.5; M is maximum moment in the shaft in N.mm; C_T is torque coefficient, 1; T is maximum torque in the shaft, N.mm.

$$J = \frac{\pi}{32}(d_o^4 - d_i^4)$$

Where d_o, 35 mm, is shaft outside diameter in mm; d_i, 25 mm is shaft inside diameter in mm; Substituting values gives:

$$J = \frac{\pi}{32}(35^4 - 25^4)$$

$$J = 108974 \text{ mm}^4$$

substituting values for M from Section B.3 and T_m for T from Section 7.4.2 in Equation (B.3) gives:

$$\tau_{max} = \frac{17.5}{108974}\sqrt{(1.5\times525792)^2 + (1\times240975)^2}$$

$$\tau_{max} = 132 \text{ MPa}$$

Substituting the value in equation (B.1) gives:

$$\sigma_{yp} = 2 \times 1.5 \times 132$$

$$\sigma_{yp} = 396 \text{ MPa}$$

This level of stress lies in the range of commercial shafts strength, therefore the selected diameters of the shaft were valid and 396 MPa was the shaft selection criterion.

B.4 Bearing selection

Each driving shaft is supported by two bearings as explained in Section 4.7.3. Bearing selection was based on the basic dynamic load rating, C (SKF 1994). To find C the following equation was used:

$$L_{10h} = \frac{1000000}{60n}\left(\frac{C}{P}\right)^p \qquad \text{(B.3)}$$

Where L_{10h} is basic rating, operating hours (3000 hr for agricultural machinery); n is the shaft speed, rev/min; C is basic dynamic load rating, N; P is equivalent dynamic bearing load, N; p is exponent of life equation (3 for ball bearings).

The shaft speed, n was calculated from the next equation:

$$n = \frac{60\,V}{\pi d}$$

Where V is climbing speed, 0.41 m/sec; d is tyre diameter, 0.35 m.

Substituting values gives:

$$n = \frac{60 \times 0.41}{\pi \times 0.35}$$

$$n = 22.37 \text{ rev/min}$$

For the time that the axial force on the bearing is zero SKF suggests the following equation for P:

$$P = F_r$$

Where F_r is radial force on the bearing, N.

The maximum forces are on the bearing positioned at point C_3 in Figure 4-5, and the maximum radial force on the bearing at C_3 is equal to the sum of its components:

$$F_r = \sqrt{C_{3z} + C_{3y}}$$

$$F_r = \sqrt{810^2 + 2883^2}$$

$$F_r = 2995 \text{ N}$$

substituting values for parameters in equation (B.3) gives:

$$3000 = \frac{1000000}{60 \times 22.37} \left(\frac{C}{2995} \right)^3$$

$$C = 4764 \text{ N}$$

Based on this value of C, a 35 mm inside diameter self aligning ball bearing with adapter sleeve and SHN-plummer block housing with designation number of 1208 EK was selected.

B.5 Frame design

The design of the frame was based on the critical load locations, where the motor and gear boxes were mounted,

"Method of joints" and "Method of sections" (Meriam and Craige 1993) were used to determine unknown forces in the frame. Sections were checked for the maximum bending moment and shear stress.

The plane stress analysis was used in combined stress conditions. The shear stress caused by torsion and transverse loading were calculated and mentioned in the design.

The critical loading was on link pj in Figure 4-8, where the bearing C3 was mounted, and the maximum force from the wheels and transmission system were applied to the link. With the assumption that link pj is a straight beam, different hollow square sections were evaluated to establish the minimum suitable dimensions. Value of C_{3y} was brought from Section B.3.

