CRANFIELD UNIVERSITY

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DIGITAL SIMULATION OF GAS TURBINE STEADY-STATE AND TRANSIENT PERFORMANCE FOR CURRENT AND ADVANCED MARINE PROPULSION SYSTEMS

PhD THESIS

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Supervisor : Prof. Pericles Pilidis

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SUMMARY

The research study focuses on the idea of simulation of an Integrated Full Electric Propulsion System. The simulation required the development of a gas turbine performance model that could predict the dynamic behaviour of the engine in response to a fluctuation of electrical load. For this purposes it was necessary to evaluate the thermodynamic working process of the gas turbine and a computer code was created. A design point model written in FORTRAN 77 had been transformed to predict the steady state and transient performance of a two-shaft gas turbine and single shaft gas turbine. The models were based on the thermodynamic law of conservation of mass. For the model of the two-shaft gas turbine controls system equations had been derived from off-design analysis and implemented as handles for operation. Both the models were then transformed to a direct link library for the SIMULINK® package. They were further implemented with an electrical network model to form a high-fidelity prime mover-electrical network-propulsion drive interface with which a complete systems analysis was done to understand the response of the three systems in parallel.

In a second part heat exchanger modelling had to be performed so as to create a gas turbine model of an intercooled-recuperated engine. This was done for the steady state behaviour and sizing problem of heat exchangers. The models were run parallel to the steady state code as a validation exercise. Due to time and project restraints the complete incorporation of the models with the gas turbine code was not performed and only a uni-directional system of heat exchanger was created. Over all the period of research parametric studies had been done for comparison of various aspects of performance.

The high fidelity model of the prime mover-electrical network highlighted the reasons for studying the impact of the propulsion drive and electrical network load dynamics on the operation of the prime movers and vice versa. The loss-

of-propulsion-load scenario case study has demonstrated the capabilities of the integrated model, showing clear interactions between the individual subsystems. The interface can now be used to analyse novel types of gas turbine engines in the future. The method adopted to simulate transient performance of gas turbines was useful in understanding the impact of bleed air on current and novel cycles. Finally the task of heat exchanger simulation emphasized the need to create better and accurate models to understand the impact of its behaviour on the gas turbine.

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LIST OF SYMBOLS

A	Area
C	Control variable
C_P	Specific heat at constant pressure
$C*$	Ratio of heat capacity
dt	Time step
D	Hydraulic diameter
e	Heat exchanger wall thickness
E	Error
F	Correction factor for heat exchanger type
F	Force
f	Friction factor
h	Specific enthalpy
H	Total specific enthalpy
h	Convective heat transfer coefficient
I	Inertia
k	Coefficient of conductivity
L	Length
$\dot{m}$	Mass flow
$\overline{m}$	Non-dimensional mass flow
M	Mach number
N	Shaft rotational speed
$\overline{N}$	Non-dimensional speed
NTU	Number of transfer units
Nu	Nusselt number
P	Power
p	Static pressure
P	Total or stagnation pressure
Pr	Prandtl number
Q	Rate of heat transfer
Re	Reynolds number

s	Entropy
S	Specific entropy
SF	Scaling factor
t	Static temperature
T	Total or stagnation temperature
t	Time
U	Overall heat transfer coefficient
v	Velocity
V	Volume
W	Work
x	Any variable
Y	Heat exchanger channel width
Z	Heat exchanger channel height

Greek symbols

β	Beta line
γ	Ratio of specific heats C_P/C_V
δ	Corrected pressure
Δ	Small change or difference
ε	Heat exchanger effectiveness
η	Efficiency
θ	Corrected temperature
μ	Dynamic viscosity
ν	Kinematic viscosity
π	Pressure ratio
ρ	Density
τ	Torque
ω	Angular velocity

Subscripts

AMB	Ambient
ALT	Altitude
AUX	Auxiliary
BL	Bleed
C	Cold fluid
CC	Combustion chamber
CMP	Compressor
FL	Fuel
GG	Gas generator
GS	Guess
H	Hot fluid
HX	Heat exchanger
IN	Inlet to component
ISE	Isentropic
MAX	Maximum
MECH	Mechanical
MIN	Minimum
OUT	Outlet of component
PS	Power shaft
PT	Power turbine
REF	Referred
TB	Compressor turbine
TH	Thermal
TOT	Total
TR	Turbine component

Acronyms

ABC	Air bottoming cycle
ARP	Aerospace recommendation practice
BLDRET	Subroutine to return bleed extractions
CBNLW	Subroutine to perform bleed extractions

CC	Combustion chamber
CF	Counterflow
COMP	Compressor
COT	Combustor outlet temperature
CT	Compressor turbine
CU	Cranfield University
DP	Design point
EGT	Exhaust gas temperature
EN	Electrical network
FAR	Fuel-to-air ratio
FFF	No control schedule on engine
FNDTTSTC	Subroutine to calculate working fluid properties
FINDSTC	Subroutine to find static fluid properties
FINDTOT	Subroutine to find total or absolute fluid properties
GASPROP	Subroutine to calculate physical properties of gas
GBXTR	Ratio of shaft rotational speed
GT	Gas turbine
GTPS	Gas turbine performance simulation
GUI	Graphic user interface
HP	High pressure
HRHE	Heat recovery heat exchanger
HRSG	Heat recovery steam generator
IC	Intercooler
ICR	Intercooled-recuperated
ICV	Intercomponent volume
IFEP	Integrated Full Electric Propulsion
I/O	Input output
IP	Intermediate pressure
ISA	International standard atmosphere
LFFHND	Logical variable to select off-design handle
LHV	Lower heating value
LMTD	Log mean temperature difference

LP	Low pressure
LSTEADY	Logical variable for control of combustor outlet temperature
LVSVSCH	Logical variable for control of variable stator vane angle
MDNR	Multidimensional Newton-Raphson
NASA	National aeronautics and space administration
OD	Off-design
OPR	Overall pressure ratio
NDMF	Non-dimensional mass flow
PF	Parallelflow
PI	Proportional integral
PID	Proportional integral derivative
PT	Power turbine
RC	Recuperator
SFC	Specific fuel consumption
SLS	Sea level static
SN	Station number
SPS	SimPower systems
TET	Turbine entry temperature
TTT	Total control of engine, all schedules operating
UoM	University of Manchester
UoS	University of Strathclyde
VSV	Variable stator vane

1. INTRODUCTION

The history of the principle of working of a gas turbine can be traced back to around the beginning of first century of the current era. Nevertheless, the credit for the idea that leads to the modern gas turbine can clearly be given to John Barber when he patented a gas turbine in 1791. Since that time the gas turbine engine has undergone enormous transformation till today when it is one of the safest machines used to produce useful work. The gas turbines that are in use now-a-days are much more complex machine with some of the best endurance levels, highest efficiencies and safety factors. The technological heights achieved to create a better turbine have produced some ground breaking research techniques and its applications are far spread beyond just a machine to produce power. The last few decades has transformed the gas turbine engine from a concept to a creation the benefits of which are multi-purpose. Gas turbine engines are used as the major propulsion systems in the aviation industry. They power most of the aircraft and some other military machines like tanks and missiles. Some engines are installed on spaceship propulsion systems as auxiliary units. The power generation industry did not fail to see the practical gains in utilising gas turbine engines to produce electricity. The first gas turbine generator set was installed in Neuchatel in 1939 which operated for 63 years. Marine vessel operators also realised the benefits of the gas turbine engine when in 1943 a warship was installed with a gas turbine based system, from a jet engine, as an auxiliary power plant. The gas turbines on board today are slowly replacing the traditional diesel engines for fast speed vessels and can also be used in fast ferries to power all the system utilities.

1.1. Aims and objectives

The gas turbine is used today by operators over a wide scope of applications. The economical cost to fit one technology for multiple areas sometimes makes the original concept non-feasible for its application. Therefore, if background research on gas turbine cycles is carried out to develop new ideas and areas of use, and if a demand arises then by reducing the initial cost of assessment the original technology can be quickly applied. The idea of this project comes from that point of view, to use a gas turbine engine in marine vessels as a prime mover, controlled by fully integrated electric propulsion system. The main objectives of this research are

- To study the emerging technologies of gas turbine engines so as to make it applicable on marine vessels.
- To create a gas turbine engine model to digitally simulate the steady state and transient performance of selected engine configuration.
- To integrate the simulated model into a system tool and produce a high fidelity prime mover-electrical network interface.
- To create a simplified model of heat exchangers so as to simulate an advance cycle as turbine power plant.
- To perform parametric studies on transient operation of the selected power plants in a full electrical propulsion system.

1.2. Thesis structure

To assist the reader in understanding the tasks performed by which some significant results are provided at the end, the thesis is separated into different chapters. Any literature review which has been performed is either included at the beginning of each chapter or discussed with the results.

Chapter 1 provides some background research on the different types of gas turbine engines. The first part introduces the current technologies while the second part provides ideas of emerging technologies of gas turbine engines with potential for use in marine applications.

Chapter 2 describes the concept of an Integrated Full Electric Propulsion; the concept and the challenges of developing a system tool for a novel application. It then shows a technique of solving the issue of integrating two accurate models to create a single high fidelity model for a gas turbine and electric network. Some case studies and results of simulation are presented.

Chapter 3 is the core program design methodology and its adaptation as a different engine cycle. After a literature review on the subject of simulation, two engines; a two-shaft gas turbine and a single shaft gas turbine are simulated. Their design point performance is predicted and different cycle operating points along with secondary effects of components are simulated for final consideration.

Chapter 4 presents the off-design methodology to predict the two-shaft and single shaft engine performance. Parametric studies are carried out on the two engine cycles and part load operation of each is studied.

Chapter 5 is the transient performance simulation of gas turbine engine. The chapter reviews state of the art techniques in digital simulation that have been applied in the last few decades. It describes the simulation methodology in the current project and provides the control schemes that have been applied.

Chapter 6 is a literature review of heat exchanger modelling with description of a few methods that were used in parallel with the gas turbine performance simulation codes.

Chapter 7 presents the results of the work conducted in this project and discussed from the author's point of view. It is a graphical representation of the research output.

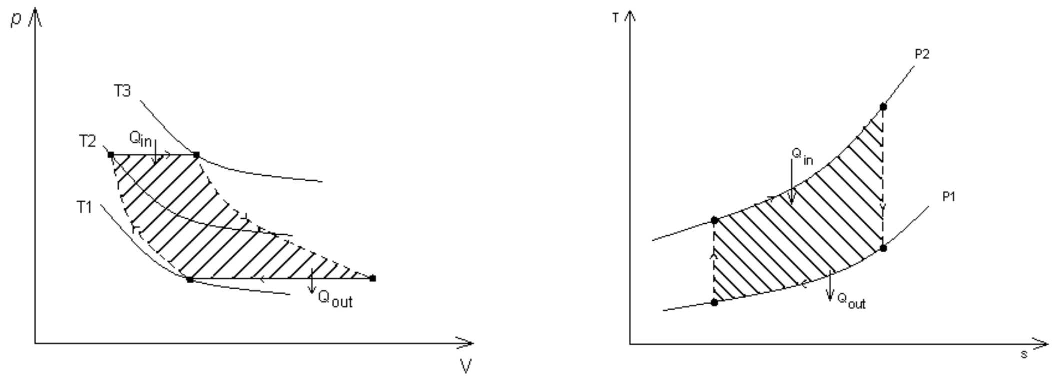
Chapter 8 has the final conclusions to the aims and objectives of this study and the goals that have been achieved. It provides future recommendations based upon the author's logic and reasoning.

1.3. Overview of Gas Turbine Technologies

The gas turbine engines used in marine vessels are similar to those used in power generation and shall be described and analysed in terms of power output. From the extensive research carried out in the development of gas turbine engines it is well known that such engines are efficient at its full power settings i.e. when the engine is running at 90-100% speed. The power system steady state output dictates the level of performance that can be attained by it, while the specific fuel consumption (or thermal efficiency) is a major factor for defining the range and endurance of the gas turbine. As a flavour of the working process in the following few sections an overview of current gas turbine engines is laid out.

To assess the working of gas turbine based engines a few basic fundamentals of thermodynamics have to be mentioned. As the name literally indicates the working medium of such propulsion systems is a gas. The gas properties like temperature, pressure, mass and volume that are contained within the gas are analyzed to understand its working behaviour. The state of the gas is determined by these thermodynamic properties and the co-relation that exits between them. These relations are derived from either theoretical background or from scientific observations. The first and second laws of thermodynamics define two additional variables, enthalpy and entropy, which can also be used to describe the state of a gas. Thermodynamic processes, such as heating, cooling or compressing the gas, change the values of the state variables in a prescribed manner. The total work and heat transferred to a gas depend on the beginning and ending states of the gas and on the process used to change the state[1]. In an engine several changes in the state of the gas occur. When the processes happen such that the final state of the gas is similar to its initial stage, it is said to have completed a *cycle*.

During a thermodynamic process a plot which shows the changes in the state of a gas is beneficial and it is generally illustrated by two types of plots, which describe this phenomenon. In the Figure 1.1(a) is plotted the pressure versus the volume, and is known as a p-V diagram. The solid lines represent constant temperature and the process occurring along it is called as an *isothermal process*. The dotted lines are processes happening at where no heat is transferred to the gas, but the temperature, pressure, and volume of the gas change. This process is called as an *adiabatic process*. As seen in the same picture, the shaded area on the p-V diagram is equal to the work performed by a gas during the process.



(a) (b)

Figure 1.1 Thermodynamic cycle of closed system illustrated by *p-V* and *T-s* diagrams

On the Figure 1.1(b) s a graphical representation of the temperature versus the entropy of the gas. This graph is referred to as a T-s diagram. Lines of constant pressure indicating an *isobaric process* are drawn on this plot. In a gas turbine, an isobaric process occurs in the combustion chamber. For an ideal cycle gas turbine engine, the process of compression and expansion occurs with no change in the entropy of the system and they are reversible and adiabatic. The process with a change in the gas state at constant entropy is known as an *isentropic* process. The shaded region outlined by the cycle under the process curve on the T-s diagram is related to the amount of heat transferred to the gas, which also indicates the total work done for compression and expansion of the gas. Engine cycles are illustrated by means of these diagrams.

Thermodynamic cycles are of two forms; closed-cycles and open-cycles. In the closed-cycle types of engines the working fluid is continuously propagated in a circle through its various components. The working medium is compressed in a compressor like any ordinary gas turbine. The medium is then raised to a higher temperature in a separate auxiliary unit instead of the conventional combustion chamber and heat is inputted to the turbine. The turbine then performs the expansion of the medium and contrary to most gas turbine engines the exhaust gas is cooled before it is re-introduced in the compressor. As the working medium in such a type of gas turbine repeats the process, these types of cycles are referred to as closed-cycles. The open cycle type of gas turbine is the common engine used in most of the current applications as a propulsion unit for aircrafts and many industrial applications.

1.3.1. Two-Shaft GT Description

The basic configuration of a two-shaft GT (a single spool gas turbine with a free power turbine) is an intake followed by a compressor, combustor, turbine, a power turbine and an exhaust. This configuration is also popularly known as a Turboshaft engine. The compressor is driven by the high pressure turbine and forms the gas generator unit for the two shaft arrangement. The compressor with its driving turbine sits on one shaft without the need of a gearbox as the two components have the same rotational speed. The power turbine may be connected to an external electrical generator or a marine propeller through a reduction gearbox as the speed of the shaft is significantly lower than the turbine shaft speed. Figure 1.2 represents the schematic layout of the two shaft gas turbine engine.

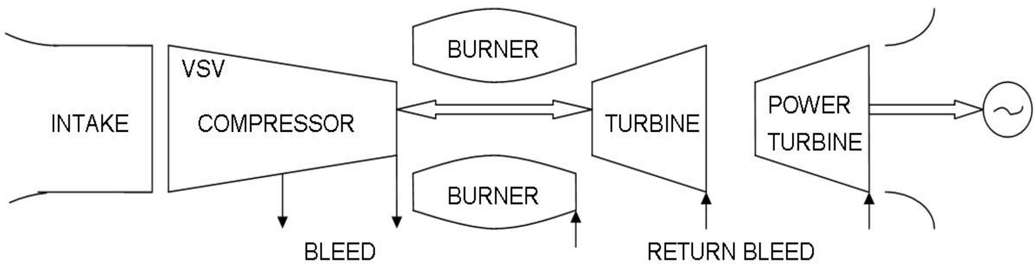


Figure 1.2 Schematic layout of two-shaft gas turbine engine

The T-s diagram of an ideal two-shaft is illustrated in Figure 1.3

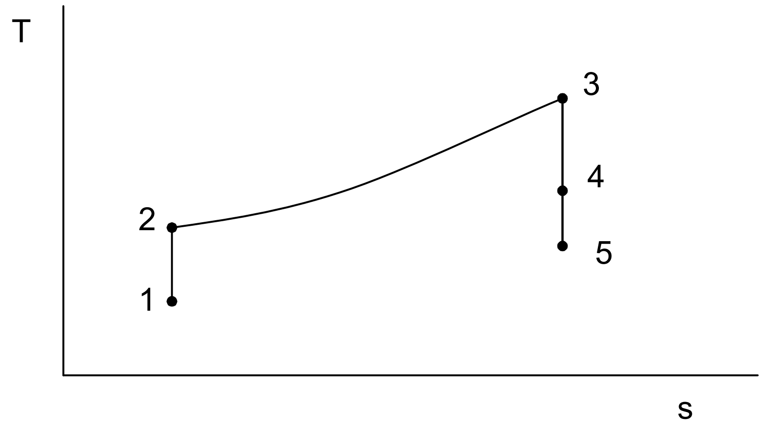


Figure 1.3 Temperature (T) vs. Entropy (S) of an idealistic two-shaft GT
1-2: compression; 2-3: combustion; 3-4: expansion (compressor turbine); 4-5: expansion (output power)

The advantage of having a two shaft gas turbine lies in its flexibility of operation. This means that for power requirements which fluctuate according to the external load, the turbine can then provide power autonomously. To start-up the engine is also easier in this case where only the gas generator needs to be brought up to design speeds. However there is the disadvantage of added weight and increased costs of manufacturing. The performance of the engine needs to be carefully monitored for rapid load shedding, which can lead to over-speeding of the turbine and even mechanical breakdown. Today many jet engines are modified into two-shaft gas turbines that are utilized as a source of power. This is achieved by replacing the exhaust nozzle of a jet engine with an expansion unit such as the turbine as described by the two shaft arrangement or by a simple turbine connected to the gas generator through a gearbox.

1.3.2. Single Shaft GT Description

When the load required, driving a propeller or electrical generator is at a fixed speed, the single shaft GT is suited the most. A single shaft gas turbine engine is the jet engine in which the nozzle is replaced by a turbine which expands the gas to obtain power necessary by the compressor to compress air and extra power that is utilized as useful work. Component-wise the single shaft engine is also like the simple cycle gas turbine engine with the difference that the power turbine is either the turbine of the gas generator with a higher number of stages or a separate turbine component connected to the gas generator with a reduction gearbox to maintain the necessary constant speed. Figure 1.4 represents a single shaft gas turbine engine.

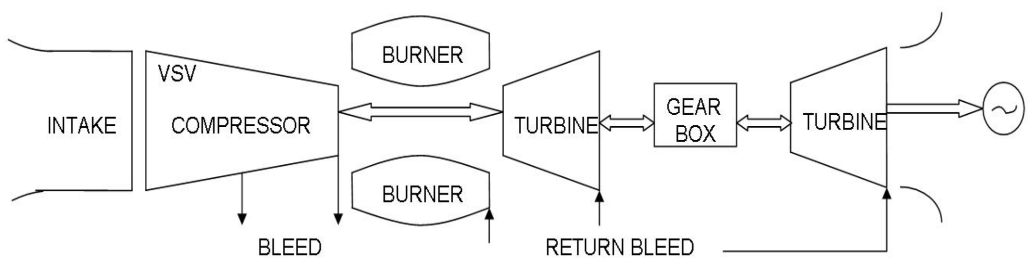


Figure 1.4 Schematic layout of a single shaft GT

The single shaft engine has the compressor, the turbine and the power turbine on the same shaft as visible in the figure. The engine is an open cycle thermodynamic process and the T-S diagram is similar to the two-shaft. In certain cases there is a minimal difference graphically where the line 3-4-5 is replaced by a line 3-4, which simply indicates that the power turbine is like aforementioned, an extended turbine.

The time required for achieving the change of load and rotational speed is not a parameter of paramount importance for such GTs. There is no danger of over-speeding for these engines because of the high inertia produced as a result of the drag of the compressor. These engines have a better performance in terms of thermal efficiency at maximum output. The single shaft gas turbine is the preferred engine for many industrial applications as it is compact and efficient.

1.4. Novel Type of GT Engines

Although the simple cycle type of gas turbines are highly efficient when working at full power making them ideally suited for low cost installations, these types of engines have low efficiencies at part-load operation making them uneconomical in such cases. Thus, novel thermodynamic cycles with a heat recovery have been designed to have acceptable part-load efficiency. The heat recovery system can be used in two different arrangements to increase cycle efficiency[2]:

1. Bottoming cycles, in which the exhaust from the simple cycle is used as an individual source of heat in an independent power cycle. These engines are also known as Air Bottoming engines.
2. Heat recovery cycles; in which the exhaust heat from the turbine outlet is used in the same gas cycle to heat up the gas prior to combustion. Such engines are called as Recuperated/Regenerative engines.

1.4.1. Air Bottoming Cycle

The Figure 1.5 shows a principle representation of an air bottoming cycle. The first/top cycle is a simple gas turbine with a free power turbine. The bottom cycle uses the exhaust gas of the top cycle, which is nearly at atmospheric pressure in a heat recovery heat exchanger similar to that of a steam turbine to heat up the gas of the bottom cycle and provide the necessary heat input for the same cycle. There is a second turbine to extract power from the bottom cycle thus increasing the overall power output by around 30% at a thermal efficiency in the range of 49%[3]. However in order to obtain the same heat capacity between the exhaust gases and the compressed air flow in the ABC, (i.e. flow rate times specific heat) and maximise the heat transfer efficiency, the compressed ABC air has a higher flow rate of approximately 3-5% in order to offset the higher specific heat of the exhaust gases[4]. The top/bottom cycle can be designed either as a single-shaft or two-shaft gas generator set with either a single or two intercoolers. Previous research indicates that the most efficient ABC would be a two-shaft engine with two intercoolers, optimum pressure ratio between 8 and 10 and a recuperator[4 5].

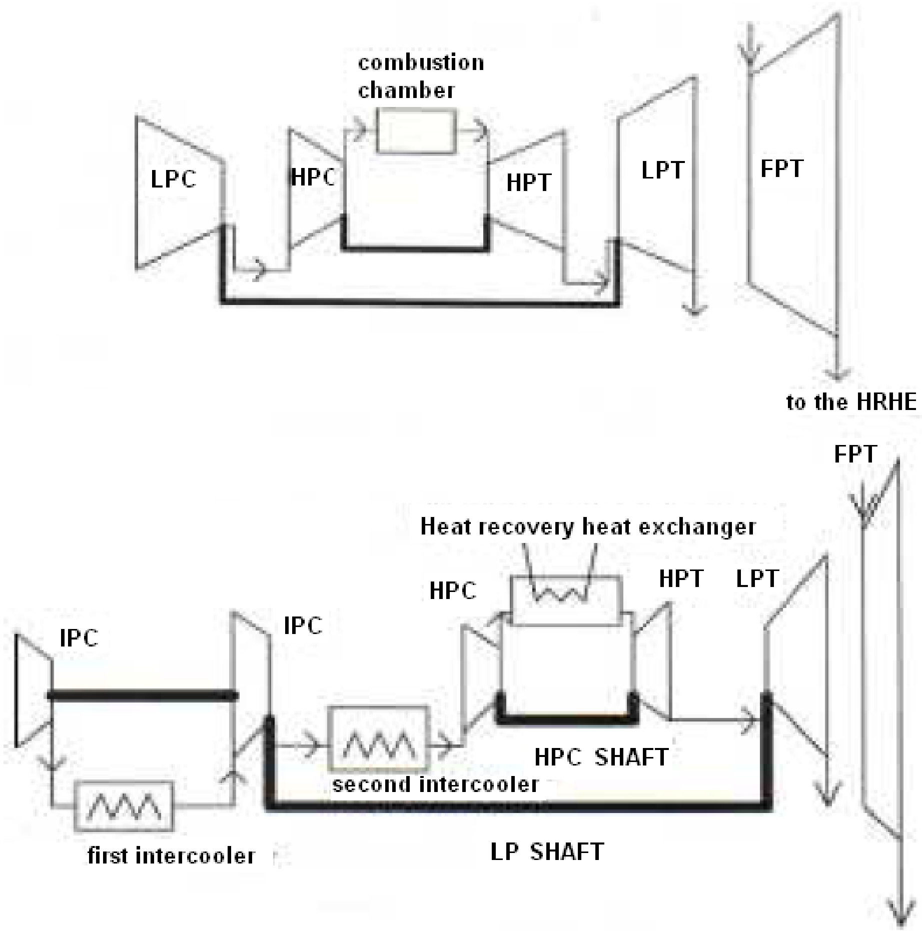


Figure 1.5 Air bottoming cycle

The ABC provides a much more feasible option for marine vessels at normal part load efficiency compared to a steam turbine as a bottom cycle (55-60% thermal efficiency[4]) with reduced hardware as components such as the boiler, steam turbine, condenser, pumps are removed and also with reduced initial capital cost, safer in operation and needs less human resources and time intervals to be maintained.

1.4.2. Steam Injected Gas Turbine

With the availability of large quantity of disposable water, the steam injected gas turbine can add to the benefit of increased efficiency cycle in a marine powerplant system. Figure 1.6 shows the working process of a Steam Injected Gas Turbine. Its basic working principle is that the exhaust gas from the turbine outlet is used to generate the steam, which is injected back into the gas path at the combustion chamber or turbine inlet. The efficiency of such a cycle is therefore higher than a simple cycle due to the added heat from the steam. To heat up the fresh water a heat recovery steam generator is required, which has to be supplied with fresh water. This arrangement provides sufficient heat to not only benefit the gas turbine but also additional heat is available form the HRSG for any other heat process. Performance calculations carried out with some hypothetical but valid numbers show that steam injected in a gas turbine from a heat recovery system generator increases the power output by more than 20% and cycle efficiency more than 5%. In terms of cost reduction this can be significant, however there are technological and weight limits to be considered. The steam injected has to be of high quality coming from treated, pure water evaporated in boilers. Though availability of it is not an issue, the cost of system equipment necessary to generate the steam coupled with large space requirements offers a substantial challenge[6].

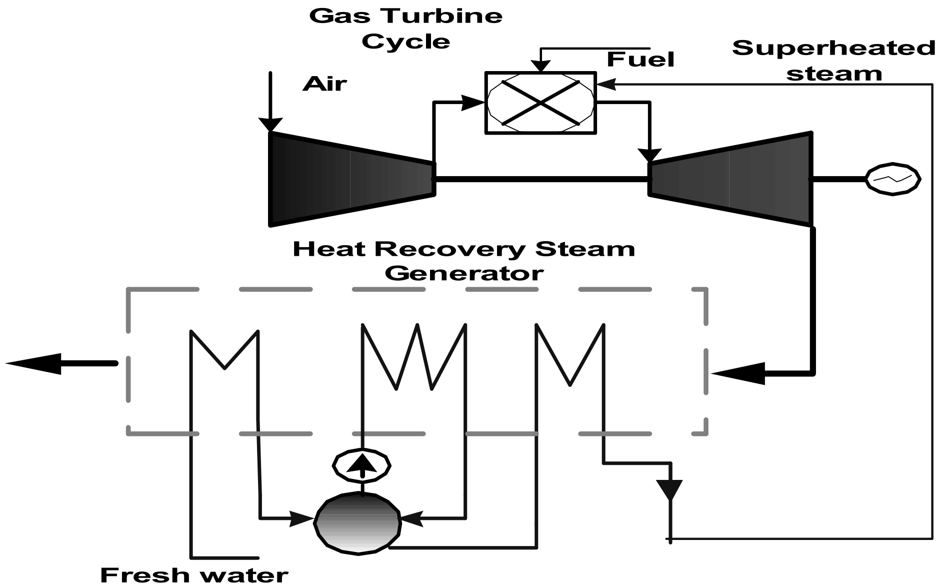


Figure 1.6 Steam Injected Gas Turbine[7]

1.4.3. Heat Recovery Cycles

To date, the use of gas turbines for marine propulsion embodying internal heat recovery within the thermodynamic cycle has been limited[8]. However, simulation techniques have shown that the part load efficiency of such novel thermodynamic cycles is better than the simple cycle gas turbine. This is due to the fact that the turbine entry temperature is high as well at part load settings, which is increased by the heat from the exhaust gas of the cycle. The heat recovery cycles include two kinds of components viz. recuperator and regenerator. These forms of two-fluid heat exchanger can be used in various combinations.

1.4.3.1. Heat Exchange cycle

In the heat exchange cycle, the compressor discharged airflow is ducted to a heat exchanger which is heated by the energy transferred from the gas turbine exhaust gases. As a result the increased temperature of the compressed flow that passes into the combustor needs less energy (fuel) to produce the same output power. For a heat exchanger assumed to have an effectiveness of 80%, the efficiency of the regenerative cycle is about 40%

higher than its counterpart in the simple cycle in part load conditions[9]. The Figure 1.7 shows a schematic representation of a heat exchange cycle. The heat exchange cycle (recuperative or regenerative according to the type of heat exchanger) provides significant improvements in the specific fuel consumption (SFC) at part-load operations[10].

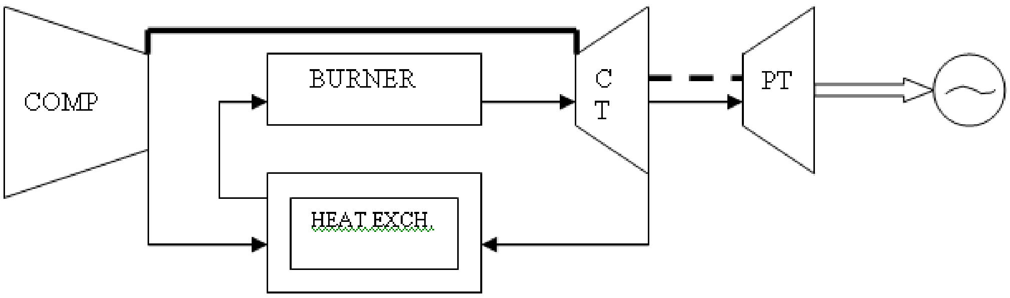


Figure 1.7 Heat exchange cycle

The specific power is about the same or slightly less comparing with the simple cycle, but the optimum pressure ratio for maximum work is the same in both cycles (simple and heat exchanged). Maximum efficiency, which is a function of the cycle temperature, is obtained at lower pressure ratios[10]. At approximately above a pressure ratio of 10, the increasing compressor-discharge temperature and decreasing turbine exit temperature, reduce the effects of a heat exchanger[8]. The engine can be designed in different spool combinations (single or multiple shafts). Gas turbines designed with power turbine operate with varying compressor speeds, and this gives them the advantage of high heat recovery capacity since the turbine exhaust temperature is higher due to the reduced mass flow at part load conditions[11]. Though the advantages of the heat exchange cycle are significant, their marine application is withheld due to the limited knowledge of design of heat exchangers suitable for marine propulsion.

1.4.3.2. Intercooled-Recuperated Cycle

The intercooled-recuperated (ICR) engine offers a 30-40% reduction in specific fuel consumption compared with contemporary gas turbines[8]. It is schematically shown in Figure 1.8; where the compressor is split into a low and high- pressure compressor with an intercooler placed between the two.

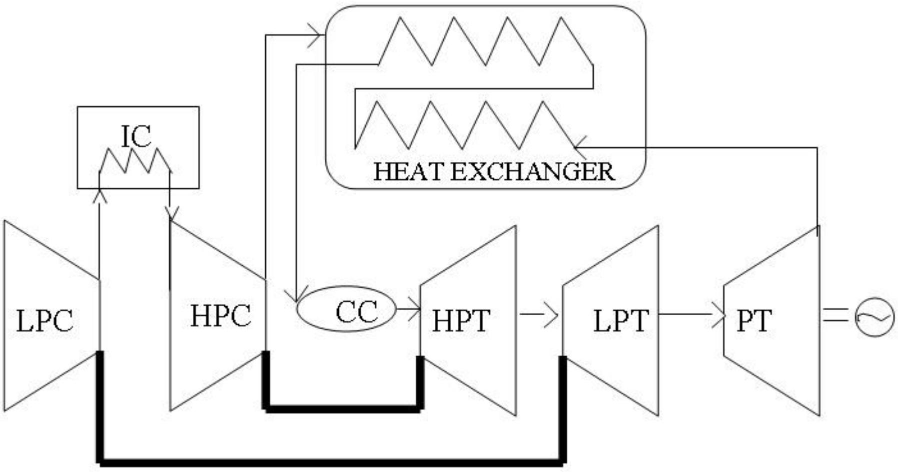


Figure 1.8 Schematic layout of an ICR engine

This provides a higher cycle pressure ratio as the two components can be well within manufacturability limits. In a more advanced configuration, two intercoolers, one between the LP compressor and the intermediate pressure compressor (IP) and one between the IP and the HP compressors, reduces the temperature and the specific volume (and also the blade height at the last compressor stages resulting lower compressor efficiency, which makes the cycle more suitable for power over 2 MW)[12]. Further there is present a heat exchanger similar to that of a heat exchange cycle thus providing a higher combustion temperature whilst reducing the fuel input. The compressed flow after it is ducted to the heat exchanger reduces the amount of energy (increasing the specific power of the cycle and the thermal efficiency[10]) provided by the turbine(s) to rotate the HPC or the IPC and HPC according to the configuration. Since the engine can be designed as a multi spool combination wherein the compressors sit on different spools to that of the power turbine, provides optimal values of flow rate, pressure ratios and rotational speeds. The ICR engine WR21 (Figure 1.9) has been developed by The Northrop Grumman Marine Systems / Rolls-Royce which boasts of a 27% reduction in fuel consumption with a power output of 25 MW[13].

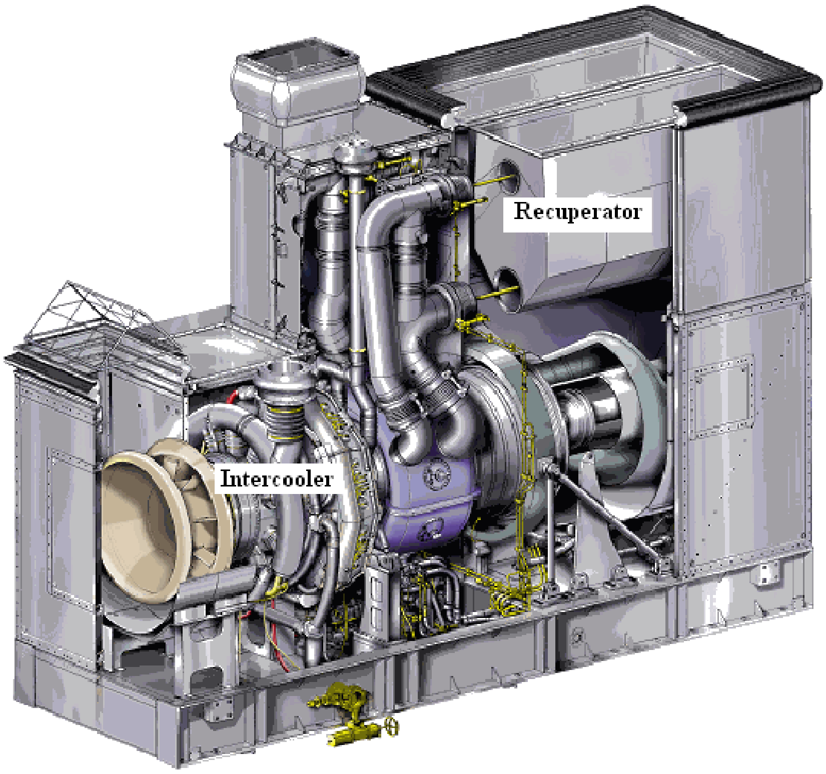


Figure 1.9 The WR-21 Marine propulsion ICR gas turbine[14]

A complete description of the WR21 is provided further in the chapter on heat exchanger modelling. The ICR is well suited for marine propulsion with electrical power generation on board due to reasons such as[8]:

- High efficiency
- Excellent part load efficiency
- Acceptable specific power
- Compact power plant compared to diesels
- Reduced thermal, infrared and acoustic signatures
- Potential for high reliability

1.4.4. Combined Cycle Engines

Although the marine vessels market is not known for employing gas turbines on board as prime movers, it is well used in conjunction with different types of engines to form combined cycle engines. Table 1-1 mentions a list of such combined cycles.