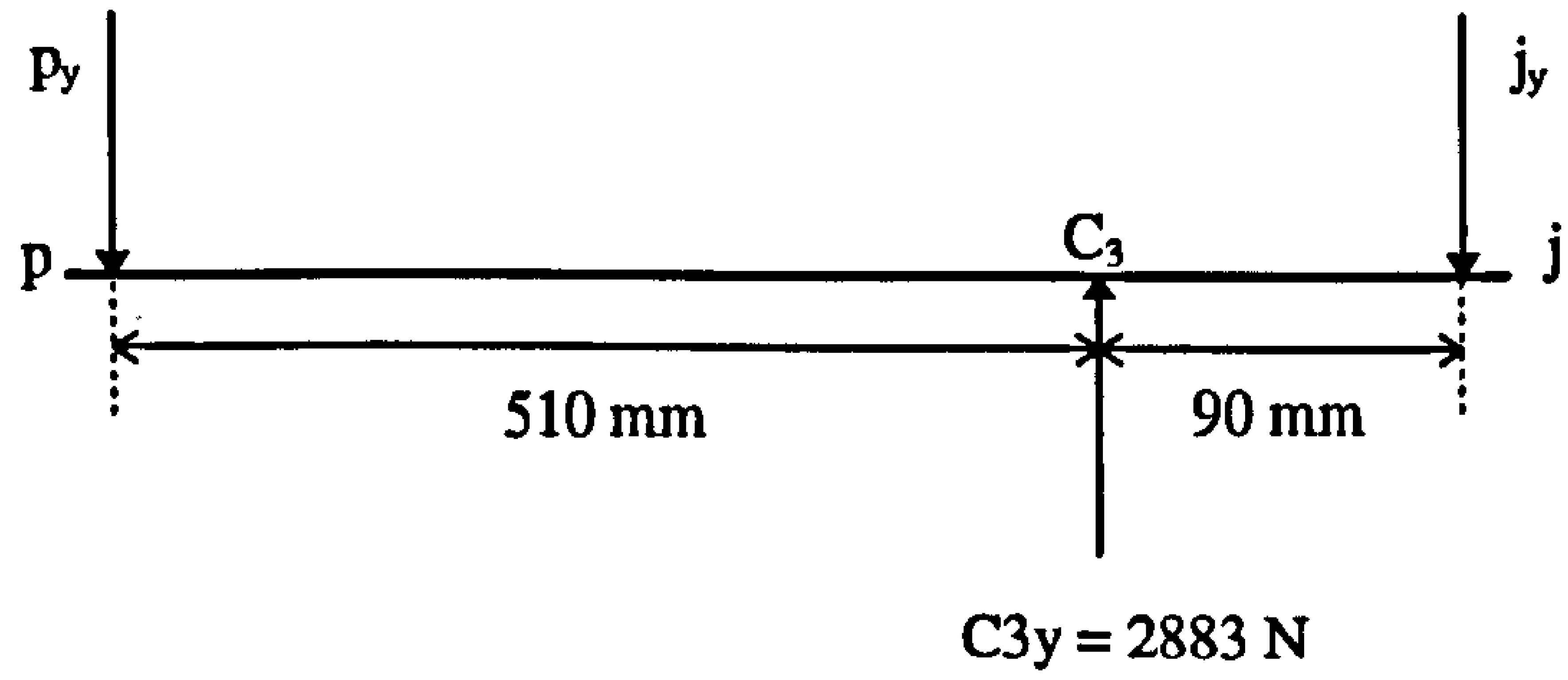

Figure B-3 Free body diagram for "the one axle unit" frame link, pj

Results are summarised in Table B- 1. Maximum bending stress in the beam was calculated from the following equation:

$$\sigma_b = \frac{MC_o}{I}$$

Where σ_b is maximum bending stress, MPa; M is maximum bending moment in the beam N.mm. C_o is maximum distance from the neutral axis. I is area moment of inertia, mm^4 ,for a square hollow section:

$$I = \frac{1}{12}(d_o^2 - d_i^2)$$

Where d_o is hollow square outside diameter, mm. d_i is hollow square inside diameter, mm.

To calculate M:

$$\Sigma M_p = 0$$

$$C_{3y} \times 510 = j_y \times 600$$

$$2883 \times 510 = j_y \times 600$$

$$j_y = 2451 \text{ N}$$

The maximum moment occurs at point C_3:

$$M = j_y \times 90 \text{ or } M = 220590 \text{ N.mm}$$

Table B- 1 Section selection for the frame

No.	Cross section, mm	C_o , mm	M, N mm	I, mm^4	σ_b , MPa
1	$20 \times 20 \times 2$	10	220590	7872	280
2	$20 \times 20 \times 2.5$	10	220590	9114	242
3	$25 \times 25 \times 2$	12.5	220590	16345	169
4	$25 \times 25 \times 2.5$	12.5	220590	19218	143

Assuming 166 MPa allowable stress for mild steel, including a factor of safety of 1.5, the minimum section dimensions are $25 \times 25 \times 2.5$ mm (row 4 in Table B- 1).

Most frame elements were made from this section for considerations of ease of manufacture.

APPENDIX C

DATA ANALYSIS AND METHODS

Appendix C : Data Analysis and Methods

C.1. Data analysis of the ground tests

This Section explains the analysis technique of the data recorded from the machine sensors as explained in Section 5.4.9.1. The data belong to each test run was saved on the hard disc under Snap Master software directory with a serial number and file extension name of "smb". These files were analysed by Excel software. To import them to the Excel spread sheet this sequence was followed:

1. Open the Excel spread sheet
2. Open a new book
3. Open all "smb" files belong to different levels of torque for each speed (e.g. n_2 replication 1) in Excel spread sheet
4. Copy the required columns of power, speed and torque from these files in to the clipboard
5. Paste these data to the new book which is opened in step 2

Now this new book contains all information belonging to different levels of torque for each speed setting. To find the file easily for further processing it was saved as an Excel file. For example the file belonging to the first replication of n_2 it was saved as $1n_2$.xls.

Imported data to Excel contain three columns, power, torque and speed. The power versus torque curves will be extracted from these data.

The sampling rate was 35 per second and data were recorded for five second, therefore, 175 data points were available for one level of torque. the average of 175 data points was calculated and this number is the representative of the power, torque or speed for that replication. The procedure was repeated for the second and third replication.

The average of the three replications is the number that represents the amount of power, torque or speed at that level. Table C-1 to Table C-5 show the processed voltage values from power, torque and speed sensors at five motor speeds settings n_2, n_4, n_6, n_8 and n_{10} through the controller (PLC). Using Torque, speed and power calibration curves explained in Sections 5.3 these voltage values were converted to the true values and the results were used to evaluate the machine performance.

Table C-1 Out put voltage of sensors at different torque levels and motor speed n_2, volt

Replication	Torque level	Sensor		
		Power	Speed	Torque
1	1	0.136	0.630	1.731
	2	0.114	0.711	1.551
	3	0.095	0.797	1.377
	4	0.067	0.887	1.051
	5	0.027	0.913	0.551
	6	0.003	1.007	0.000
2	1	0.118	0.810	1.531
	2	0.122	0.820	1.567
	3	0.106	0.872	1.420
	4	0.082	0.906	1.172
	5	0.050	0.933	0.818
	6	0.031	0.945	0.572
3	1	0.126	0.648	1.648
	2	0.109	0.722	1.482
	3	0.088	0.780	1.279
	4	0.057	0.816	0.930
	5	0.058	0.840	0.926
	6	0.033	0.873	0.609
	7	0.001	0.968	0.000

Table C-2 Output voltage of sensors at different torque levels and motor speed n_4, volt

Replication	Torque level	Sensor		
		Power	Speed	Torque
1	1	0.201	1.714	1.949
	2	0.191	1.714	1.856
	3	0.048	1.844	0.644
	4	0.123	1.794	1.308
	5	0.010	1.925	0.000
2	1	0.011	2.009	0.000
	2	0.017	1.937	0.312
	3	0.134	1.834	1.299
	4	0.175	1.794	1.634
	5	0.155	1.834	1.512
	6	0.158	1.764	1.534
3	1	0.039	1.901	0.504
	2	0.078	1.881	0.865
	3	0.138	1.838	1.343
	4	0.146	1.803	1.425
	5	0.155	1.767	1.514
	6	0.188	1.679	1.753

Table C-3 *Output voltage of sensors at different torque levels and motor speed n_6, volt*