Table 1-1 Combined cycle engines

COSAG	Combined steam and gas
CODAG	Combined diesel and gas
COGAG	Combined gas and gas
COSOG	Combined steam or gas
CODOG	Combined diesel or gas
COGOG	Combined gas or gas

This is a popular list for current markets of power generation, however not the complete. The nomenclature is easy to follow where 'combined' is abbreviated as *'CO'* and steam, diesel, gas turbine as *'S'*, *'D'*, *'G'* respectively. Note that in the list the gas turbine is simply mentioned as gas, which is the medium of work in the engine. The configurations possible are in terms of and/or that is the *'A'*, *'O'* in the nomenclature. If either of the different engines, when two or more cycles are combined, can be used for power together or separately then the configuration is *'A'*. When the output for just one of the engine is accessible at one time then the combination is *'O'*. A few advantages of these engines are shortlisted:

- When in cruise, the ship can access power from the higher efficient engine making it more profitable to operate the journey.
- To reach target speed when needed, like for war ships, a power boost can be provided by the second engine.
- The steam and diesel engines provide better efficiency at part load while the gas turbine provides compactness and increased power requirements.

2. ADVANCED MARINE ELECTRICAL PROPULSION SYSTEM

At the current technological level a system utilising different types of prime movers associated to a flexible electrical power distribution network and connected to advanced propulsion drives is a task not unaccomplishable to provide a systematic solution for current and future naval vessels. Such a system commonly known as an Integrated Full Electric Propulsion (IFEP) system add value to the development process of a vessel by reducing the through-life cycle costs, reducing harmful emissions and enhancing operational capabilities and increased availability. The systems engineering approach paved way for an academic/industrial partnership into designing a holistic simulation tool to investigate the behaviour of the IFEP. The end-product was to be a tool for the marine industry in the pre-design stage to allow proof of concept and thus providing technological back-up and offering risk-mitigation. University of Strathclyde, University of Manchester and Cranfield University were the academic institutions involved. Theirs was the main role to develop the systems package. The project received funding from the EPSRC while Lloyd's Register, Carnival Lines and Rolls-Royce provided some useful technical data, comments and recommendations.

The task of creating the systems simulation tool was sub-divided between the universities as follows:

1. University of Strathclyde (UoS): to prepare the electrical simulation models and integration of electrical and mechanical models
2. University of Manchester (UoM): preparation of models for electrical drives, propellers and ship
3. Cranfield University (CU): simulation of prime mover models, techno-economic environmental and risk analysis of marine power plants

The work between the universities was an inter-dependent model as illustrated in the next Figure 2.1

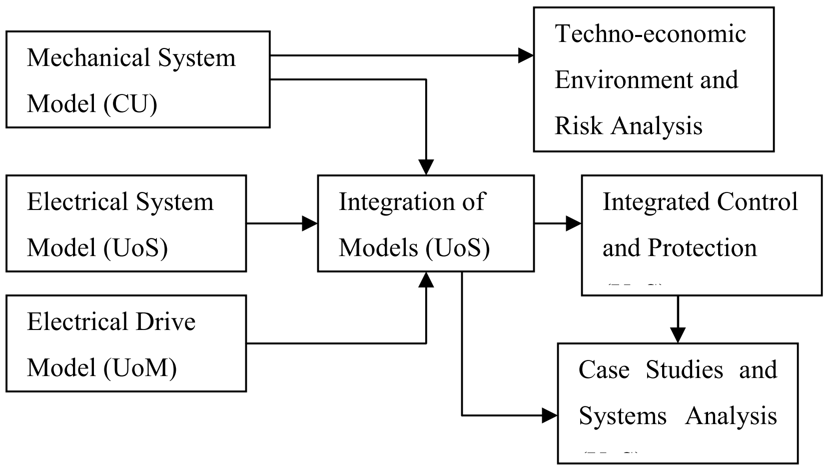


Figure 2.1 Task dependencies

Advanced Marine Electrical Propulsion System (AMEPS) is a product of this consortium. The main result achieved by AMEPS was the development of high fidelity models of each subsystem and integrating them in the systems tool. This research has not been carried out before in the marine vessels design and development industry and AMEPS can be considered as a huge step forward in the direction of high fidelity modelling. There are many models available that represent electrical networks, gas turbine steady state and dynamic performance, mechanical and electrical drives, propeller analysis, ship design etc. Their physics is well defined and they yield rather accurate results as a single entity. AMEPS took the study one step forward by incorporating three major models, which define a full electric propulsion system and focused on the system behaviour with various load disturbances and fluctuations. To perform this task, models founded on validated techniques were created and integrated into one holistic tool.

At the end of the first phase of AMEPS, case studies of various scenarios, which may occur during a journey of the marine vessel, had been run using the tool. The results were then studied and further modification were made of the tool towards a better network distribution and the performance outputs were fine tuned based upon engineering logic. Thus with the AMEPS a framework is set for designing smarter control strategies based on the outputs of highly accurate performance patterns, suggest different standards of networking for full electric propulsion, provide a capability for de-risking a marine system development program, reduce initial costs of development by through a techno economic environment and risk analysis and contribute toward the knowledge of eminent system modelling techniques.

2.1. Simulation Tool Environments

To create a high fidelity system tool it is of utmost important that a right choice of development packages is selected for every prominent subsystem. To this idea the simulation of the subsystems was carried out on different software packages that could handle the level of accuracy demanded by AMEPS. Initially to simplify the issues of various models when integrated into one and determine the input/output criteria a decision was made to use a single overall modelling environment. Once the parameters of exchange were decided the choice of the software was left independent to the modelling expertise of each university to obtain a better simulation tool. By this kind of approach, the project never came to a standstill if any one platform failed as a choice and constant interaction meant that there was always work needed to be carried out Table 2-1 outlines the software platforms used for modelling of the subsystems;

Table 2-1 Modelling software for AMEPS

Task	Software	Implementer
Gas Turbine models	FORTRAN	Cranfield University
Techno-economic environment and risk analysis	TURBOMATCH + MATLAB®	Cranfield University
Electrical network models	MATLAB+SIMULINK® (SIMPOWERSYSTEMS™)	University of Strathclyde
Integration of models	FORTRAN+MATLAB+ SIMULINK	University of Strathclyde
Electrical drive models	MATLAB+SIMULINK	University of Manchester
Propeller and ship models	MATLAB+SIMULINK	University of Manchester

The gas turbine engine models were coded in FORTRAN as Cranfield University has already developed extensive programs to analyse the performance of such engines. These models simulated the design point, off-design and transients, with the main objective of understanding the dynamics of the system. Fortran was developed specifically for scientific and numerical programming. A thermodynamic scheme of GT is a mathematical process and thus the choice of Fortran is obvious. A few advantages of this software are:

- Fortran is much better suited to scientific world than other languages like C, C++, and JAVA.
- There is plenty of open source information available in the form of complete programs, subroutines, examples etc.
- As a language to learn, Fortran is easy to understand for the novice programmer.
- The mathematical implementation is more logical making the language robust and easy to use.

Turbomatch, which is a propriety performance code developed by Cranfield University is also written in Fortran. The combination of Turbomatch with Matlab was used to develop the TERA code. The design point and off-design calculations using Turbomatch, while using Matlab for TERA. Further reading on the methodology and results from TERA can be found in the PhD thesis of Tsoudis (2008)[15].

The electrical network, electric drive, propeller and ship models have all been simulated using Simulink of the Matlab environment. Modelling using this software is best suited as writing a new code to represent the physical model of IFEP using any programming language will run into tens of thousands of lines. Diagnostics and optimization is not very feasible in this case. “SimPowerSystems (SPS) extends Simulink with tools for modelling and simulating the generation, transmission, distribution, and consumption of electrical power”[16]. To understand the concept of electrical network and drives modelling papers presented by Elders[17] et al and Apsley[18] are referenced for the reader.

2.2. Determining I/O Parameters

Simulating the subsystems of an IFEP in different software packages required to determine a list of variable parameters between these systems to be transferred across in a bi-directional manner. A change in any one of the models should arouse a reaction in the whole system, which is truly representative of a complicated network model. To facilitate such action-reaction type of behaviour the whole electro-mechanical network was split into three parts and was interlinked into a Gas Turbine (Propulsion System)-Electrical Network (GT-EN) and an Electrical Network-Propulsion Drives (EN-PD) interface.

2.3. Gas Turbine-Electrical Network interface

The types of propulsion systems modelled were a Two-shaft GT and a Single-shaft GT. The Electrical network (Figure 2.2) model is based upon the Type 45 destroyer of the United Kingdom Royal Navy[19].

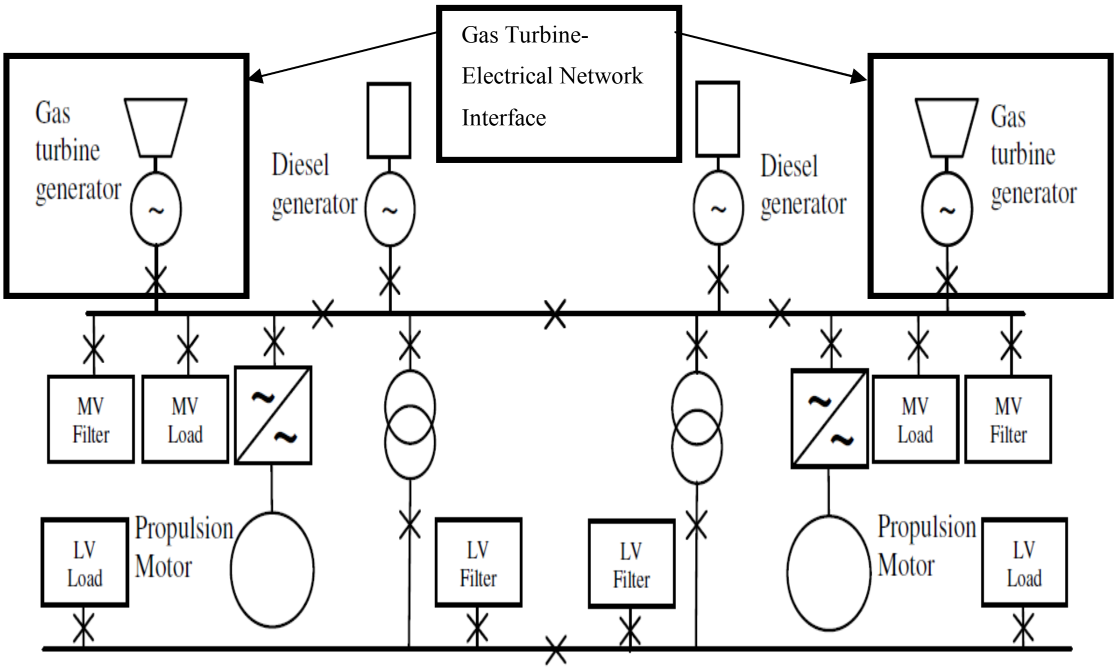


Figure 2.2 Type 45 single line network diagram[19]

In the GT-EN phase the engine models besides it thermodynamic properties, provide an output of power demanded by the network. This power demand is gathered by the sum of the power requirements to maintain a certain vessel speed, driving the various generators and any other loads such as hotel demands, that are all modelled into the electrical network and propulsion drives subsystem.

Traditionally it would be appropriate to solve the differential equation of power turbine and generator spool speeds (eqn. 2.1)[19] within the GT model.

$$\omega = \frac{1}{I}\int \Delta P dt \tag{2.1}$$

This will lead to designing the control system of the engine as an integral part of its model and the control system of the generator, electrical network, propulsion drives as part of their individual models making it too complicated to be handled at the IFEP systems level. Thus the power output from the GT is an input to the generator model to overcome any additional matching blocks to be simulated. The network model result, besides analytical data for network management and inputs to the propulsion drive model, is the generator spool speed and the fuel flow which is an input to the gas turbine. The gas turbine-electrical network interface is shown in Figure 2.3

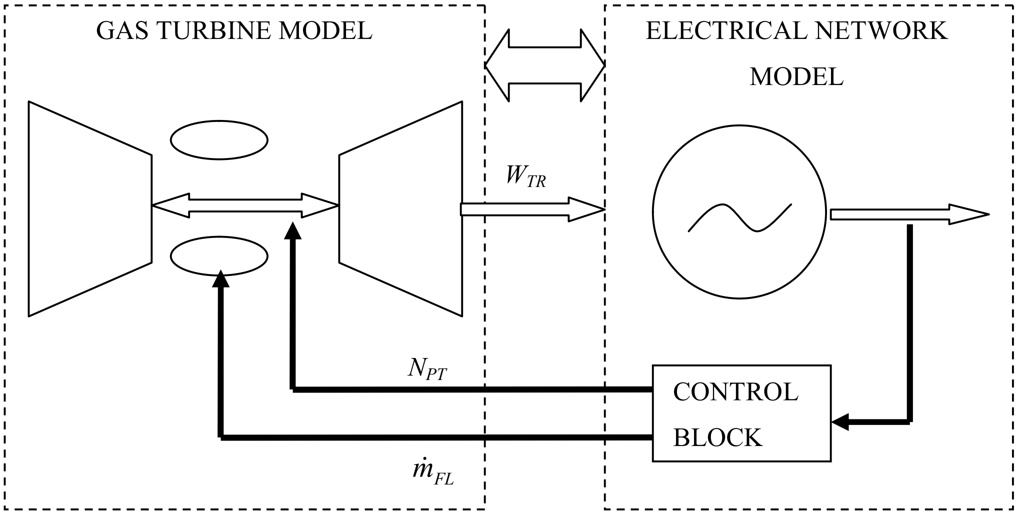


Figure 2.3 Gas turbine-Electrical network Interface linkages

The control block is a Power-Integral (PI) governor toolbox of Simulink. The PI is fed with two speeds, viz. the speed of the generator and the speed desired from the propulsion drive. Based on the control theory it performs a matching and the result is a fuel demand which is fed into the engine. The Fortran GT code with the fuel and the electrical generator inputs calculates the power delivered and feeds it back into the network model to the part of propulsion system. Setting up this interface gave the primary control of power demand to the network making it the central model of the IFEP. Contrary to its name, the control block has no significant impact on the engine operation. The gas turbine model has internal schedules to manage its mechanical limits and also the transient calculation of the gas generator is carried out. Chapter 5 explains much more about the dynamic modelling of the gas turbine, but at this point it needs to be mentioned that the simulation of the GT model can be carried out at a different time step than the simulation of the EN model.

The Electric Network-Propulsion Drives interfacing was carried out jointly by the University of Strathclyde and University of Manchester.

2.3.1. Integration of Models

Previous works[20,21] on modelling a physical system like the IFEP or an electrical or propulsion system typically focused on one important factor to represent the system and assumed a level of performance of the other subsystem(s). A technique of integrating several highly accurate simulation tools into one overall model was until today either not possible because of technological levels of computing power available at cheap costs or required specialist programmers or engineers fluent in programming with a high level of knowledge in various fields of engineering. Splitting the systems tool into smaller subsystems and providing between them a rigid interface allows the creation of a very powerful model while requiring easily available resources and average manpower.

The task of integration requires constant updates in all the subsystem models. When case scenarios were run for the IFEP systems tool a disturbance was initiated at one level. As the noise propagated through the system, a high risk of computing failure was detected. Even though the models were very robust individually, when integrated together, a certain command not supported at any one level resulted in a complete system failure. Thus, some techniques were adopted in safeguarding the system from crashes. For the gas turbine code much effort was focused on creating a code whereby it would consume system memory as low as possible and at the initial stages avoid the use of complex mathematical solutions in the code so as to safeguard the computing system.

As integration of the GT-EN interface to create a high fidelity system required constant data swap between the two universities it formed crucial part of the authors work in development of the gas turbine model. The unavailability of SPS at Cranfield and the Fortran model in Glasgow and to maintain the proprietary rights of the individual universities, accentuated the demand for an accurate and highly robust interface of data transfer. The modelling platforms being different added to the complexity of the integration resources available.

The solution was found in a way of setup explained further. At this point it can be stressed that the interface achieved met the requirements set above and the result is a high fidelity model of the prime mover and the electrical network and the interfacing can be considered a success at the very first time.

To setup the GT-EN interface it was necessary to integrate the Fortran models into the Simulink environment. Simulink 6.3 supported Digital Visual Fortran v5.0 and 6.0 and Compaq Digital Fortran version 6.5 and 6.6. In spite of the numbering system, these are consecutive versions from the same product family. The supported compilers use pointers to pass variables between the Simulink model and the Fortran module. Although pointers exist in Fortran, the form used in the interface is non-standard. Matlab 7.1/Simulink 6.3 supports using the type 1 s-function, which only requires the user to supply Fortran source code. The type 2 or second generation s-function uses a C wrapper to call Fortran subroutines. This is simple if the C and Fortran have the same version and supplier and the C complier is explicitly supported by Simulink. In the type 2 s-function, Simulink supports the C wrapper, but it is up to the user to implement the C calls to Fortran.

2.3.2. Calling Fortran from within Simulink

Three elements are needed to call Fortran code from within Simulink.

1. The Type-1 s-function block. This can be found in the standard Simulink library block under user-defined functions and is placed in the Simulink model.
2. The Fortran file *Simulink.f.* This file acts as the interface between Simulink and the custom Fortran code. It is supplied with Simulink and should not need editing. It needs to be located in a directory which is included in the Matlab path. It is compiled with the user code using the Matlab *mex* command, which is fully documented in Matlab.
3. The desired Fortran code. The code needs to comply with the template given in *sfuntmpl_fortran.f*. This template provides a set of subroutine calls that Simulink can access, as listed in Table 2-2. The template is

fully documented in the Simulink help files. The code may include calls to external modules or libraries.

Table 2-2 Subroutine calls from Simulink to Fortran

Subroutines	Scope
SIZES	Defines the model input/output, states, whether it is continuous or discrete, and whether the output depends directly on the input.
INITCOND	This is called once at the start of the simulation and can be used to set initial conditions.
DSTATES, DOUTPUT	These define the values of discrete states and output at each update interval.
DERIVS, OUTPUT	These define the values of continuous states and output at each update interval.
TSAMPL	This passes timing information back to Simulink
SINGUL	This can be used to set default values if a singularity is detected.

2.3.3. Operation of the Fortran Code within Simulink

All data flow between the Fortran code and the Simulink model is initiated by Simulink, which calls the subroutines as listed in table 2.2 above. The Fortran code must be fitted to these subroutine calls. However it is also possible to link to external library or source files and to read and write data files within the Fortran code.

Integration of continuous states is performed by Simulink, according to the algorithm specified in the runtime parameters. Either a variable or fixed time-step algorithm may be selected.

For discrete variables, the user needs to define the new discrete state as a function of the previous states and current inputs. Hence the discrete integration algorithm is performed in the Fortran code. Somewhat oddly, the discrete Fortran models run with a variable step integration algorithm. However the Simulink solver detects that the Fortran code is discrete and needs only be computed at fixed update intervals.

To run the models

Step 1: In the Matlab command window, run the command *mex –setup* to identify the Fortran complier installed on the system. The user may either ask Matlab to identify it or select from a list. This need only be done once.

Step 2: Again in the Matlab command window, run the command *mex filename.f simulink.f –v*, to build the Fortran code into a *'dll'* (direct link library), which Simulink can access. Other Fortran source or object codes can be included in the file list. In the above command *filename.f* should be replaced by the name of the file given to the GT engine model. The –v switch provides information about the build process.

Step 3: Open and run the Simulink model. Choose the s-function toolbox from the user-defined box. The block can be renamed to the name of the Fortran file to avoid confusion with any other s-functions.

Step 4: For the discrete model define the variable *Ts*. This is the value used by the Fortran code and defined in the text file *params.dat*. It defines the time step for discretization.

2.4. Comments

The workspace of Matlab should include all the files required by the Fortran source code. The s-function block includes a field to define *s-function parameters* or constant terms to be passed to the Fortran code. However it is not clear how to access these values within the Fortran and they were defined into a text file as an alternative approach to pass the values as variables. The files which are opened should be closed to avoid Simulink from giving errors and crashing. Any write statements in Fortran should be to a file. The *'dll'* created cannot print statements on the screen in Simulink. The *tsampl* subroutine passes data about the Fortran code timing to Simulink but was not employed. Appendix A shows the example of integrating the Fortran source code of the two-shaft gas turbine, which was used to create the *'dll'* which was integrated into the electrical network.

3. GAS TURBINE PERFORMANCE SIMULATION STEADY STATE MODELLING

The design point of a gas turbine is usually defined as the operating point of the engine for a maximum power output with desired cycle efficiency under sea level static (SLS) conditions of the international standard atmosphere (ISA). The GT engine performance is the function of its cycle parameters. In the conceptual design phase, a model that works over a wide range of operating conditions and over large parametric variations in engine design is generally aimed for. True and validated critical inputs such as component maps and control schedules are usually not available at this stage of design. Generalized engine models use alternate methods that work with available data of component maps and operate simplified assumptions such as constant component efficiency, fixed pressure losses, choked turbines and a fixed geometry of blades etc at all working points to compute the GT engine steady-state performance. These types of engine models are not expected to reproduce the performance of any specific engine cycle. The primary purpose is to quickly estimate the engine performance during parametric studies, with reasonable levels of accuracy. Hence a robust framework with emphasis on parametric studies for configuration definition, preliminary sizing and rough performance outputs is required at the beginning. Estimated preliminary design performance values within a range of ±5% if the characteristics of the engine components and precise DP data are unknown can be a suitable starting point. However, accurate representation of GT performance for any dynamic analyses should be within a tolerance of ±1%. The large errors occur only at a few discrete points, like at low power values or if out of the range of the available component maps. These points may not form part of the typical journey profile, and are tackled as standalone models sometimes. Any loss in accuracy would be equally reflected in all of the parametric design combination[22]. Thus, an initial baseline design model to identify the regions of the optimum solution is a good starting point to simulate the performance of GT engines.

The current work involves digital simulation of GT engines to not only establish a preliminary design but a complete performance model of the majority of the journey profile. The digital simulation has to be engine specific and must be able to accurately represent its behaviour. The model also needs to incorporate the off-design and transient performance of the engine. As a part of the AMEPS project the digital simulation also needs to interact in a two-way path with the electrical network models. The simulated model uses component maps received from personal sources for representing component behaviour. Besides experimental data, component maps can also be generated using theoretical analysis of available data in the public domain. A brief description of the analysis will mentioned further. Major parameters such as compressor pressure ratio, mass flow, spool speed, turbine entry temperature, and exit pressure are either chosen from a prototype or plainly assumed based on intelligent logic. The component efficiencies are assumed based on the current technological level. Thus the digital simulation of an engine is a highly useful tool to develop the control schedule under which it can operate at the optimum condition, with an adequate surge margin and power levels which meet the mission performance.

3.1. Introduction to Modelling Techniques

The gas turbine engine has developed over the years as the primary source of aircraft propulsion systems. Although the first industrial gas turbine unit was installed in 1939 in Neuchatel, Switzerland, pioneering research on design had been carried out usually on aircraft gas turbine engines. With the arrival of aero-derivative engines for land based applications the literature review on turbojet, turbofan and other types of GT engines is helpful in understanding the design point modelling techniques. The design point modelling of a GT unit is a mathematical procedure which involves a stepwise process of solving equations either simultaneously or iteratively. Each component of the GT has a set of thermodynamic variables which represent its behaviour. The process of finding these values using computing power can be termed as digital simulation of GT. In the early days of simulation the simplest representation of a gas turbine was by assuming the engine to be a linear system. Although such a model could be solved by standard mathematics, which was accurate for small perturbations, it failed to show the behaviour of gas turbines over the whole range of its operations. At the time when computing technology was also in its developing stages, any small perturbation arising from a change in the operating point of the engine was a very time consuming process for engine remodelling. The task was also hindered as it required inputs from previous analyses or experimental tests, which completely destroyed the idea of a preliminary design. Linear models had been used anyways for control system studies due to lack of computing power to analyse systematic behaviour and difficulties occurred with secondary effects that completely undermined the control system developed on its basis.

A good DP model for initial calculations should include the following data:

- Availability of compressor and turbine maps.
- Initial values for DP compressor pressure ratio, TET, mass flow, spool speed.
- Total power output required with desired thermal efficiency.
- Typical values for component efficiencies.
- Design point ambient operating conditions. This is generally ISA SLS but can be different for engines whose sole working condition is otherwise.
- Any pressure losses, Mach numbers which have a significant effect on the DP simulation.
- In case of bleeds, logical assumptions of its measures. Typical values can be found from a gas turbine handbook.
- Type of fuel to be used in combustors represented by its calorific value.
- Any other variables that define the engine if it is different from the Brayton cycle.

The conditions mentioned above are non linear functions. Thus it would be highly inaccurate to represent an engine with a linear model.

A basic design point calculation to calculate the power does not require the use of powerful computers. In fact it can be simply done with the help of an ordinary calculator or an excel spreadsheet and is a good starting point to understand the working procedure of the GT. It gives a brief idea of the thermodynamic properties (temperature, pressure) of the gas turbine. However, a simple spreadsheet will not be able to calculate the off-design and transient performance of the gas turbine for the simple reason that such performance models require a set of equations to be solved and use a number of iterations to achieve the required outputs. Any change in the input parameters of the DP will also lead to different engine geometry at the fixed operating condition. For the concept design phase the component design points are usually at the same operating condition as the gas turbine DP. In detailed design phase this may not be true, which leads to a new set of requirements for DP modelling[23].

3.2. Literature Review of Gas Turbine Performance Models

When the gas turbine came into existence as a complete machine during the 1940s, its performance was evaluated for the steady-state and transient operation. As the majority of them were installed on aircraft, much of the initial literature focused on aero-engines. Till today majority of research is focused on gas turbine aero-engines with its derivative being used for other applications. Thus majority of the references quoted in this project might relate to GT installed in the aviation industry but stand true as an overall unit irrespective of its practise.

The concept of component models to analyze gas turbine performance has been the fundamental idea for performance prediction. A component modelling approach uses the characteristics of the major individual components of the engine to simulate and predict the overall power output and thermodynamic variables of the gas turbine. The behaviour of each of the GT components is well understood and analyzed. The relationship between the components is fixed by the physical layout and by identifying the necessary modules and connecting them appropriately by means of thermodynamic link represent the engine.

However, with the accessibility of today's modern computer it is fair to say that the restrictions which stood in the way of better models have been overcome and better and comprehensive methods have been used for simulations. Hung[24] has presented a mathematical model of a gas turbine generating unit to simulate its dynamic behaviour when the machine observes a system disturbance. Although the paper presented some theoretical background for simulating a GT unit, its main objective was to develop a better control system. The model was experimentally verified against test results obtained from a 13 MW GT generator and results were compared. This paper has been useful in understanding some modelling basics and control theory for systems simulation.

Gas turbine engine models generally deal with a lot of data. For the developer of the models it is necessary to have this information in numerical form, but from a user's point of view who at first instance does not require all of it, browsing through pages and pages of data can be tedious and unproductive. Thus for the past few years simulation work to an extent has been focused on creating friendly graphic user interface (GUI) models. The models on primary level allow the analysis of GT engines through a selection of few parameters as output. At the secondary and consequent levels they are very comprehensive models with the added benefit of visual information as operating trends at first level. This concept was implemented at NASA by Curlett and Ryall[25] to develop a GUI for propulsion system analysis. There are various computer codes developed at NASA to analyse the performance and weight characteristics of propulsion systems. Based on these codes and the programming benefits of the C programming language with some debate comparing it to other languages like FORTRAN, the authors provide a prediction tool for GT engineers. The principle, description, benefits and conclusions from the author's paper is a good starting point for any future work to modify the current code into a GUI. In 1995 when JAVA® was introduced as an object oriented programming language, it was recognised by many institutions as a better language. The benefits were recognised by the authors Reed and Afjeh[26] who started development of an interactive graphical "Java Gas Turbine Simulator". JAVA also has the advantage that it is supported by majority operating systems, and this software makes for a perfect tutorial on the simulation fundamentals of gas turbine engines.

Ganji, Khadem and Wilcutts[27] have developed "a general purpose simulation package for steady state and transient simulation of gas turbine based propulsion systems". Their code is split up in three modules: the GUI to "build" the engine layout, a Match module which performs engine matching (refer to chapter 3 for explanation) and a Dynamics module to analyse transient performance. The software known as "Gas Turbine Software (GTS)" is able to model a GT with or without a control system. The paper presents useful insight into modern day design methodology and control theories for transient simulation.

Different from the usual generic models, Bettochi, Spina, and Fabbri[28] concentrated on the modelling of a single-shaft GT using SIMULINK (Mathworks 1991) blocks in the software MATLAB®. Their work focuses on dynamic analysis and uses the concept of component modelling. In addition to the work carried out by the authors mentioned above, works of Koenig and Fishbach[29] and Pilidis[30] have been extremely beneficial in understanding the ideas and concept of gas turbine engine modelling.

A very simplistic design point performance model can also be obtained by using the set of non-dimensional or referred parameters which are explained further in this chapter and scaling them by the order required. Walsh and Fletcher[23] outline a complete chart with the various parameter groups which can be scaled. For this method type it is assumed that the geometrical dimensions of the gas turbine are changed linearly. Performance results achieved from linear scaling would be correct to first order accuracy giving us same values of the non-dimensional parameters; whereas parameters such as the mass flow, rotational speed, power output, efficiency etc. will change to reflect scaling. It is fair to say that performance trends when using this approach is similar to real figures, however the actual values can be completely untrue. For example, if a smaller engine were to be designed by simply scaling it down by half, the cost and manufacturing implications of producing power might be high or not possible at all by scaling the engine weight to the cubical value of the chosen factor. Having the benefit of quick transformation of engine design of the same layout, the disadvantages of the method far outweigh the advantages to be accepted as a performance code.

At the end of the current literature review it would be fair to say that majority simulation and modelling carried out in the gas turbine industry uses the component model approach. Various programming languages and platforms can be applied to model a GT based upon this concept, and it seems the logical method of choice and few advantages of it can be listed as follows:

- Estimates used at the initial design stage can be corrected as the simulation progresses.
- The model is valid over the whole range of engine operation and results from this model are truly representative of the engine's thermodynamic behaviour.
- The method can accommodate engine effects such as bleed flows and variable geometry.
- The framework of the individual components can be made flexible so as to allow higher complexity in order to analyze other effects like thermal capacitance, blade-lifing etc.

3.3. Design Point Parameters Selection

For an accurate and realistic gas turbine design point model, it is necessary to choose a set of constants that imply the characteristics of the individual components. The constants can be divided into groups of:

- Boundary conditions which have to be attained or cannot be exceeded
- Thermodynamics and fluid dynamics parameters like mean specific heats at constant pressure and at a constant volume, gas constant etc
- Technological constants which represent current levels of manufacturability for the engine
- Geometrics, such as characteristic volumes, areas and lengths where applicable

The fixing of two main thermodynamic variables, turbine entry temperature and compressor pressure ratio defines the design cycle completely for a fixed geometry engine[31]. However to avoid any significant changes in the simulation at a later stage it is vital to list all the possible constants, which may affect the design point simulation. The program should not only be able to provide the user with entries for operational envelop data, but also engine components design conditions.

To begin the simulation, the values of the DP parameters maybe read from a start-up data file or can be manually inserted at the beginning of each run. The former approach allows the programmer to implement changes without having to modify parts of the simulation code that use the constants. The comparative analysis of a given type of engine is also made easier by defining the set of constants. In addition to the mentioned constants, at the start of the simulation it is necessary to know all values in the initial steady state condition[28].

Table 3-1 shows the cycle design point data of the GTPS. A comprehensive list of initial conditions is shown in appendix A. It is based on values chosen from engine specifications of the GE LM2500[32] and WR-21[33]. Some data was selected based on open literature and intelligent guess. The original data was also modified to fit the interfacing needs with the network models.

Table 3-1 Design point data for the GTPS design point model

Type of variable/constant	Value
Operating pressure ratio (single compressor)	15
Combustor outlet temperature	1400 K
Component isentropic efficiencies	$0.87<\eta<0.90$
Ambient temperature	288 K
Ambient pressure	101.3 kPa
Gas generator shaft rotor speed/inertia	9800/150 kg-m^2
Power turbine shaft rotor speed/inertia	3600/100 kg-m^2

3.4. Component Characteristics Scaling

An accurate performance model of a gas turbine engine would involve use of original experimental data of component maps. Such data is obtained only after operating either the component or the complete engine on an actual test bed. The characteristics obtained after conducting such tests would be the property of the operators and/or of the contractors. As this project lies outside the scope of availability of such data, alternate methods to either obtain the maps or fit the available component maps in the public domain to the current project were reviewed.

Orkisz and Stawarz[34] list several potential sources of problems that can be encountered while modelling the compressor and turbine characteristics:

- Determination of the shape of characteristics since the compressor map at low speeds is nearly horizontal while at high speeds is nearly vertical and for turbines operating choked multiple lines of non-dimensional speed have the same non-dimensional mass flow, thereby presenting a problem.

- The non-uniqueness problem. Each compressor has a characteristic map which is its fingerprint. These are usually obtained from experimental data thereby creating difficulties in providing a numerical solution.
- The ill-conditioning problem or where some small changes in the variable of one coordinate produces large changes in the other coordinate variables and vice versa.
- The large variation encountered in the variables if the full envelope of the compressor or turbine characteristics is desired
- The differences in order of magnitude of variables (scaling problem) which is associated to the possible values of pressure ratios and mass flows currently possible.

The authors suggest a method to model the turbine and compressor characteristics which is based on the adaptation of the functions of component pressure ratios, mass flows, rotational speeds and rescaling the data to achieve a complete characteristic map. The proposed methodology is extremely useful when a small amount of original test data is available, to analyze and to simulate transient operation.

Another method to overcome this problem is a method referred to as component scaling. A typical compressor map provides information about the compressor pressure ratio π_{CMP} (eqn. 3.1), non-dimensional mass flow $\overline{m}$ (eqn. 3.3) and non-dimensional rotational speed $\overline{N}$ (eqn. 3.4). If the chosen compressor selected is to have variable geometry then for every given angle of inclination of its blade the characteristics are slightly different. In addition to the previously mentioned variables, a compressor map also contains information about its efficiency η. Similarly the turbine map represents its pressure ratio π_{TB}, non-dimensional mass flow $\overline{m}$, non-dimensional speed $\overline{N}$ and efficiency η.Thus the problem of choosing the right values of these variables to analyze the overall performance is made more complicated when working with 3-D graphs.

To define the component map at a given point, a numerical value is obtained from the following set of eqns (3.1 - 3.4):

$$\pi_{CMP} = \frac{P_{OUT}}{P_{IN}} \tag{3.1}$$

$$\pi_{TB} = \frac{P_{IN}}{P_{OUT}} \tag{3.2}$$

$$\overline{m}_{CMP/TB} = \frac{\dot{m} \cdot \sqrt{T}_{CMP/TB}}{P_{CMP/TB}} \tag{3.3}$$

$$\overline{N}_{CMP/TB} = \frac{N_{CMP/TB}}{\sqrt{T_{CMP/TB}}} \tag{3.4}$$

To scale the component characteristics, the values obtained from an external source like a performance deck or information gathered from public domain are first stored in a file as tabulated data. To eliminate the risk of uncertainty as to where the data is obtained from the parameters $\overline{m}$ and $\overline{N}$ are then corrected to standard atmospheric conditions giving us referred mass flow (eqn. 3.5) and referred speed (eqn. 3.6);

$$\overline{m}_{CMP/TB.REF} = \frac{\dot{m} \cdot \sqrt{\theta}}{\delta} \tag{3.5}$$

$$\overline{N}_{CMP/TB.REF} = \frac{N}{\sqrt{\theta}} \tag{3.6}$$

where,

$$\theta = \frac{T_{CMP/TB}}{288.15} \tag{3.6a}$$

$$\delta = \frac{P_{CMP/TB}}{101325} \tag{3.6b}$$

Equations 3.6a and 3.6b are typically used to describe overall engine performance referred to ISA SLS. Thus referred variables take the values that the non-dimensional parameters would have at standard atmospheric conditions. Mathematical methods of interpolation and extrapolation are then further applied to obtain a complete envelope of operation for the chosen component. This method would yield results close enough to real engine parameters and depending on the necessary complexity of the performance model, mathematical techniques of linear, quadratic or Lagrangian interpolation can be utilised. The maps obtained are characteristics based on scaling laws that change data from one component map into a new component map. Another advantage of this technique is that the computational power required is average and can be solved by any modern day personal computer. The scaling eqns. (3.7 – 3.10)[29] used for the compressor maps are:

$$SF_{\Pi} = \frac{\pi_{DP} - 1}{\pi_{MAP,DP} - 1}\left(\pi_{MAP} - 1\right) + 1 \tag{3.7}$$

$$SF_{\overline{m}} = \frac{\overline{m}_{DP}}{\overline{m}_{MAP,DP}} \times \overline{m}_{MAP} \tag{3.8}$$

$$SF_{\overline{N}} = \frac{\overline{N}_{DP}}{\overline{N}_{MAP.DP}} \times \overline{N}_{MAP} \tag{3.9}$$

$$SF_{\eta} = \frac{\eta_{DP}}{\eta_{MAP,DP}} \times \eta_{MAP} \tag{3.10}$$

The eqns. 3.7 – 3.10 are then used in off-design performance to obtain a suitable operating point on the characteristics. For the steady state and transient models in the current project, the characteristic were generated by in-house programs, known as COMPCHICS for compressor and TURBCHICS for the turbines.