Replication	Torque level	Sensor		
		Power	Speed	Torque
1	1	0.006	2.932	0.000
	2	0.074	2.763	0.577
	3	0.299	2.438	1.865
	4	0.310	2.357	1.959
	5	0.148	2.619	1.132
	6	0.148	2.619	1.132
	7	0.258	2.444	1.732
2	1	0.006	2.900	0.000
	2	0.054	2.844	0.505
	3	0.115	2.769	0.948
	4	0.181	2.689	1.333
	5	0.237	2.532	1.669
	6	0.259	2.450	1.769
	7	0.135	2.671	1.083
3	1	0.022	2.748	0.228
	2	0.054	2.736	0.493
	3	0.082	2.693	0.717
	4	0.123	2.684	0.976
	5	0.222	2.589	1.593
	6	0.262	2.528	1.807
	7	0.249	2.522	1.742
	8	0.248	2.491	1.739
	9	0.281	2.232	1.977

Table C-4 Output voltage of sensors at different torque levels and motor speed n_8, volt

Replication	Torque level	Sensor		
		Power	Speed	Torque
1	1	0.003	3.871	0.000
	2	0.109	3.728	0.699
	3	0.196	3.701	1.194
	4	0.247	3.493	1.559
	5	0.286	3.250	1.795
	6	0.299	3.152	1.840
2	1	0.002	3.872	0.000
	2	0.098	3.750	0.635
	3	0.259	3.387	1.585
	4	0.313	3.102	1.902
	5	0.319	3.047	1.930
	6	0.333	2.972	1.979
3	1	0.002	3.886	0.000
	2	0.118	3.751	0.670
	3	0.202	3.329	1.205
	4	0.233	3.157	1.399
	5	0.275	2.902	1.676
	6	0.324	2.558	1.965

Table C-5 Output voltage of sensors at different torque levels and motor speed n_{10}, volt

Replication	Torque level	Sensor		
		Power	Speed	Torque
1	1	0.005	4.869	0.000
	2	0.008	4.846	0.031
	3	0.057	4.600	0.287
	4	0.127	4.169	0.740
	5	0.214	3.701	1.262
	6	0.277	3.343	1.664
	7	0.298	3.203	1.784
	8	0.341	2.925	1.992
	9	0.108	3.914	0.825
2	1	0.309	2.733	2.009
	2	0.315	2.738	2.023
	3	0.306	2.811	1.996
	4	0.252	3.085	1.659
	5	0.197	3.388	1.265
	6	0.115	3.832	0.671
	7	0.005	4.530	0.000
	8	0.128	3.056	1.071
3	1	0.005	4.684	0.000
	2	0.111	3.524	0.925
	3	0.069	4.394	0.354
	4	0.142	3.984	0.835
	5	0.199	3.697	1.182
	6	0.302	3.225	1.804
	7	0.350	2.978	2.082
	8	0.307	2.163	2.216

C.2. Data analysis of slippage and power tests

To analyse the data of slippage and power tests explained in Section 5.4.10.1 the slippage value for each test was recorded from the counter display fitted on the machine after each run. It was then converted to the true value using Equation 5.6 and put in to the relevant test run cell of Table 5-12 prepared for slippage tests.

To analyse the power and speed sensor data recorded by computer, the data were imported to the Excel spread sheet. The process is similar to the ground test data analysis in Section C-1 but these data contain only two columns instead of three in ground tests which are power and speed columns. The torque measurement was not required because it was calibrated against the power sensor in ground tests and can be calculated from power sensor readings and using ground test curves in Section 6.5.