3.5. Beta Lines

In choosing two variables to define the operating point, it is necessary that the variables must not be co-linear and that each pair of variables should produce a unique operating point[35]. In a common compressor map (figure 3.1), the speed lines for low power settings are nearly flat. To select a suitable point on the map which would represent the operating point in these circumstances becomes non-feasible. To alleviate this problem, fictional lines called as Beta lines are drawn on the compressor maps. These lines are drawn parallel to the surge line and any values can be assigned to them. General definition suggests that the surge line be numbered Beta = 1 with the order of magnitude decreasing as it moves away from it. The numbering has absolutely no effect on the performance prediction simulation of a gas turbine. These lines are indicated by the dotted lines in the Figure 3.1

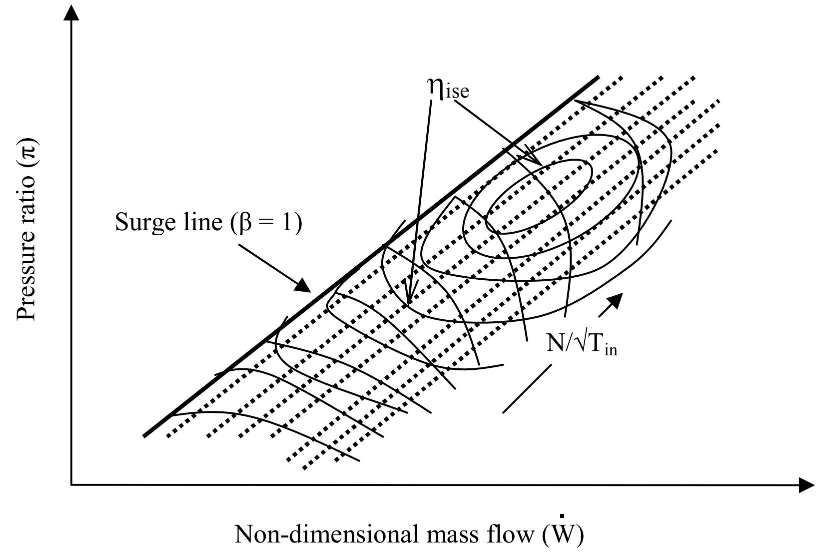


Figure 3.1 General representation of compressor map with beta lines

Defining betas eliminates the co-linearity of two points, but to define a point on the map the value of non-dimensional speed is added to our initial condition. Thus an operating point in now determined by three variables. When performing the off-design and transient calculation, an interpolation between the three parameters has to be made to achieve the desired number. This can increase the computational time marginally and compromise has to be made by adopting this method of representation during the transient performance simulation.

3.6. Gas Turbine Performance Simulation Model Program

The Gas Turbine Performance Simulation model is a computer code written using the fixed format of the Fortran language also known as FORTRAN 77. GTPS was used to model a single spool gas turbine with a free power turbine, popularly known as the Turboshaft engine in aviation or also the two-shaft gas turbine as it is called further in this project and a single shaft gas turbine engine. The general methodology suggested by Fawke and Saravanamuttoo has been deployed for GTPS and a source code which simulated the design point performance from the author's supervisor was provided. However modifications to this code were required so as to be able to perform GT performance prediction of two engine configurations. The engines chosen for analyses were a two-shaft gas turbine and a single shaft gas turbine. The two-shaft GT is currently utilised in marine propulsion systems as a prime mover, while the single shaft engine is the traditional type of gas turbines to be installed on industrial applications. These gas turbines propulsion systems for marine applications can be installed in various combinations, for example like a standalone unit as a prime mover coupled to the propeller shaft either through an electrical generator or through a mechanical transmission drive. Another way of using the gas turbine would be in a combined configuration with a steam or diesel powerplant on board. This method of application allows for flexibility in operational speeds of the marine vessel. If the gas turbine engines are used, like in the case of AMEPS, for an electro-mechanical propulsion system, then the two-shaft configuration has the benefit of the power turbine shaft providing for the variable speed load if it is connected to the propeller. The response of the gas turbines is similar if the turbine is in co-ordination with the electrical network.

A program line in a FORTRN 77 version extends to 72 spaces, the first 6 of which are either empty, used to define line numbers or to continue a statement from a previous line. Basic programming practise prescribes to start the code with the statement *implicit none* and declaring all the variables with a type. This eliminates the risk of working with non real numbers as an error is indicated by the compiler upon program execution. The values of constants and cycle parameters are defined in a text file. The engine cycle can be modified by changing the values in this file before any further changes to the code. Characteristic maps of the compressor and turbine are stored in another file, whilst the results of engine performance are also output as a text file.

Table 3-2 File management for the two-shaft engine model

Filename	File description	Note
YDpt11.DAT	Design point data of GT	Later data included for transient modelling. Renamed to YDpt11TS.DAT for GT-EN interface
YODP13.DAT	Data for off-design block	Removed from the GT-EN interface
ZMAPS15.DAT	Compressor and turbine characteristic maps	Arranged into angles, betas and speeds
ZOUTDGN8.DAT	General diagnostics of engine performance	Outputs only if required very specific data
ZOUTDGN10.DAT	General diagnostics of engine performance	Outputs only if required various data
ZOUTGEN12.DAT	Design point and off-design results	Thermodynamic and performance output
ZOUTMAP16.DAT	Component characteristics printed	Written or scaled maps
ZOUTPLT14.DAT	Off-design performance output	Generic results at different engine power output levels.

Once the gas turbine is schematically laid out; to define the process at the beginning and the end of the components numbers are assigned to define the section. These numbers are called as station numbers and the industry tend to follow a set of rules by the Aerospace Recommendation Practise (ARP). Figure 3.2 shows the station numbers for the GTPS model. The simulation model of a performance code can run into tens of thousands of lines depending on its complexity and configuration. Finding an error in calculation becomes difficult if appropriate sections of the code are not properly defined. The station numbers can be used in this case when during the debugging of the code a variable with its station number can define the position of this error. The higher advantage of following a set of rules for numbering and sticking to it is the transfer of data unambiguously and quick modification of the code to fit newer configurations. Also as an initial guess of the mathematical procedure going wrong due to the physics process involved, the station numbers can be beneficial.

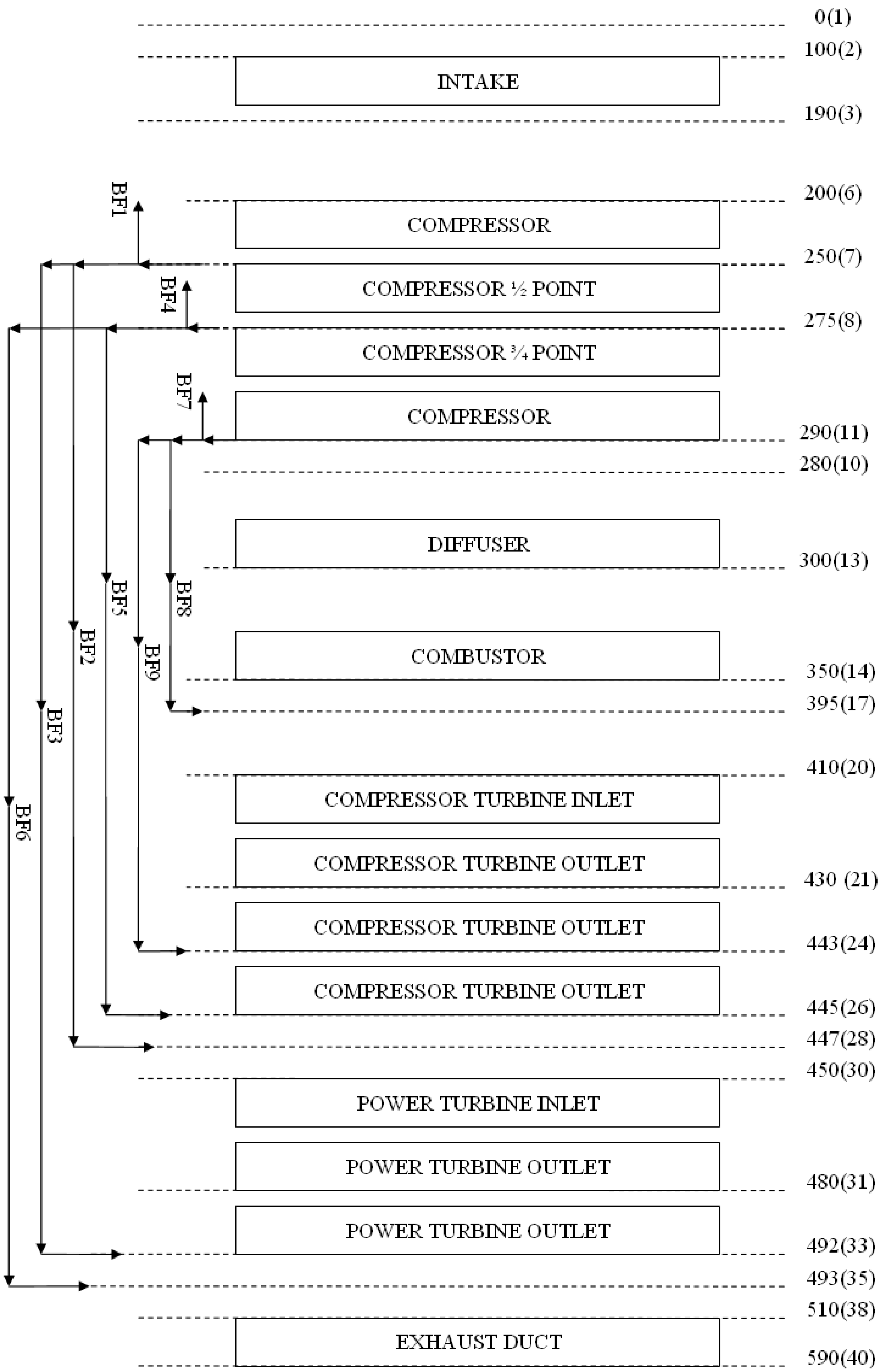


Figure 3.2 Two-shaft GTPS model station numbers and bleed flows

3.6.1. State Vectors

A component of GT can be thermodynamically completely defined by certain physical properties of the medium passing through it. These properties are called station vectors in gas turbine simulation codes. Although the air flows in different directions within the gas turbine, the term 'vector' is used here in the sense of an ordered set of values, without any directional significance[36]. Sets of data at various stations are defined and an individual component is comprised as an interface of inlet and outlet conditions. For any conventional type of simple cycle GT, the list of quantities outlined in Table 3-3 is sufficient.

Table 3-3 Station vectors for simple cycle GT

Vector No.	Station vector	Vector No.	Station vector
1	Station Number	7	Static Pressure
2	Fuel-to-Air Ratio	8	Total Pressure
3	Mass Flow	9	Static Temperature
4	Non-Dimensional Flow	10	Total Temperature
5	Velocity	11	Area
6	Mach Number		

By splitting the model into stations as previously mentioned, and assigning the state vectors the station numbers there is a tremendous scope of complex modelling to include effects of heat soakage, tip clearances, bleeds, variable stator vanes etc. Without following these two rules a performance simulation model would be very hard to create.

3.6.2. Bleed Flows and Customer Extractions

In a modern GT the turbine operates from relatively high to very high temperatures. Today's technology of manufacturing turbine blades employs various methods; like the use of composite alloys with high melting point as blade material or smart materials like temperature responsive polymers that can adapt its dimensions based upon temperature stimuli. These methods of production are not cheap and can significantly raise the price of the GT module. Prior to the advent of such engineering, turbine blades were able to withstand high temperature values by using cooling flow through the blades. The flows, extracted from various stages along the compressor called as a bleed flow, are taken typically from the rear stages of a high pressure compressor and are used to provide for the turbine cooling. Another major application of the bleed extractions in a GT during in-flight use of an aero-engine is to provide air conditioning for the passengers. This may also be an option in marine cruise ships.

Having bleeds in the engine is a compromise between engine performance and control of compressors. During the design point the bleed extractions add to the work necessary to drive the compressor. The returned bleeds for turbine cooling recover some of the work, but there is a loss of energy however. Depending upon the amount of bleeds extracted the loss in thermal cycle efficiency can be significant. This means a rise in the specific fuel consumption and maybe the operating costs of the gas turbine. The effects of bleeds are not always disadvantageous as it may sound. In the operation of a compressor, pressure gradients across the rotor blades help to achieve the pressure ratio across the compressor. The blades theoretically speaking should be designed such that maximum pressure rise is always gained across a stage. But due to some inefficiency in the airflow and compressor design can cause the flow to reverse, and the phenomenon of pressure pulsating front and back occurs, which is known as *surging*. To reduce the risk of these occurrences, more airflow can be allowed to pass through the compressor when required by opening up the bleed valves. This does not change the characteristics of the compressor map but allows the running line to shift away from surge. Thence, bleed flows at certain operating condition can ensure the safety of the GT.

When the GT performance code was created, besides cooling and air conditioning, compressor bleed flows were chosen to be implemented for the third function: to design an effective control system for the transient reaction of the engine. To decrease the response time of an engine during transient operation, bleed valves can be opened or closed, thereby regulating the air flow to achieve a desired time. In case of emergency dynamics a rush of airflow passing through the compressor can result in pushing the operating point beyond the mechanical limits of surge. The bleed valves can be utilised in this case and by closing some of them will bring back the point in the region of stability. Differently if there is an occurrence of deceleration, there can be a *flame-out* in the combustor. To prevent this, bleed valves can be opened to accommodate more air thus keeping the FAR within the working region of the combustor.

Summing up the bleed flows for all the above mentioned applications can be more that 10% - 15% of the compressor mass flow. Thus for an accurate performance model, bleed flows should be accounted for, though its use may not be necessary. Proper adjustments have to be made in terms of station numbering due to bleed flows and all station vectors should be calculated and accounted for across the change. Table 3-4 outlines the points at which bleed flows have been extracted and their appropriate return stations.

Table 3-4 Bleed flow data of the GTPS

Bleed from	Bleed to	Data value (% of mass flow)
SN 250 (C_{middle})	Overboard	BF1 = 0.5
SN 250 (C_{middle})	SN 447 (PT_{IN})	BF2 = 1.0
SN 250 (C_{middle})	SN 492 (PT_{OUT})	BF3 = 0.3
SN 275 ($C_{3/4th}$)	Overboard	BF4 = 1.0
SN 275 ($C_{3/4th}$)	SN 445 (PT_{IN})	BF5 = 0.2
SN 275 ($C_{3/4th}$)	SN 493 (PT_{OUT})	BF6 = 1.0
SN 290 (C_{out})	Overboard	BF7 = 1.0
SN 290 (C_{out})	SN 395 (HPT_{IN})	BF8 = 10
SN 290 (C_{out})	SN 443 (HPT_{OUT})	BF9 = 4

3.6.3. Data Transfer of Parametric Values

The GTPS program is split into subroutines, which can be considered to be a program on its own: a 'subprogram'. This kind of structural hierarchy is based upon the previously mentioned component models approach. Lots of links between parts of the code are typed and unlike the flow process in a gas turbine the links cannot be unidirectional. The addition of secondary effects like that of need for iterating when data is unavailable adds to the need of efficient program management. When a *CALL* is made to a 'subroutine' all the necessary data to execute it is transferred from the calling point. The two methods of data transfer that are used to execute the performance program are not the only ways of accomplishing the task of transferring information and returning to back to the point, but are the most common and safe procedures that ensures proper arrangement.

Method 1: Common blocks

All the required variables are listed using the COMMON statement. This command defines one or more contiguous areas, or blocks, of physical storage that can be accessed by any of the scoping units in an executable program. For best practise, it is wise to list the initial conditions and station vectors into one single common block. Then based on the type of subroutine other blocks can be defined. For the GTPS model of the two engines, the Table 3-5 indicates the common blocks utilised and describes the data within.

Table 3-5 Common blocks for GTPS

Common block name	Description
COMPM	Stores compressor map characteristics
AUXM	Contains auxiliary arrays for map operations
AUX1	Common block for any other auxiliary array
TURBM	Stores turbine map characteristics
DESPT	Information on design point variables values
ODP	Off-design performance variables
MASSFLW	Transfers data for mass flow adjustment at design point
TRANSIENTR	Contains real values of transient information
TRANSIENTI	Contains integer values for transients
FFCONTROL	Logical common block for control

Method 2: Direct transfer

Data can also be transferred directly from the call statement to the subroutine by listing them in within parentheses after the name of the subroutine. When it is not required to work with all the variables between the call statement and subroutine, this method is useful. The names of the variables can also be changed in the subroutine if necessary. For variables which are crucial in determining the output of the component, which describe the working process it is vital that they are completely unambiguous and rightfully found. By storing them together for each module, the data is accessed at the right place thus minimising the risk of mathematical errors. This method also assists in the debug procedure, where only the chosen set of variables display a figure while all the rest are undefined in the subroutine.

Figure 3.3 Methods of data transfer in GTPS

Figure 3.3 is a snapshot of the two methods for data transfer being applied in a CALL – SUBROUTINE setup. Both the methods can be combined, but the parameters cannot be repeated in the common block and within the parentheses. One important rule has to be followed for the common block that all variables referred are in the same order throughout all the blocks.

3.6.4. Design Point Subroutines: Function and Description

The design point performance model was modified keeping in mind the steps involved in calculating the performance. The calculations should start at the inlet and proceed subsequently to the flow path of the working fluid. Although certain assumptions of efficiencies, Mach number and other variables are made, it should be avoided to deviate from a typical gas path analysis mentioned in many references. The flowchart as shown in the Figure 3.4 represents the subroutines which are used to perform the DP calculations of the turboshaft engine. The region under the dotted lines follows the gas flow path within the GT module conforming to the station numbering in Figure 3.2, taking into consideration the effect of bleed flows. The process starts with a call to the subroutine READMAPS and continues downwards and then from left to right within the dotted line area.