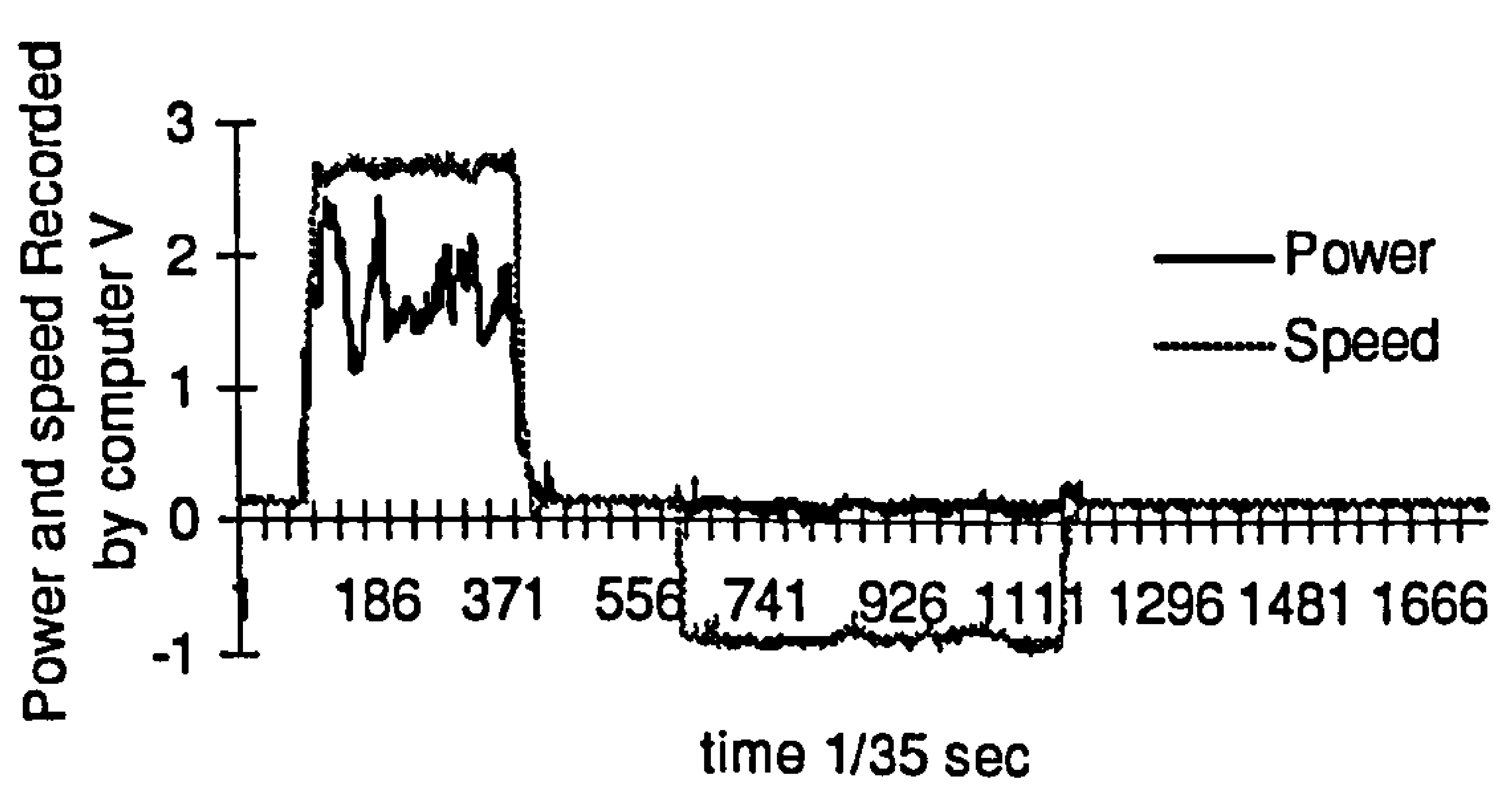

Figure C-1 Sample curve of Power and motor speed against the time. 2nd replication of 20 kg payload, wheel spacing, D1= 300 mm, Spring tension T1= 555 N

The data were recorded with the sampling rate of 35 per second and for a period of 50 sec for each run. Two columns of the row data including power sensor and motor speed sensor voltage made from 35×50 =1750 points were plotted against the time in Excel spread sheet to check the data for continuity and noise before the analysis. A sample curve is shown in Figure C-1. The positive part of the speed curve shows the climbing period and the negative part shows the landing part of a run. The landing part is negative because the speed sensor rotates in opposite direction and the polarity of the voltage changes. The power curve variations is caused by the uneven tree surface. Finally when the curve was checked the average value of the data points for the climbing part of the run (not landing) were calculated, the result is one number for speed voltage and one number for power sensor voltage representing the power and the motor speed for that run. These numbers fill one cell of table 5-12 prepared for speed and power and the process should be repeated for other settings to complete the table. These values were converted to the true values of torque speed and power using calibration curves in Section 5.3. The true values then were converted to the required power and efficiencies using equations explained in Chapter 6.

For each slippage, power and efficiency test a table similar to Table 5-12 was completed. the data then were rearranged in new tables to be analysed by MSTATC (statistical analysis software). These tables and results are presented in the following Section and the graphical presentation of the results is in Chapter 6.

C.3. Processed data and statistical analysis of slippage and power tests

To study the effect of wheel spacing, spring tension and payload on the machine performance a factorial analysis of variance was carried out on the test results. The results were analysed using Experiment Model Number 13 of MSTATC statistical analysis software. This model is a Randomised Complete Block Design for Factor A (wheel spacing), with Factor B (spring tension) as a Split Plot on A and Factor C (payload) as a Split Plot on B which fits the test design shown in Table 5-12. The following variables and replications were adopted for the factorial analysis of variance:

Replication (Var 1: replication) with values from 1 to 3.

Factor A (Var 2: wheel spacing, D) with values from 1 to 2.

Factor B (Var 3: spring tension, T) with values from 1 to 2.

Factor C (Var 4: Load) with values from 1 to 4.

Variable 5: slippage for wheel slippage test, power for power tests and efficiency for efficiency tests.

The data table and the resultant analysis of variance table for each test is presented in the following tables. D1T1, D1T2, D2T1 and D2T2 represent the setting of wheel spacing and spring tension. The results are discussed in Chapter 6.

Table C-6 wheel slippage, %

Rep	Payload, kg	D1T1	D1T2	D2T1	D2T2
1	0	14.6	11.0	23.7	17.9
2	0	12.8	9.1	20.9	19.4
3	0	11.0	11.0	17.9	16.3
1	20	17.9	12.8	25.1	19.4
2	20	12.8	12.8	23.7	17.9
3	20	14.6	11.0	17.9	16.3
1	40	17.9	12.8	27.6	19.4
2	40	14.6	12.8	20.9	17.9
3	40	14.6	12.8	19.4	19.4
1	60	19.4	14.6	30.0	19.4
2	60	14.6	12.8	23.7	17.9
3	60	14.6	14.6	20.9	19.4
1	80		16.3		19.4
2	80		16.3		22.3
3	80		14.6		22.3

Table C-7 Analysis of variance tablefor wheel slippage

No.	Source	Degrees of freedom	Sum of squares	Mean square	F value	Prob
1	Replication	2	90.9	45.4	7.12	0.12
2	Factor A	1	565.8	565.8	88.70	0.01
3	Error	2	12.8	6.4		
4	Factor B	1	141.4	141.4	10.51	0.03
5	AB	1	8.2	8.2	0.61	
6	Error	4	53.9	13.5		
7	Factor C	3	58.0	19.3	14.56	0.00
8	AC	3	3.5	1.2	0.87	
9	BC	3	3.0	1.0	0.76	
10	ABC	3	4.8	1.6	1.21	0.33
11	Error	24	31.9	1.3		
	Total	47	974.1			

Table C-8 Input electric power, watts

replication	Payload, kg	D1T1	D1T2	D2T1	D2T2
1	0	975.2	1021.6	910.6	1013.6
2	0	968.7	1039.8	931.7	1007.1
3	0	898.3	1049.9	971.6	999.9
1	20	1066.6	1086.9	1007.8	1099.2
2	20	1054.3	1106.5	1013.6	1097.1
3	20	1057.2	1110.1	1057.9	1085.5
1	40	1160.2	1180.5	1120.3	1187.0
2	40	1143.5	1197.2	1113.0	1192.8
3	40	1137.0	1207.3	1163.8	1214.6
1	60	1260.3	1269.0	1193.5	1284.9
2	60	1233.4	1271.2	1229.8	1295.1
3	60	1221.8	1273.3	1255.9	1295.1
1	80		1376.3		1365.5
2	80		1374.2		1412.6
3	80		1362.6		1380.0

Table C-9 Analysis of variance table for input electric power

No.	Source	Degrees of freedom	Sum of squares	Mean square	F value	Prob
1	Replication	2	2304.1	1152.0	0.43	
2	Factor A	1	428.4	428.4	0.16	
3	Error	2	5402.7	2701.4		
4	Factor B	1	37643.1	37643.2	14.67	0.02
5	AB	1	103.8	103.8	0.04	
6	Error	4	10265.2	2566.3		
7	Factor C	3	502084.4	167361.5	477.48	0.00
8	AC	3	960.7	320.2	0.91	
9	BC	3	2719.8	906.6	2.59	0.08
10	ABC	3	1380.8	460.3	1.31	0.29
11	Error	24	8412.2	350.5		
	Total	47	571705.3			

Table C-10 Axle power, watts

Rep	Payload, kg	D1T1	D1T2	D2T1	D2T2
1	0	416.4	429.6	409.3	442.8
2	0	417.9	433.3	413.1	441.2
3	0	398.0	431.6	430.5	427.4
1	20	439.7	449.6	433.3	453.4
2	20	434.0	437.6	441.9	456.3
3	20	436.8	441.4	449.0	445.7
1	40	455.7	449.2	457.5	469.1
2	40	451.1	460.3	463.1	467.5
3	40	451.9	458.4	465.1	474.1
1	60	480.9	469.1	454.5	467.7
2	60	459.4	466.3	477.8	479.6
3	60	455.6	461.1	482.3	477.9
1	80		476.2		479.6
2	80		470.3		490.1
3	80		469.7		473.6

Table C-11 Analysis of variance tablefor axle power

No.	Source	Degrees of freedom	Sum of squares	Mean square	F value	Prob
1	Replication	2	16.3	8.2	0.03	
2	Factor A	1	793.0	793.0	3.36	0.21
3	Error	2	472.0	236.0		
4	Factor B	1	965.7	965.7	7.10	0.06
5	AB	1	25.7	25.7	0.19	
6	Error	4	543.7	135.9		
7	Factor C	3	14231.9	4744.0	118.0	0.00
8	AC	3	53.4	17.8	0.44	
9	BC	3	558.0	186.0	4.63	0.01
10	ABC	3	18.7	6.2	0.15	
11	Error	24	964.9	40.2		
	Total	47	18643.3			

Table C-12 Climbing power, watts

Replication	Payload, kg	D1T1	D1T2	D2T1	D2T2
1	0	286.5	294.3	258.7	275.8
2	0	291.7	298.0	264.4	276.6
3	0	295.2	291.7	277.4	278.2
1	20	322.4	335.1	287.9	303.7
2	20	328.1	322.4	294.8	309.7
3	20	328.1	327.2	314.1	317.7
1	40	353.0	363.3	322.7	340.4
2	40	361.2	366.5	335.8	340.4
3	40	356.0	363.3	341.3	341.3
1	60	385.1	404.2	318.5	366.7
2	60	396.1	395.0	355.4	373.6
3	60	387.3	395.0	373.6	369.7
1	80		435.0		381.6
2	80		424.1		390.4
3	80		432.5		389.4

Table C-13 Analysis of variance table for climbing power

No.	Source	Degrees of freedom	Sum of squares	Mean square	F value	Prob
1	Replication	2	622.2	311.1	1.03	0.49
2	Factor A	1	7708.9	7708.9	25.47	0.04
3	Error	2	606.6	303.3		
4	Factor B	1	958.5	958.5	5.08	0.9
5	AB	1	146.6	146.6	0.78	
6	Error	4	754.4	188.6		
7	Factor C	3	59928.0	19976.0	613.06	0.00
8	AC	3	23.2	107.7	3.31	0.04
9	BC	3	134.0	44.7	1.37	0.28
10	ABC	3	64.1	22.0	0.66	
11	Error	24	782.0	33.0		
	Total	47	72028.9			

Table C-14 Tractive efficiency, %

Rep	Payload, kg	D1T1	D1T2	D2T1	D2T2
1	0	68.8	68.5	63.2	62.3
2	0	69.8	68.8	64.0	62.7
3	0	74.2	67.6	64.4	65.1
1	20	73.3	74.5	66.4	67.0
2	20	75.6	73.7	66.7	67.9
3	20	75.1	74.1	70.0	71.3
1	40	77.5	80.9	70.5	72.6
2	40	80.1	79.6	72.5	72.8
3	40	78.8	79.3	73.4	72.0
1	60	80.1	86.2	70.1	78.4
2	60	86.2	84.7	74.4	77.9
3	60	85.0	85.7	77.5	77.3
1	80		91.3		79.6
2	80		90.2		79.7
3	80		92.1		82.2

Table C-15 Analysis of variance tablefor tractive efficiency

No.	Source	Degrees of freedom	Sum of Squares	Mean square	F value	Prob
1	Replication	2	33.0	16.5	10.59	0.09
2	Factor A	1	600.0	600.0	384.54	0.00
3	Error	2	3.1	1.6		
4	Factor B	1	4.9	4.9	0.78	
5	AB	1	3.6	3.6	0.57	
6	Error	4	25.0	6.2		
7	Factor C	3	1272.9	424.3	220.43	0.00
8	AC	3	17.6	5.9	3.06	0.05
9	BC	3	34.0	11.3	5.89	0.00
10	ABC	3	3.8	1.3	0.65	
11	Error	24	46.2	1.9		
	Total	47	2044.1			

Table C-16 Axle efficiency, %

replication	Payload, kg	D1T1	D1T2	D2T1	D2T2
1	0	42.7	42.1	44.9	43.7
2	0	43.1	41.7	44.3	43.8
3	0	44.3	41.1	44.3	42.8
1	20	41.2	41.4	43.0	41.2
2	20	41.2	39.5	43.6	41.6
3	20	41.3	39.8	42.4	41.1
1	40	39.3	38.1	40.8	39.5
2	40	39.4	38.4	41.6	39.2
3	40	39.7	38.0	40.0	39.0
1	60	38.2	37.0	38.1	36.4
2	60	37.2	36.7	38.9	37.0
3	60	37.3	36.2	38.4	36.9
1	80		34.6		35.1
2	80		34.2		34.7
3	80		34.5		34.3