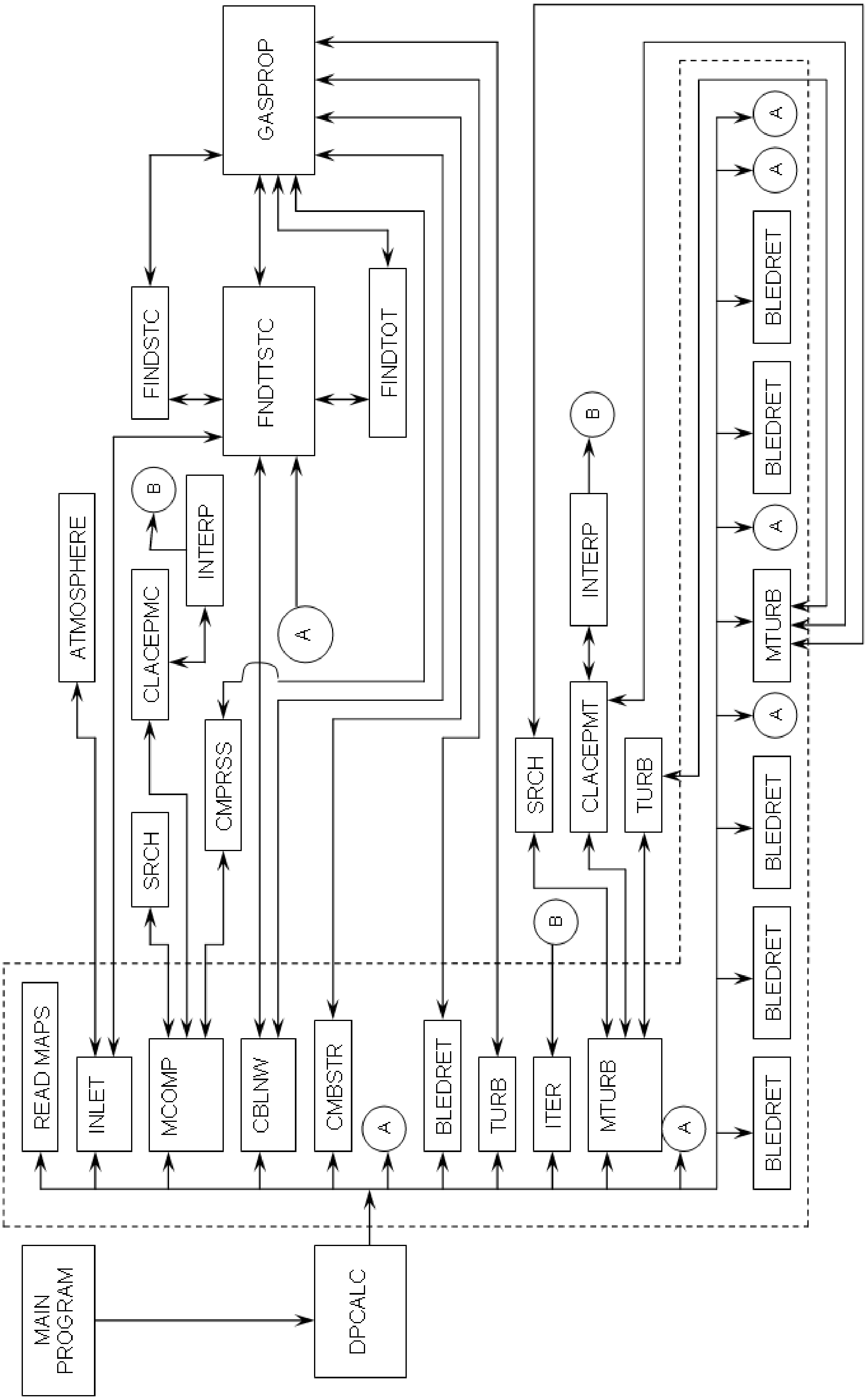


Figure 3.4 Flow chart of two-shaft and single shaft GTPS model

Based upon the computations occurring in the subroutines, they are split into purpose specific, generic and complex. A generic subroutine is one which carries out calculations which are purely mathematical, whilst the purpose specific subroutine is for a particular design point calculation. A complex subroutine would be one in which based on input parameters, various calculations are made to arrive at the result. In the TSGT model most of the subroutines are of complex nature. The individual function of each subroutine is listed below.

DPCALC: Main program that initialises the design point calculations. The very first step involves reading initial conditions of the design point from the data file 'YDpt11.DAT'. This subroutine is executed at the very beginning of the IFEP simulation as an initial condition.

READMAPS: Reads the compressor and turbine maps and aligns it systematically according to betas, speeds and angles.

INLET: Determines the ambient conditions at inlet. It calculates station vectors at inlet entry and inlet exit. Ram conditions of air are the inputs to this routine.

ATMOSPHERE: As a function of altitude and temperature deviation from ISA, calculates ambient pressure and temperature.

FNDTTSTC: Performs flow calculations to work out the temperature (total and static), pressure (total and static), flow velocity, Mach number and flow area. This subroutine requires three inputs from the aforementioned values in addition to the flow rate to calculate the remaining values.

GASPROP: Calculates the physical properties of the gas. Using the inputs of temperature and the fuel-to-air ratio, outputs of specific heat capacity, heat capacity ratio, specific enthalpy and entropy are provided.

FINDTOT: Given flow rate, static temperature and pressure and Mach number finds total temperature, total pressure, flow velocity and area. In ambient conditions or zero Mach the static and total variables are the same with a null velocity and area is considered infinite

MCOMP: Calculates the compressor module within the GT program. The maps which have been read are scaled according to the requirements of the non-dimensional speed, beta and stator angle position of the design point.

SRCH: Locates the array coordinates of a value of a chosen variable larger and smaller than the variable. The possible choice is between angle, speed and beta.

CALCEPMC: Given the values of angle, speed and beta performs interpolation to find design point or desired pressure ratio, non-dimensional mass flow and isentropic efficiency of the compressor.

CMPRSS: Subroutine to calculate properties associated with compression of air. The result obtained is the outlet parameters of the compressor module.

INTERP: Routine to perform either linear or quadratic interpolation.

CBLNW: Subroutine to do the bleed extractions and work calculations needed after the compressor. Compressor work calculated includes work done to extract bleed flows.

FINDSTC: Subroutine to determine static temperature and pressure, mach and area using inputs of total temperature and pressure, mass flow and velocity.

CMBSTR: Performs calculations of properties associated with combustion of fuel. The subroutine requires the input of either the fuel flow or COT. In the case of design point it is the COT and for off-design performance either one can be chosen

BLDRET: Returns a bleed to the main stream and performs a mass and energy balance.

TURB: Calculates the physical properties of gas when expanded through a turbine. Output is the turbine work and turbine exit values.

ITER: Subroutine to perform iteration. Given two variables: x_i and x_j iterations are carried out so that x_i is achieved satisfying the numerical condition of x_j.

MTURB: Performs calculations for the turbine module within a GT. Scaling of turbine map is executed based on scaling factors at the design point.

CALCEPMT: Given the values of angle, speed and beta rearranges look-up tables and performs interpolation to find pressure ratio, non-dimensional mass flow and isentropic efficiency of the turbine.

3.7. Two-shaft gas turbine engine description

The two-shaft gas turbine is split into elementary modules, modules in which there is a change of either state, mass or energy. The design point is then performed by linking them appropriately based upon the schematic in Figure 1.2. Typically the pressure ratio of the compressor and COT will define the design point of the engine. The main components of the engine were modelled based on performance theory described further.

3.7.1. Intake: Routines INLET and ATMOSPHERE

The intake is an important and sometimes critical part of an engine although there is no transformation of state and no work is involved. The idea of design of an intake is to keep the pressure loss minimum as it has an effect on the cycle efficiency. To prevent any mechanical damages caused as a result of distorted or non-uniform mass flow the intake will keep the flow velocity constant described in GTPS as constant Mach number.

Intake module is initialised by a call to the INLET subroutine. This routine basically fixes the atmospheric conditions at which the engine is designed. ATMOSPHERE can be used to calculate the ambient temperature and pressure of air based on equations from reference [23] given as follows: for ambient temperature as a function of altitude (metres), relationships (eqns. 3.11 – 3.13) stand true in this model

$$0<ALT<11000 \qquad t_{amb} = 288.15 - 0.0065 \cdot ALT \tag{3.11}$$

$$11000<ALT \qquad t_{amb} = 216.67 \tag{3.12}$$

$$24994<ALT<30000 \qquad t_{amb} = 216.67-0.0029892 \cdot (ALT-24994) \tag{3.13}$$

Ambient pressure (kPa) with respect to altitude and temperature is given by eqns. 3.14 – 3.16

$0<ALT<11000$ $\quad p_{amb} = 101.325\cdot(288.15 / t_{amb})^{-5.25588}$ (3.14)

$11000<ALT$ $\quad p_{amb} = 22.63253 / e^{0.000157689\cdot(ALT-10998.1)}$ (3.15)

$24994<ALT<30000$ $\quad p_{amb} = 2.5237\cdot(216.65 / t_{amb})^{11.8}$ (3.16)

The equivalence derived is based on ISA SLS. If t_{amb} is different from standard then by adding that difference (ΔT) to the new value the initial condition at inlet entry are obtained.

3.7.2. Compressor: Routines MCOMP and CMPRSS

The elementary 'compressor' module is not a single module which performs the compressor DP. The subroutines MCOMP & CBNLW together provide the outputs at compressor outlet. Foremost depending on the choice of beta, non-dimensional speed and stator angle, subroutine MCOMP finds the compressor pressure ratio, non-dimensional mass flow and isentropic efficiency. The maps are then corrected and scaled as per the scaling factors to provide for the GTPS to conduct an off-design and transient analysis.

To evaluate the complete set of physical properties at the compressor outlet and subroutine CMPRSS is called. The GTPS model takes into account the presence of compressor bleeds and total compressor work necessary to drive it, a call to CBNLW is executed. In subroutine CMPRSS eqns. 3.17 and 3.18 make it possible to calculate, respectively, outlet compressor temperature and compressor pressure, while the total compressor work including the work required for bleed extraction is calculated by the eqn. 3.19

$$\eta_{CMP.ISE} = \frac{\pi_{CMP}^{\frac{\gamma-1}{\gamma}} - 1}{\frac{T_{CMP.OUT}}{T_{CMP.IN}} - 1} \qquad (\gamma = 1.4) \tag{3.17}$$

$$P_{OUT} = P_{IN} \cdot \pi_{CMP} \tag{3.18}$$

$$W_{CMP.TOT} = W_{CMP} + W_{BL} \tag{3.19}$$

where,

$$W_{CMP} = \dot{m}_{CMP.IN} \cdot (H_{OUT} - H_{IN})_{CMP} \tag{3.20}$$

$$W_{BL} = \dot{m}_{CMP.IN} \cdot (H_{OUT} - H_{IN})_{BL} \sum \dot{m}_{BL} \tag{3.21}$$

$$H \ = h + \frac{v^2}{2} \tag{3.21a}$$

3.7.3. Combustor: Routine CMBSTR

For the design performance of combustors, its corresponding module in GTPS is CMBSTR, which is considered as a simple duct where fuel is mixed with air from the compressor and ignited to give a rise in temperature. At DP, the aim is to find the fuel necessary to achieve the temperature rise for a chosen COT.

For a fully rigorous calculation the effects of specific enthalpy has to be considered and an iterative process has to be performed to arrive at the right fuel flow. The COT is one of the input parameters of the engine cycle and an intelligent guess of fuel is made by the eqn. 3.22.

$$\dot{m}_{FL.GS} = \frac{\dot{m} \cdot 1150 \cdot \left(T_{CC.OUT} - T_{CC.IN}\right)}{LHV \cdot \eta_{CC}} \tag{3.22}$$

The fuel-to-air ratio is then solved in eqn. 3.23 from the guessed fuel flow.

$$FAR_{CC.OUT} = \frac{\left(FAR_{CC.IN} \cdot \dot{m}\right) + \dot{m}_{FL}}{\dot{m} + \dot{m}_{FL}} \tag{3.23}$$

This ratio does not consider the effects of specific enthalpy so it has to be updated. To do this the subroutine GASPROP is called and specific enthalpy at combustor exit is found. Set upon this value, a new rate of fuel input from eqn. 3.24 is calculated.

$$\dot{m}_{FL} = \frac{H_{CC.OUT} - H_{CC.IN}}{LHV \cdot \eta_{CC}} \cdot \left(\dot{m} + \dot{m}_{FL.GS}\right) \tag{3.24}$$

Now again amend eqn. 3.23 for an enthalpy corrected FAR. To reach at the final value of fuel flow iteration is performed between 3.23 and 3.24 and the latter equation together with GASPROP executes the fuel flow for a given COT. It has been assumed that fuel enters at the air temperature and dissociation is ignored. In the case of high turbine entry temperatures the effect of dissociation in the combustion chamber has to be considered as typically the fuel flow required is higher. The new values of fuel is then also dependent on the pressure which has to be introduced in the chart of combustion temperature rise and FAR.

3.7.4. Turbine: Routines MTURB and TURB

The expansion of gas occurs in the turbine adiabatically. Though there is mass flow returned into the turbine module, the physical properties at its outlet do not take into consideration the effect of mass flow variation. Mixing of the bleed flow with the main gas path is focused upstream and downstream. By assuming this scenario and the isentropic efficiency of the turbine, find the output temperature and pressure from the eqns. 3.25 and 3.26

$$T_{TB.OUT} = T_{TB.IN} - \frac{W_{CMP.TOT}}{C_P \cdot \dot{m}_{IN}} \tag{3.25}$$

$$P_{TB.OUT} = P_{TB.IN} \left(\frac{T_{TB.IN} - \dfrac{T_{TB.IN} - T_{TB.OUT}}{\eta_{TB.ISE}}}{T_{TB.IN}} \right)^{\frac{\gamma}{\gamma - 1}} \tag{3.26}$$

Where γ = 1.33 and is true in the case of the turbine connected to the compressor. The equations are then corrected based upon the properties of specific enthalpy and specific heats. Again this is done with the help of GASPROP. The turbine maps are then referred to find the mass flow and turbine efficiency and scaled to the design point parameters. The above mentioned equations are solved by the subroutine TURB.

In the case of the power turbine, based on standard practise the DP model starts at the exit where its pressure equals to pressure at ambient conditions and together with the previous set of known properties, the expansion ratio is found. The turbine is considered to be choked. From the turbine map, with the known expansion ratio, speed line and beta calculate the station vectors from the non-dimensional mass flow from eqn. 3.3.

The two-shaft gas turbine has a free turbine which provides power output from the complete cycle. Essentially it is a turbine in which expands gas and solution is executed by the subroutine MTURB which gives the work done by a turbine. The work output in MTURB is the turbine work (eqn. 3.27) obtained from the last calculation. This work in case of the gas generator is equal to compressor work and for the power turbine it is useful work.

$$W_{TB} = \dot{m}_{TB} \cdot (H_{OUT} - H_{IN})_{TB} \tag{3.27}$$

The overall cycle thermal efficiency is then given as, where the numerator is the useful work created by the power turbine;

$$\eta_{TH} = \frac{W_{PT}}{\dot{m}_{FL} \cdot LHV} \tag{3.28}$$

Thus for the power turbine the solution starts at subroutine MTURB to define its operating point, which provides the pressure drop across it and the non-dimensional mass flow. The formulae 3.2, 3.3, 3.25 and 3.26 define the station vectors at its inlet and outlet and eqn. 3.27 calculates the useful work which is the power output of the GT at its design point.

3.7.5. Other important routines: Routines CBNLW and BLDRET

When there are bleeds extracted from the compressor or air is supplied for cooling of the turbine blades, there is a change of mass flow between the component's inlet and outlet. This creates an imbalance in work between the point at which the fluid flows along the normal flow path and the point at which bleed is returned. For steady state modelling, the conditions of mass, energy and momentum balances need to be satisfied. The balancing equations for mass flow and energy, which are solved to match the inlet and outlet, are respectively shown in eqn. 3.29 and 3.30. The effect of momentum unbalance is neglected for the current design point model.

$$\dot{m}_{OUT(i)} = \dot{m}_{CMP.IN} \cdot \dot{m}_{BF(j)} + \dot{m}_{IN(i)} \tag{3.29}$$

$$H_{OUT(i)} = \frac{\dot{m}_{IN(i)} \cdot H_{IN(i)} + \dot{m}_{CMP.IN} \cdot \dot{m}_{BF(j)} H_{BF(j)}}{\dot{m}_{CMP.IN} \cdot \dot{m}_{BF(j)} + \dot{m}_{IN(i)}} \tag{3.30}$$

Where, *i* is the station at which the bleed flow is returned and *j* is the percentage of mass flow extracted as bleed. All interior bleeds for cooling of turbine blades and any overboard bleeds are modelled at the compressor inlet flow rate. The routine CBNLW performs the calculations to determine the state variables at the inlet and exit stations of bleed and the total work required to perform the task of air removal. This work is given by the eqn. 3.21.

When the bleeds are returned into the gas flow path, the mass flow and energy figures are equated at the corresponding stations by solving the aforementioned equations. These equalities are executed in subroutine BLDRET. The turbine nozzle guide vanes are cooled with the bleed air and only the upstream returned flow adds toward the total work output in the turbine.

3.7.6. Routines FNDTTSTC, FINDTOT, FINDSTC

Another important part of the GTPS is the subroutine FNDTTSTC. At various points along the model the physical properties are either calculated or obtained in their static or total form. The set of variables which are station vectors have both types of these parameters. Mass flow is known throughout the steady state modelling. Now by assuming Mach number values for the stations the list of station vectors are fulfilled between the three subroutines. The options available in FNDTTSTC are listed in Table 3-6

Table 3-6 Operations available for calculation of thermodynamic variables

Operator mode value (NCALC)	Given variables	Calculated variables
1	$T\ ,P\ ,\dot{m},A$	t,p,v,M
2	$T\ ,P\ ,\dot{m},M$	t,p,v,A
3	$T\ ,P\ ,\dot{m},v$	t,p,M,A
4	$t,p,\dot{m},A$	$T\ ,P\ ,v,M$
5	$t,p,\dot{m},M$	$T\ ,P\ ,v,A$
6	$t,p,\dot{m},v$	$T\ ,P\ ,M,A$
7	$T\ ,P\ ,\dot{m},t$	p,v,M,A
8	$T\ ,P\ ,\dot{m},p$	t,v,M,A

This process is split in two subroutines; FINDTOT given the mass flow, static temperature, static pressure and Mach determines the variables of the fifth row in the above table and FINDSTC provides the static temperature and pressure, Mach and flow area corresponding to the inputs of the third row in the same table. By solving the set of eqns. 3.31 – 3.35 based on inputs mentioned in column 2 of the table find the necessary values required in the next column.

$$M = \frac{v}{\sqrt{\gamma \cdot R \cdot t}} \qquad (3.31)$$

$$\rho = \frac{p}{R \cdot t} \qquad (3.32)$$

$$A = \frac{\dot{m}}{\rho \cdot v} \qquad (3.33)$$

$$t = T\ - \frac{v^2}{2 \cdot C_P} + \left(h_{IN} - h_{OUT}\right) \cdot \Delta T \Delta H \qquad (3.34)$$

$$S - s = \int_t^T C_P dT + R \ln \frac{P}{p} \qquad (3.35)$$

where,

$$\Delta T\Delta H = \frac{T \quad - t}{H \quad - h} \tag{3.36}$$

3.7.7. Routine GASPROP

The routine GASPROP is created to evaluate the gas properties C_P, γ, H and S for any given temperature and FAR. The process of evaluating the thermodynamic properties at the station is by iteration of the temperature and H, which is also a function of temperature. By making use of the equations for temperatures, pressures, efficiencies, etc. and looping the process of its calculations with the properties of gas that are derived from the polynomials provided in [Walsh & Fletcher, 1998] the results obtained are accurate (within <0.5%) of GT engine performance.