Table C-17 Analysis of variance tablefor axle efficiency

No.	Source	Degrees of freedom	Sum of squares	Mean square	F value	prob
1	Replication	2	0.7	0.4	0.54	
2	Factor A	1	14.7	14.7	21.81	0.04
3	Error	2	1.3	0.7		
4	Factor B	1	22.1	22.1	79.55	0.00
5	AB	1	0.3	0.3	1.08	0.36
6	Error	4	1.1	0.3		
7	Factor C	3	230.5	76.8	316.08	0.00
8	AC	3	1.6	0.6	2.17	0.12
9	BC	3	0.0	0.0	0.03	
10	ABC	3	1.0	0.3	1.37	0.28
11	Error	24	5.8	0.2		
	Total	47	279.3			

Table C-18 Total efficiency, %

replication	Payload, kg	D1T1	D1T2	D2T1	D2T2
1	0	29.4	28.8	28.4	27.2
2	0	30.1	28.7	28.4	27.5
3	0	32.9	27.8	28.5	27.8
1	20	30.2	30.8	28.6	27.6
2	20	31.1	29.1	29.1	28.2
3	20	31.0	29.5	29.7	29.3
1	40	30.4	30.8	28.8	28.7
2	40	31.6	30.6	30.2	28.5
3	40	31.3	30.1	29.3	28.1
1	60	30.6	31.9	26.7	28.5
2	60	32.1	31.1	28.9	28.9
3	60	31.7	31.0	29.8	28.5
1	80		31.6		27.9
2	80		30.9		27.6
3	80		31.7		28.2

Table C-19 Analysis of variance tablefor total efficiency

No.	Source	Degrees of freedom	Sum of squares	Mean square	F value	Prob
1	Replication	2	2.6	1.3	3.65	0.21
2	Factor A	1	46.2	46.2	132.28	0.01
3	Error	2	0.7	0.3		
4	Factor B	1	7.9	7.9	4.35	0.10
5	AB	1	0.4	0.4	0.21	
6	Error	4	7.3	1.8		
7	Factor C	3	10.1	3.4	7.81	0.00
8	AC	3	3.3	1.1	2.53	0.08
9	BC	3	4.2	1.4	3.24	0.04
10	ABC	3	1.4	0.5	1.11	0.36
11	Error	24	10.3	0.4		
	Total	47	94.4			

APPENDIX D

CALCULATIONS OF MACHINE COSTS

Appendix D : Calculations of Machine Costs

This appendix includes the detail machine cost calculations where the results were used in Chapter 7. Method and equations were adopted from Witney (1988) and Hunt (1973).

D.1 Annual machine costs

The annual machine cost is defined as:

$$MCY = MFC + MRC \qquad (D.1)$$

where MCY is annual machine cost, Rials. MFC is annual machine fix cost, Rials. MRC is annual machine running cost, Rials.

The main machine fixed costs, MFC, are defined as:

$$MFC = DE + IC \qquad (D.2)$$

Where DE is depreciation and IC is interest charge.

The straight - line depreciation was calculated from the next formula:

$$DE = (PP - SV) / LI \qquad (D.3)$$

Where DE is average annual depreciation, Rials. PP is purchase price, Rials. SV is resale value, Rials. LI is period of ownership , year.

Substituting SV = 0.10 PP and life of 10 years in Equation D.3 gives:

$$DE = 0.09PP \qquad (D.4)$$

The interest charge when the straight - line annual depreciation is used is equal to:

$$IC = i_n \times (PP + SV) / 200 \qquad (D.5)$$

Where IC is interest charge, Rials. i_n is net interest rate, % and is defined as:

$$i_n = i_i - j \qquad (D.6)$$

Where i_i is investment interest rate, % and j is Inflation rate, %.

If $i_i = 14\ \%$ and $j = 7\ \%$ then $i_n = 7\ \%$. Substituting i_n and SV equal to 0.1PP in Equation D.5 gives:

$$IC = 0.039PP \tag{D.7}$$

Substituting Equations D.7 and D.4 in Equation D.2 gives the annual fixed cost as follows:

$$MFC = 0.129PP \tag{D.8}$$

The machine running costs vary in proportion to the utilisation of the machine and comprise:

$$MRC = \text{fuel and oil} + \text{repairs} + \text{labour} \tag{D.9}$$

The fuel price is 180 Rials per litre and the generator consumes 1 litre per hour, therefore, the fuel cost would be 180 Rials per hour.

The oil price is 500 Rials per litre and for generator capacity of 1 litre and oil changing interval of 50 hours, the oil cost would be 10 Rials per hour, thus, the total fuel and oil cost is 190 Rials per hour.

The main repair cost for the machine is the cost of new tyres and batteries. The accumulated repair costs estimated to be 50 % of the purchase price for the machine life, so the annual cost will be 5 % of the PP for the machine life of 10 years.

The full time labour charge is 10 000 Rials per day and considering 8 working hours per day the labour cost is 1 250 Rials per hour.

Substituting calculated values of Running cost elements in Equation D.9 gives:

$$MRC = (\ 190\ \text{Rials/hr} \times MAU\ ) + 0.05PP + \ (1250\ \text{Rials/ hr} \times MAU)$$

or

$$MRC = 1440 \times MAU + 0.05PP \tag{D.10}$$

Where MAU is machine annual use in hour and is defined as follows:

$$MAU = MHT \times NT \tag{D.11}$$

Where MHT is machine harvesting time in hours and NT is the minimum number of trees to be harvested annually.

Substituting Equation D.11 in D.10 gives:

$$MRC = 1440\ (MHT \times NT) + 0.05PP \qquad (D.12)$$

Equation D.12 and D.8 in Equation D.1 gives the annual cost as follows:

$$MHC = 0.129PP + (1440\ (MHT \times NT) + 0.05PP) \qquad (D.13)$$

Rewriting Equation 7.2:

$$NT = \text{annual machine cost} / \text{hand harvesting cost per tree}$$

Substituting Equation D.13 for annual machine cost in Equation 7.2 and 3 000 Rials for the hand harvesting cost per tree, explained in Chapter 7, then, the minimum number of trees can be calculated as follows:

$$NT = (0.129PP + (\ 1440\ (MHT \times NT) + 0.05PP\ ))\ /\ 3000$$

Solving the equation for NT and converting the pp to million Rials gives:

$$NT = 3580.2PP\ /\ (60 - 0.48MHT) \qquad (D.14)$$

Equation D.14 calculates the minimum number of trees, NT, to be harvested for the economical justification of the machine (i.e. mechanised harvesting costs equal to hand harvesting costs) as a function of PP, machine purchase price in million Rials, and MHT, machine harvesting time (required time to harvest one tree) in minutes.

D.2 Machine main parts price

Table D.1 shows the price of machine main material and parts (supplied in UK) used to estimate the purchase price in Chapter 7.

Table D.1 Machine material and parts purchase prices

No.	Name of the part	No. off	Total cost £
1	main wheel	6	120
2	Stabiliser wheel	2	20
3	Motor	2	218
4	Controller (PLC)	2	168
5	Main wheels shaft	3.6 m	94
6	Hand winch	2	60
7	spring	2	33
8	Iron sheaves	2	32
9	Shaft bearings	6	60
10	Chain drive system	2	150
11	Frame steel and bolts		70
12	Bunch cutting unit	1	200
	Total		1225

APPENDIX E

LIST OF MATERIALS AND DRAWINGS

Appendix E : List of Materials and Drawings

List of materials used to made the experimental machine are presented in this Appendix. It also includes drawings of the machine parts. Table E-1 shows list of material used to make the machine.

Drawing No. 1 shows the three views of the date harvester. Full drawing of the machine parts drown by ACAD 13 are presented in two attached floppy diskettes.

Table E-1 List of materials used to make the experimental machine

No.	Part name	Specification	Required unit	No. off	Supplier
	Two axle unit chassis steels:				
1	Hollow square	25 × 25 × 2.5 mm	611 mm	2	
2	Hollow square	25 × 25 × 2.5 mm	1042 mm	2	
3	Flat	25 × 3 mm	68 mm	4	
4	Plate	2 mm	148 × 163 mm	4	
5	Flat	40 × 3 mm	68 mm	1	
6	Bush	18 × 8 mm	25 mm	2	
7	Hollow square	25 × 25 × 2.5 mm	924 mm	2	
8	Flat	25 × 2 mm	650 mm	2	
9	Hollow square	25 ×25 × 2.5 mm	1184 mm	2	
10	Flat	40 × 3 mm	61 mm	4	
11	Hollow square	25 × 25 × 2.5 mm	85 mm	1	
12	Hollow square	25 × 25 × 2.5 mm	75 mm	1	
13	Hollow square	25 × 25 × 2.5 mm	254 mm	2	
	One axle unit chassis steel:				
14	Hollow square	25 × 25 × 2.5 mm	432 mm	2	
15	Hollow square	25 × 25 × 2.5 mm	397 mm	2	
16	Flat	40 × 3 mm	76 mm	1	
17	Flat	40 ×3 mm	62 mm	1	
18	Plate	3 mm	200 × 160 mm	2	
19	Hollow square	25 × 25 × 2.5 mm	99 mm	1	
20	Hollow square	25 × 25 ×2 mm	150 mm	1	
21	Plate	3 mm	79 × 73 mm	1	
22	Flat	40 × 3 mm	81 mm	1	
23	Hollow square	25 × 25 × 2.5 mm	207 mm	1	
24	Hollow square	25 × 25 × 2.5 mm	187 mm	1	
25	Hollow square	25 × 25 × 2.5 mm	310 mm	2	
26	Hollow square	25 × 25 × 2.5 mm	1050 mm	3	
27	Hollow square	25 × 25 × 2.5 mm	236 mm	1	
	Pivot:				
28	Plate	2.5 mm	198 × 160 mm	2	
29	Flat	40 × 3 mm	40 mm	1	
30	Flat	25 ×3 mm	175 mm	1	
31	Flat	25 × 3 mm	115 mm	1	
32	Flat	25 × 3 mm	367 mm	1	

Table E -1 List of materials used to make the experimental machine (continued)

No.	Part name	Specification	Required unit	No. off	Supplier
	stabiliser chassis steels:				
33	Hollow square	25 × 25 × 2.5 mm	435 mm	2	
34	Flat	25 × 2.5 mm	75 mm	4	
35	Bush	18 × 3 mm	10 mm	4	
36	Flat	25 × 2.5 mm	73 mm	4	
37	Bush	18 × 3	25 mm	2	
38	Plate	2 mm	137 × 141 mm	2	
39	Hollow square	25 × 25 × 2.5	455 mm	2	
	Harvesting arm steels:				
40	Hollow square	25 × 25 × 2.5 mm	1100 mm	1	
41	Hollow square	25 × 25 × 2.5 mm	701 mm	1	
42	Hollow square	25 × 25 × 2.5 mm	683 mm	1	
43	Hollow square	25 × 25 × 2.5 mm	246 mm	1	
44	Hollow square	25 × 25 × 2.5 mm	150 mm	1	
45	Bush	14 × 2 mm	25 mm	2	
	Driving shaft elements:				
46	Tube	35 × 5 mm	1132 mm	3	
47	Tube	41 × 3	58 mm	4	
48	Shaft	25 mm	106 mm	2	
49	Pin	8 mm	50 mm	3	
	Bolts:				
50	for bearings	M 10 × 1.5	40 mm	14	
51	for Motor, pulley, sheaves	M 8 × 1.25	70 mm	14	
52	for Springs suspension	M 8 × 1.25	50 mm	4	
53	for Winch, arm, arm pointer	M 8 × 1.25	40 mm	7	
	Wheels:				
54	Axle A, B and C	d = 300 mm		6	Bowley & Colman Ltd.
55	Stabiliser wheels	d = 250 mm		2	Key
	Bearings:				
56	Y - bearings for shafts	P35K		6	SKF
	Power system:				
57	Motor	PM63		2	EMD
58	Worm gear box	GB2		2	EMD
59	Electric Break	with hand release		2	EMD
60	Controller (PLC)	For PM 63 motor		2	EMD
61	battery	40 Ahr		2	RS
	Chain drive:				
62	sprocket - pilot bore	pitch = 9.5 mm teeth = 17		2	Fenner
63	sprocket - pilot bore	pitch = 9.5 mm teeth = 57		2	Fenner