3.8. Single shaft GTPS model for design point

Another aim of the author's work included the performance simulation of an open-cycle single shaft gas turbine engine to be incorporated into the GT-EN interface of an IFEP system. In common literature this configuration is given by Figure 3.5.

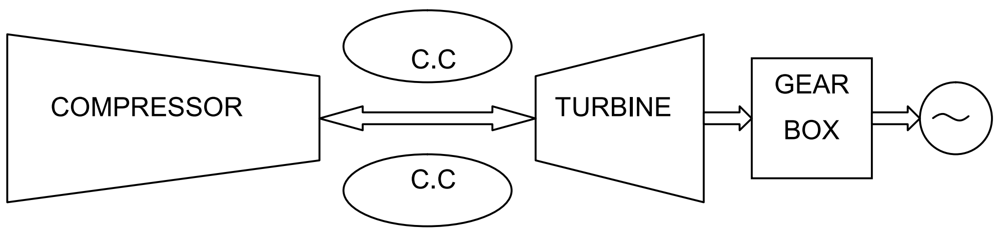


Figure 3.5 A typical single shaft gas turbine

Its typical layout is the intake, followed by the compressor, the combustion chamber to gain a temperature rise and the turbine which expands the air mixed with fuel. There is generally an exhaust duct at the exit of the gas generator, but this is typically to direct the exhaust gases into the atmosphere than use it for any work like in the aircraft engines. This type of GT configuration is called as a single shaft GT because there is just one shaft connecting the compressor and turbine in the engine layout. In some engines there is an additional booster on a single shaft with the primary compressor. However, the configuration can be modified to include multiple gas generators and the engine is known as a multi-spool GT.

The single shaft gas generator in industrial and propulsion systems can be connected to another mechanical drive, like to the rotor of a helicopter, to the propeller of a ship or as in the case of IFEP to an electrical generator via a gearbox. The term *'gearbox'* in this context is used vernacularly to indicate a complicated transmission drive. A gearbox provides a speed difference between the two shafts that it connects and is torque-force relationship. Thus given two speeds define the gearbox by a gearbox ratio to change shaft speeds from N_1 to N_2 by eqn. 3.37.

$$GBXTR = \frac{N_1}{N_2} = f(\tau, F) \tag{3.37}$$

This gearbox ratio was adopted into the previous two-shaft engine model to simulate the single shaft GT and it was incorporated into the model represented by the Figure 1.4. This ratio does not have an effect on the design point, but is more of an off-design and dynamic factor for simulation. By assuming this relationship, the previous model has been completely retained and with just a few changes in initial conditions a new engine configuration was attained.

4. OFF-DESIGN GAS TURBINE PERFORMANCE SIMULATION MODELLING

A gas turbine engine is rated based upon the design point parameters. When an engine is installed for any application, majority of time its working regime is out of these initial conditions. Thus it is extremely essential to understand the process when there is any form of deviation in the input to the gas turbine. The evaluation of the thermodynamic and physical properties at such points is termed as off-design performance. Depending upon the user of the GT an off-design condition can be almost any variable that defines the behaviour of the gas generator. However, typically an OD performance model is created for the following reasons:

- Analyse the effects of ambient conditions (temperature and pressure) on cycle parameters.
- Determine the limits of operation (surge line).
- Understand and/or predict the part-load performance of an engine.
- To aid with the development of control systems.
- Examine the behaviour of individual components and optimize performance.
- Comparative analysis of different engine types.

4.1. Methodology for OD prediction

Contrary to the DP modelling, where most of the parameters are defined or assumed and simple mathematical derivation can be applied to get the results, in off-design performance there are many unknowns which have to be worked out before a result can be reached. The modelling methodology consists of reaching a steady state point through a multi-dimensional simulation of the flow and flow interaction with the components in a GT. To perform a multi-dimensional simulation a repetitive process of solving equations which represent the state of the component is executed, until a convergence is reached. When the operating point is at steady state, there is a perfect balance in the mass flow, speed and work ensuring continuity between the compressor and turbine. If for any particular reason there arises a difference then to regain a steady state the flow has to be matched by nullifying the difference in mass flow across the two adjacent stations, which are used to represent the GT modules and the condition of compressor work equalling to its appropriate turbine work has to be met. Figure 4.1 illustrates the off-design concept based upon the above mentioned philosophy. The off-design GTPS methodology describes a way to tackle the problem of mismatch and provides a literary background on OD performance simulation.

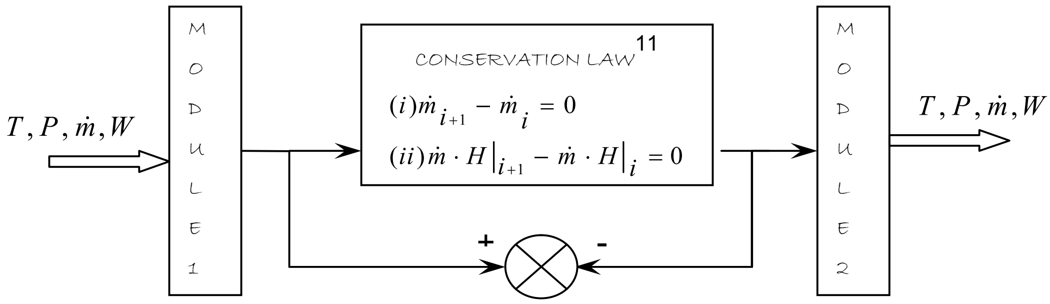


Figure 4.1 Off-design concept for GTPS

4.1.1. Selection of control variables

For OD operating points, flow matching needs to be performed. A point on the characteristic map of any component gives the physical properties which satisfy only one condition of the engine operation. Once there is a mismatch of any parameter that does not satisfy the laws[11] of conservation of mass (eqn. 4.1), conservation of momentum (eqn. 4.2) or conservation of energy (eqn. 4.3) the whole set of engine variables change.

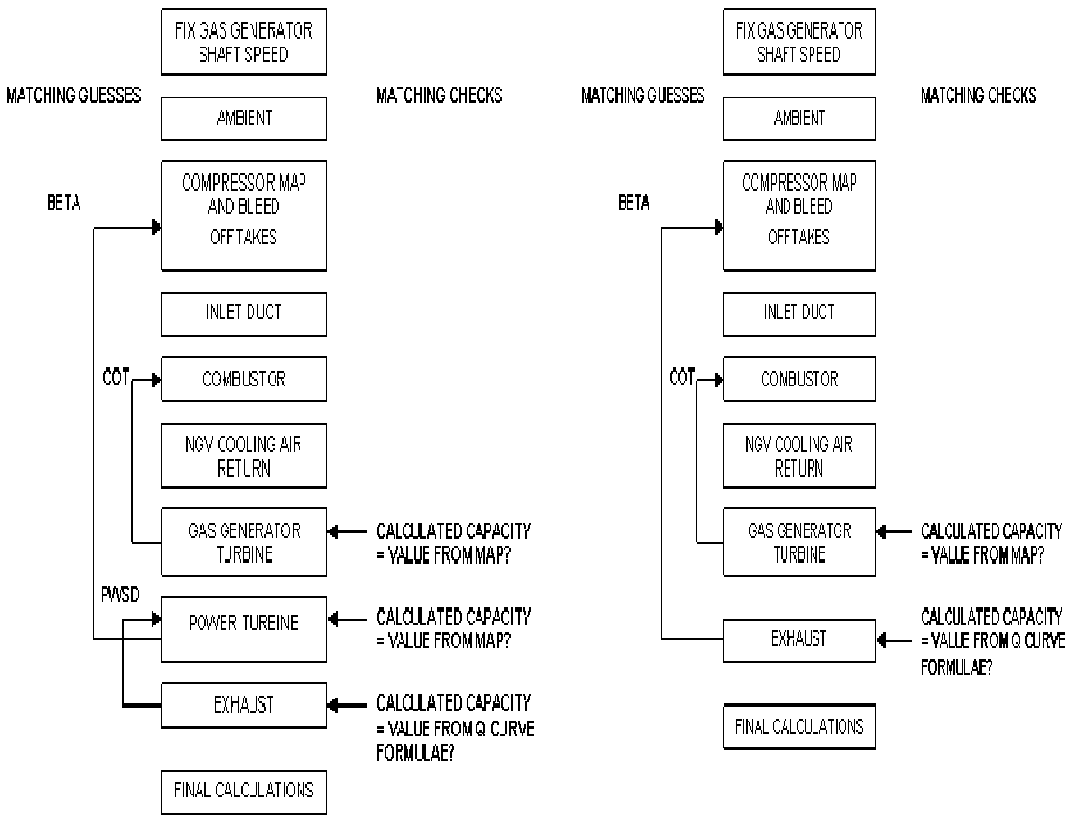
Mass flow: $$\dot{m}_{i+1} - \dot{m}_i = 0 \quad (4.1)$$

Momentum: $$\dot{m}_{i+1}v_{i+1} - \dot{m}_i v_i + p_{i+1}A_{i+1} - p_i A_i - F = 0 \quad (4.2)$$

Energy: $$\dot{m}_{i+1}H_{i+1} - \dot{m}_i H_i - \dot{Q} + W_{TOT} = 0 \quad (4.3)$$

To alleviate this mismatch, a new set of parameters have to be defined and the difference between the previous values and the new values between the respective components is called ‘error’. Although all the parameters in the engine will change at OD, a few ‘errors’ are chosen based upon the type of

GT engine configuration, complexity of the simulation model and accordingly the new operating point is defined by a set of independent 'control parameters' that are guessed. The number of control variables equals the number of errors. The control variables are guessed at the first iteration and then subsequently solved to get a result, very much likely not the correct one. The solution procedure takes form of a design point performance calculation. Accurate performance at off-design conditions generally takes a number of iterations between the guess and checks of control variables to converge. This is the so called 'matching' of gas turbine engine. For the two-shaft gas turbine Figure 4.2 (a) suggests the matching process and Figure 4.2 (b) is suitable for the single shaft configuration.



(a) Two-shaft gas turbine (b) Single shaft gas turbine

Figure 4.2 Engine matching diagram for off-design performance[23]

Table 4-1 provides the reader with some suggestive ideas of the control variables and the checks required for two types of engine configurations. The intercooled-recuperated engine is suggested because it is an advanced cycle GT. The two-shaft GT engine can be modelled with a nested loop approach of iterations following a pattern of solving, while it would be best to use a multi dimensional iterative analysis to model an ICR or a novel cycle engine.

Table 4-1 Guesses and corresponding checks for two engine types

Engine	Guessed variable	Matching constraints
Two-shaft GT	$Beta_{PT}$	$P_{PT.OUT} = P_{AMB}$
	$Beta_{TR}$	$NDMF_{TR.OUT} = NDMF_{PT.IN}$
	$Beta_{CMP}$	$NDMF_{CC.OUT} = NDMF_{TR.IN}$
	N_{TR}	$W_{TR} = W_{CMP}$
Intercooled	$Beta_{LPC}$	$NDMF_{LPC.OUT} = NDMF_{HPC.IN}$
Recuperated GT	N_{LPC}	$W_{LPC} = W_{LPT}$
	$Beta_{HPC}$	$NDMF_{CC.OUT} = NDMF_{HPT.IN}$
	N_{HPC}	$W_{HPC} = W_{HPT}$
	$Beta_{HPT}$	$NDMF_{HPT.OUT} = NDMF_{LPT.IN}$
	$Beta_{LPT}$	$NDMF_{LPT.OUT} = NDMF_{PT.IN}$
	$Beta_{PT}$	$P_{PT.OUT} = P_{AMB}$
	T_{HX}	$T_{HX.OUT} = T_{HX.CALC}$

The list can be expanded or compressed depending on the engine layout and statistical or physical analysis aimed for. If the simulation is at a development process then it is best to keep the variables to a minimum. To realise this certain guesses can be taken as constant values to be input as and when needed. When the project proceeds to take into account secondary and tertiary effects, guesses and/or constants can be added. The work of simulation is made complex when there is use of many variables. It is complicated to derive flow logic of the gas path and the answer lies in smart solving techniques of simultaneous equations.

4.1.2. Error Solution Techniques

The gas turbine engine is assumed to be at a steady state when the guessed values of control variables satisfy the corresponding check equalities. The procedure of satisfying the above set condition is by iteration. When in a single iteration each constant variable is linked to one particular error and calculation is proceeded to minimize only one error at a time, a nested-loop approach has to be adopted. By this method, OD performance can be evaluated either by performing one guess followed by one check which is the concentric iteration in Figure 4.3 or one can follow the crossover iteration method by which repetition can be performed how and when needed. There are no apparent advantages or disadvantages of one method over another because both of them require more than one single loop of calculation.

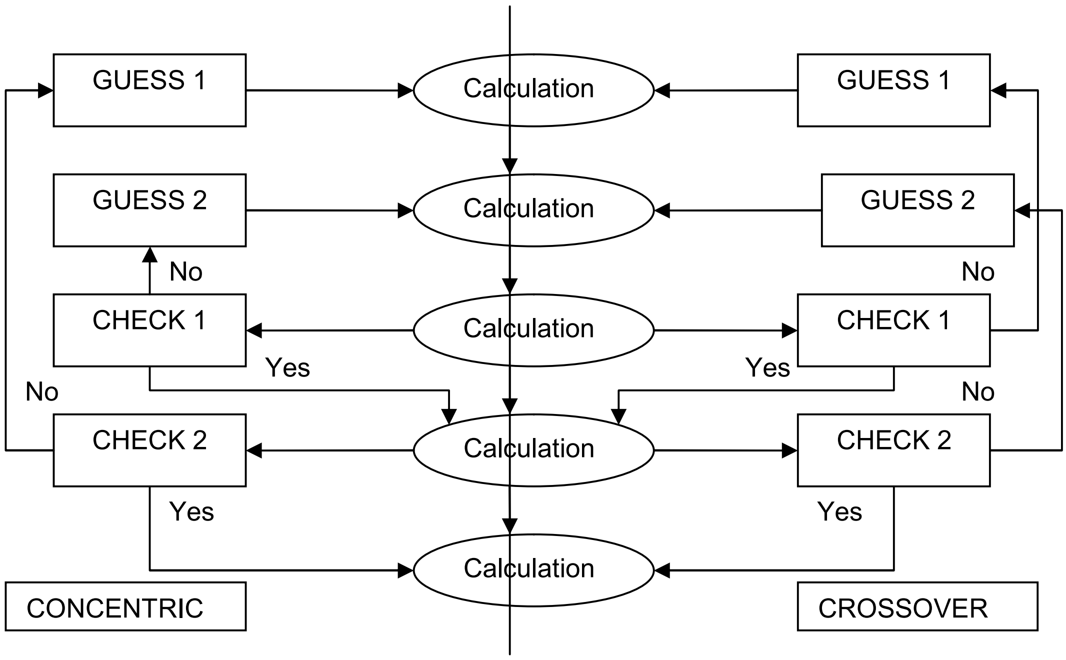


Figure 4.3 Iterative methods typical for off-design solution[37]

Multidimensional Newton-Raphson is an alternate technique of iteration commonly applied in GT performance research methodologies. The set of equations defining the errors are solved simultaneously and the errors in the matching constraints are updated all at same time. In the Newton-Raphson method the chosen error is represented by an error vector E, which is a function of the independent control variable C. Taylor expansion, can help to derive a relationship where E can be expressed in terms of C. The corresponding equation can be written as;

$$E = E_0 + \sum \left(\frac{\partial E}{\partial C_j} \right) \delta C_j \tag{4.4}$$

To make this error E equal to zero δC is obtained by multiplying the (eqn. 4.4) on both sides by the inverse of the partial derivative matrix. The new value of the control variable is thus given by eqn. 4.5

$$\delta C = -J^{-1} E_0 \tag{4.5}$$

Where J is a Jacobian matrix of the partial derivatives of E w. r. t. C_j

Start from an initial guess for E, viz. E_0. The corresponding control variable is C_0 is obtained and the Jacobian matrix is determined with respect to the error E. Eqn. 4.5 then calculates the necessary change in δC and the process is repeated such that the error E is lower than a tolerance value set before hand. The solution by MDNR represents the behaviour of the engine much more closely to its real path.

4.1.3. Nested Loop vs MDNR

Both the methods explained above have their advantages and disadvantages. The nested loop iteration requires more steps to converge and it can get heavy on computational resources and increase simulation time, while MDNR iteration is highly time efficient as convergence of the independent variable is reached by squaring the error at each step.

The MDNR method requires a good guess of the control variable to be able to provide quick results and be numerically stable. At the development stages of an engine program, working knowledge of the GT behaviour is limited. Only after decent analyses has been done on the performance, that MDNR can be a powerful method for engine simulation. The nested loop is numerically more stable as it provides insight of the convergence pattern. One can understand the guess-check error path of achieving equilibrium and if the simulation does fail due to non-convergence the error tolerance band can be reduced for the given variable.

When modelling engine dynamic behaviour, the MDNR technique is more suited to the need of intercomponent volume method, while both of the iterative methods provide quick results for the constant mass flow method in dynamic analysis. To simulate a complete gas turbine performance package, it is advisable to start with simple loops for iteration and then move towards complex integrating techniques and possibly combine the different methods for a real-time simulation.

4.2. Digital Simulation Numerical Treatment

The digital simulation model for off-design performance begins after the execution of the design point calculation. All the physical properties have been defined either by the input data file or from the thermodynamics of the gas turbine. To model the off-design performance of GTPS select an engine ‘handle’ define an operating range of the engine for which results are desired. The ‘handle’ is set to the requirement of power output. This requirement is not considered a direct input to the GTPS model, but is a function of fuel flow. The calculation of engine performance is carried out by using the subroutines mentioned in Chapter 3. Additional subroutines which are used here for OD and their description are mentioned below.

ODCARPET: Subroutine to set the engine handle and scope of OD procedure. The fuel flow is divided into smaller steps, which defines the operating range for which off-design analysis are desired and is also the boundary condition for the ‘handle’.

SSPT: Subroutine to calculate one off-design point. It carries out interpolation to match compressor and turbine work.

ODMFLW: For a given fuel flow and rotational speed of the shaft, performs matching of flow rate. The guesses and checks for iteration are performed in this routine.

At a fixed operative condition and main engine parameters, such as Mach number, ambient temperature, turbine inlet temperature, there exists a univocal relation between compressor pressure ratio and engine performance. From the knowledge of the pressure ratio it is possible to compute the thermal cycle of the engine and from this the total and specific performance, such as total power output, thermal efficiency or specific fuel consumption[38].