Table E -1 List of materials used to make the experimental machine (continued)

No.	Part name	Specification	Required unit	No. off	Supplier
	Chain drive:				
62	sprocket - taper lock	pitch = 12.7 mm teeth = 17		2	Fenner
63	sprocket - taper lock	pitch = 12.7 mm teeth = 57		2	Fenner
64	Chain	pitch = 9.5 mm	L = 900 mm	2	Fenner
65	Chain	Pitch = 12.7 mm	L = 1000 mm	2	Fenner
66	Connecting link	Pitch = 9.5 mm		2	Fenner
67	connecting link	Pitch = 12.5 mm		2	Fenner
	Traction mechanism:				
68	Hand winch	T max. = 2500 N		2	Key
69	Spring	rate = 0.84 N		2	SPEC
70	Winch wire rope	d = 3 mm	L = 1600 mm	2	Key
71	Grips	for 3 mm wire		2	Key
72	Cast iron sheaves, plain bore	for 3 mm wire		2	Key

All the steels were supplied from AHA. Full detail of suppliers are presented in references.

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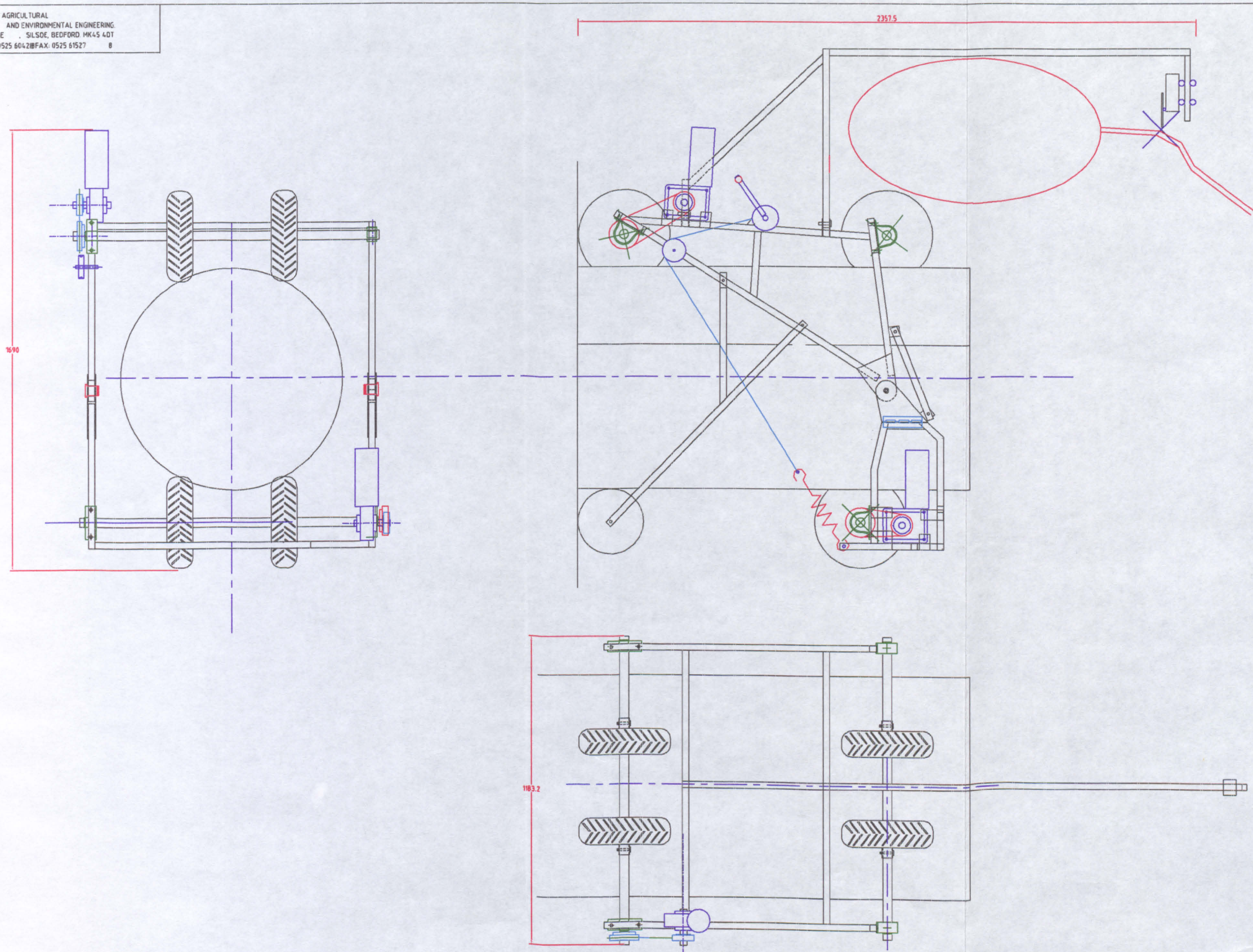

DRAWN	APPROVED	CHECKED	MATERIAL		PROJECT NAME Date Harvesting Machine	PROJECT No. MS 625
M. Shamsi	J. Kilgour	A. Parish	TOLERANCE	No.OFF	PART NAME	
DATE	DATE	DATE	FIRST THIRD ANGLE PROJECTION	SCALE	PART NUMBER	SHEET..1.OF...12