A common procedure for steady-state modelling of the two-shaft and single shaft gas turbine in the current project was carried out as follows (refer to Figure 3.2 for station numbering, any additional stations indicated shall be explained);

Step 1: Begin the process by defining the engine handle for the steady state model. The handle normally chosen should reflect the matching conditions of the engine. They can be rotational speed N_{CMP}, COT etc. The author has selected fuel flow $\dot{m}_{FL}$ as this was able to satisfy one of the requirements of the GT-EN interface for AMEPS.

Step 2: Call subroutine SSPT. At the first loop beginning with the speed N_{CMP} from the design point and the fuel flow from step 1, one pass is made through the off-design calculations.

Step 3: To match the engine cycle from the initials conditions, call subroutine ODMFLW. At the end of the calculations derive the work done by the power turbine, which is the useful work defined at the initial co-ordinates (N_{CMP}, $\dot{m}_{FL}$).

Step 4: Perform intake calculation by assuming that the ambient conditions are known and get $T_{200} = T_{100} = T_0$ (i.e. neglect any ram temperature rise in the intake) and $P_{200} = P_{100}\ x\Delta P$ which will give the entry conditions at compressor.

Step 5: Starting at the design point non-dimensional speed $\overline{N}_{CMP}$ initialise the subroutine MCOMP to perform compressor results. At this point the pressure ratio π_{CMP} is unknown and has to be assumed or guessed. The compressor maps have this in the form of beta (β_{CMP}) and a guess is made.

Generally a good guess would be the starting point at of the design value. For a loop besides the first one, the value from the latest point is another acceptable guess. Now the compressor is completely defined and the solution should proceed as outlined in section 3.7.2

Step 6: Calculate bleed flows extracted with the new mass flow rate. Determine the compressor work and signify the values of compressor outlet to the combustor inlet. At this point all the parameters are defined and no guesses have to made, but simple calculations are carried out in the subroutine CBNLW.

Step 7: From the value given for fuel flow to begin the off-design calculations, calculate the temperature at combustor outlet from eqn. 4.6

$$T_{CC.OUT} = T_{CC.IN} + \frac{\dot{m}_{FL} \cdot LHV}{\dot{m} \cdot C_P} \quad (4.6)$$

As the solution follows a path similar to standard gas path analysis in a GT, the FAR from can be calculated from eqn. 3.23 by assuming $FAR_{CC.IN} = 0$. After finding the ratio, update the temperature at the outlet of combustion to include the effects of total enthalpy and specific heat C_P by using the subroutine GASPROP.

Step 8: Guess a value for the expansion ratio of the compressor turbine. Start off at the value of the design point which is defined in terms of beta (β_{TR}). Using $\overline{N}_{TR} = N_{TR} / \sqrt{T_{20}}$ (as $T_{20} = T_{17} = T_{CC.OUT}$) and beta, from the turbine map find the non-dimensional mass flow $\overline{m}_{TR}$ and turbine efficiency. The turbine temperature drop is found from the subroutine TURB, which also gives the work provided by the turbine.

Step 9: Return bleed flows downstream of the compressor turbine. Execute the subroutine BLDRET.

Step 10: The calculation has reached the power turbine inlet. The known variables are the station vectors at the compressor turbine exit. A fourth guess is made at this point. The value of beta (β_{PT}) at design for the power turbine is selected. The following is known $\overline{N}_{PT} = N_{PT} / \sqrt{T_{30}}$ (for $T_{30} = T_{29} = T_{TR.OUT}$) and the point on the power turbine map is selected. Calculate the outlet conditions as per section 3.7.4

The procedure from step 4 to step 10 is one pass of calculations to determine useful work of the power turbine at a given operational condition. Along the steps there are some guesses made based on the logic that at the exhaust of the power turbine the pressure should be as close to the ambient conditions of air. In Figure 3.2 starting at the top and moving towards the exhaust it can be represented that:

$$P_0 = P_{590} \tag{4.7}$$

The same equation can be written as (neglecting the pressure losses due to bleeds and ducting);

$$\frac{P_0}{P_{590}} = \frac{P_0}{P_{200}} \times \frac{P_{200}}{P_{300}} \times \frac{P_{300}}{P_{410}} \times \frac{P_{410}}{P_{450}} \times \frac{P_{450}}{P_{590}} \tag{4.8}$$

In the above equation the pressure ratios; $P_0/P_{200} = \Delta P_{ITK}$; $P_{200}/P_{300} = f\,(\beta_{CMP})$; $P_{300}/P_{410} = \Delta P_{CC}$; $P_{410}/P_{450} = f\,(\beta_{TR})$ and $P_{450}/P_{590} = f\,(\beta_{PT})$, indicate that there are at least three conditions which are unknown and have to be satisfied. These unknowns are therefore guessed. Also with knowledge of the above pressures and the corresponding temperatures it is possible to derive majority of station vectors by using suitable formulae and component characteristics, so it is the best possible solution to guess these values. Also, from the above equalities it can be noticed that the final ratio is that of the power turbine. Thus there is just one right pressure gradient which will satisfy the equalities of eqn. 4.7 and 4.8 and the first check for the guess of β_{PT} is the pressure drop given by equation 4.7.

The aim of the OD routine was to find a suitable point on all the components which would satisfy the compatibility of flow throughout the gas path. In other words, the non-dimensional mass flow at a particular station in the engine is constant for a steady state operating point. This variable for the compressor and turbine is mapped on their respective characteristic maps. For a given pressure ratio and non-dimensional mass flow, there is the non-dimensional speed which provides the pressure ratio. If this speed is known then all the station variables can be computed and hence it is chosen as the starting point for the off-design matching.

At steady state, flow compatibility dictates that NDMF of compressor turbine at its exit should be equal to the NDMF entering the power turbine. That can be written using the turbine entry conditions as (4501 is an additional station between 447 and 450, added to realise the effects of bleeds assuming mass flow equality across the turbine i.e. $\dot{m} = \dot{m}_{4501} = \dot{m}_{410} + \dot{m}_{BL}$):

$$\left.\frac{\dot{m}\cdot\sqrt{T}}{P}\right|_{450} = \left.\frac{\dot{m}\cdot\sqrt{T}}{P}\right|_{4501} \tag{4.9}$$

$$\left.\frac{\dot{m}\cdot\sqrt{T}}{P}\right|_{4501} = \left.\frac{\dot{m}\cdot\sqrt{T}}{P}\right|_{410} \times\sqrt{\frac{T_{4501}}{T_{410}}}\times\frac{P_{410}}{P_{4501}} \tag{4.10}$$

This equation is another check for off-design matching, because the equality between the right and left terms indicate that $\overline{m}_{PT} = f\,(\beta_{TR})$

Similarly the NDMF at the exit of the combustor is equal to the NDMF at the inlet of the compressor turbine. By introducing another station 4101 between station 395 and 410 the mass flows are $\dot{m} = \dot{m}_{4101} = \dot{m}_{350} + \dot{m}_{BL}$ and eqns. 4.11 and 4.12 can be satisfied

$$\left.\frac{\dot{m}\cdot\sqrt{T}}{P}\right|_{410} = \left.\frac{\dot{m}\cdot\sqrt{T}}{P}\right|_{4101} \tag{4.11}$$

$$\left.\frac{\dot{m}\cdot\sqrt{T}}{P}\right|_{4101} = \left.\frac{\dot{m}\cdot\sqrt{T}}{P}\right|_{200} \times\sqrt{\frac{T_{4101}}{T_{200}}}\times\frac{P_{200}}{P_{300}}\times\frac{P_{300}}{P_{4101}} \tag{4.12}$$

The latter equation suggests that $\overline{m}_{TB} = f\,(\beta_{CMP})$. In eqn. 4.12 $P_{300}/P_{4101} = \Delta P_{CC}$ and the other variables are known either by calculation or by previous station. Therefore a successful check of eqn. 4.11 will be a match against the compressor pressure ratio value.

Step 11: Execute the first check given by eqn. 4.7. Proceed to next step if successful, however if the terms do not match go back to step 10 to perform iteration of β_{PT} against a level of tolerance till the value is lower than required.

Step 12: Check to see if eqn. 4.9 is satisfied. Proceed to next step if satisfactory within a tolerance limit. If the results are out of range then iterate between β_{TR} and error limit till convergence is reached. The iteration executes by taking the pointer to step 8 and calculations are performed again until this check point.

Step 13: Check for equality of eqn. 4.11 and like the previous steps perform iteration beginning from step 5 by changing the value of β_{CMP} till convergence is achieved.

Step 14: When all three conditions are satisfied, check if the compressor work is absolutely equal to the corresponding compressor turbine work within a limit of tolerance of between 10^{-5} to 10^{-6}. If the condition is not satisfied select a new rotational speed for the gas generator and begin calculations again from step 3. When compressor work is the same as turbine work, it is said to have reached steady state and the simulation of the operating point terminates.

Step 15: Based upon a new value of fuel flow repeat the whole process right from the beginning with the selection of the gas generator speed to get the working line of the gas turbine.

For the single-shaft gas turbine the matching occurs differently as described before. The number of checks and guesses are typically lower. The matching set of errors and independent variables can be found in any literature explaining the off design performance of a gas turbine. In the current study a condition of the reduction gearbox between the compressor turbine and the power turbine was set at design point. This eliminates the need to represent the physical model of a transmission drive to couple the gas turbine with the electrical network model. Routines for matching the components remain absolutely similar to the previous description and so does the hierarchy. For the single shaft GT matching replace the steps 2, 3 and 14 of the two-shaft engine and follow the next steps accordingly, repeating the remaining steps as mentioned before:

Step 2: Start with N_{PT} and the selected handle to begin the matching procedure.

Step 3: From eqn. 3.37; find the gas generator spool speed which is $N_{TR} = N_{PT} \times GBXTR$ at the beginning of subroutine ODMFLW and with the handle, continue from step 4 through to step13.

Step 14: Check if compressor and compressor turbine work is equal within a certain tolerance limit. At this point, in case of inequality, select a new spool speed N_{PT} and repeat the process from step 2 until convergence is reached. It should be noted here that the power turbine is connected to an external load which has a constant speed, and by changing it the program essentially changes the pressure ratio along the same speed line on the characteristic map as the turbine is operating choked.

4.3. Off-design Analysis Considerations: Ambient Conditions, VSV

The gas turbine engine is a very sensitive machine to handle. A minor change in the gas flow path can bring about significant and sometimes disastrous results. It is thus crucial to understand the behavioural tendencies of the parameters which can lead to a substantial difference of output. The compressor characteristic map (Figure 4.4); plotted to show the variation of the overall pressure ratio and non-dimensional mass flow for different non-dimensional speeds, which is a factor of the compressor inlet temperature, is a graphical representation of the working process for this module. This map together with the inlet temperature defines the working process within a compressor. There are points on this map that satisfies the continuity of mass flow rate for the engine over its operational envelop. By joining these points, a steady state or the 'equilibrium line' of the gas generator is defined. For a case when the GT would be either accelerated or decelerated there is change in mass flow or if there are any secondary or tertiary effects like bleeds and variable geometry considered the operating points change position and there is a shift in the line. The map is therefore used to understand and explain the engine behaviour and the equilibrium line indicates the part-load performance of the whole system w. r. t. the compressor behaviour.

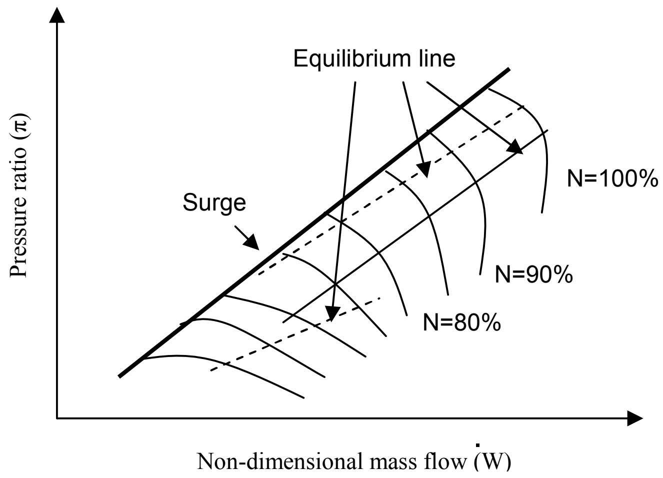


Figure 4.4 Multi-stage compressor map indicating equilibrium line and OD probability

The gas turbine engine is manufactured upon the basis of certain requirements which define the engine cycle. However for the purposes of research and predicting the behaviour it is economically exepnsive to manufacture a prototype and then conduct experiments to gather data. Any air breathing engine is provided with its main source of work from the environment, which is as the name suggests air. The gas turbine engine's working medium is also this air, and depending upon the place of its location and general atmospheric conditions it's performance will change. This is assured as basic thermodynamics suggest that work is a function of temperature and the purpose of any power and propulsion system is to produce useful work. Therefore it is necessary to analyse the off-design characteristics with changing conditions of ambient temperature to examine the feasibility of the same GT engine as a power system in different regions of the globe and under different operating conditions from a standard deviation. Also, in case of the first compressor the atmospheric temperature will be used to define the point on its map that is the ambient temperature minus any losses in ram temperature. Thus the ambient condition differences are a good starting point for an off-design performance and steady state modelling

An axial compressor is divided into rotors and stators to achieve a pressure rise across it. One set of these components is one stage of the compressor and the pressure gradient can be in the region of 1.15 – 1.60. Without going into details about compressor design as of how to achieve a high overall pressure ratio, a compressor with more than one stage will do the job on hand. A single stage of the component has its own unique map and these individual stage maps are stacked together to represent a multi-stage compressor by one single map. When the geometry of the blades is fixed the map is fixed, but by changing its geometry essentially the compressor has a new map. By employing variable stator vanes, an effective control schedule for transient modelling can be chosen depending on the referred speed setting of the spool. Thus from the point of view of engine control and off-design analysis of the compressor map shift is necessary for different stator angles.

Various analyses have been carried out in this work to precisely understand the off-design impact. The process is repeated between two types of engine to make a comparative study of the working process. In the first study based upon different ambient conditions, total amount of useful work is derived and compared for two engine types using the performance code. The running line is plotted on the compressor map by selecting different engine settings. This process is then carried out with varying ambient conditions and different stator angles. A process like this is repeated for several set of conditions and the explanation to the behaviour is suggested in the chapter presenting the results.

5. GAS TURBINE TRANSIENT MODELLING

A transient state is usually what a military machine is in and gas turbines for commercial and industrial applications vary between a steady state and dynamic state. Much of the current research work in the area of performance prediction is focused on the analysis of transient operation of gas turbine based engines. The response rate to a change in power demand is extremely important in fast moving machines. In AMEPS, the gas turbine provides thrust power through an electrical generator that renders power for the drive and additional system loads. The transient modelling is all the more necessary in such a scenario where the power demand can fluctuate with load sharing. The implementation of a transient model in AMEPS makes it possible:

- To predict transients due to change in thrust demand and electrical power demand.
- To minimise the risks at development and subsequent design stages for the power system.
- To design an efficient and optimized control system meeting the requirements for mechanical and electrical stabilities.
- To suggest new standard of operational limits for completely validated system and sub-system models.

5.1. Literature Review on Transient Modelling

Gas turbines were modelled as the commercial success of the engine grew and its steady state performance prediction was the item on the agenda. With progressing research capabilities using computing power and logically speaking transient modelling was the next step. At the early stages of development, the dynamic models related to individual components. Typically one dimensional mathematical correlations solved with available resources were created and the various modules according to engine layout were assembled to form an overall representation of the gas turbine.

Kuhlberg et al[39] describe one such model for turbine compressor dynamic simulation. The paper presents the results on the study of a multi-stage compressor dynamics to one-dimensional external disturbances. Of the different techniques applied to predict the behaviour of a turbine compressor their conclusion was that a mathematical model utilizing the basic conservation of momentum and continuity is a fair representation of the compression system. They further went on to adapt the model to a fan-compressor simulation and provide some very useful output on the effects of the geometry of individual stages of a compressor for various disturbances generated by thrust fluctuations. DYNTECC by Hale and Davis[40] was a computer code "to analyze post-stall behaviour in a turbine engine compression system as well as predict the onset of system instability". The code was set to use "a finite difference numerical technique to solve the mass, momentum and energy equations with turbomachinery source terms". The potential of the code also stretched to multi-spool compression systems just like the work of the previous authors; plus, it was capable of foretelling the stability margins for external inlet temperature distortion and pressure disturbances. With modulation of DYNTECC, the code can be extended to simulate a 'what if' scenario and used for further analyses of hardware or control system actions on the compressor.

Fawke and Saravanamuttoo[41] in the author's opinion can be regarded as the pioneers of the current techniques applied in the field of gas turbine performance simulation and prediction. Their theory of using component models to represent an engine is the most successful methods of behavioural analysis. They first tackled the problem of dynamics prognosis based upon two methods; viz. the iterative method and the method of intercomponent volumes. The iterative method assumes "that at every point in time during operation of a gas turbine engine, flow compatibility is satisfied, even if engine is operating transiently". The second method suggested assumed "that during transient operation, flow mismatch occurs and that this flow mismatch may be used to calculated the rates of change of pressure at the various stations in the engine". In this paper a comparison is made between the two methods and the glitches that may occur while digitally simulating a gas turbine based engine. Although, the latter was mostly on part of the technological limits at the time and can be easily overcome. Their results with the use of intercomponent volume was in close conjunction with experimentally provided data

The authors extended their research on to experimental verification of a twin spool turbojet engine[42] and analysis of more complex configurations of gas turbine engines[43]. For verifying the model, engine tests were set up and a series of transient manoeuvres with fixed and variable nozzle schedules were performed. The main achievement can be considered to be the digital control system developed to perform the calculated transients to conduct a wide range of parametric studies in a short period of time. The important feature of the first paper for the author was their idea of control system to incorporate a fuel flow schedule and a variable geometry configuration, which was used for AMEPS. Further they carried on their proposed methodology to understand the transients of a twin-spool turbofan with mixed exhausts. Previously the simulations did not account for any heat transfer effects to be considered. The response of a gas turbine to an increase in fuel flow i.e. an acceleration is different for a cold start, from idle to full power and from max power to idle and back to max power (this phenomenon is called as a 'hot reslam'). Because the metallic parts of an engine take some time to absorb the heat energy from the gas, the response time to an instant step up in power demand changes. This effect was studied by Fawke and Saravanamuttoo[43] in their simulation of the turbofan engine. The important results of their investigation show that the effects of heat transfer are significant while predicting the response time during transients, however, the working line on the compressor characteristics did not follow a path significantly different from the path of normal transients.

Pilidis in his Doctorate thesis[30] discusses the work carried on gas turbine modelling since its early days and explains his choice of methodology for a digital simulation of GT performance. His approach was based on the method of constant mass flow with which eleven configurations of aero-engines were studied. The models incorporated heat transfer phenomena and an additional effect of seal clearances on the transient time showed that “the compressor and turbine clearance movements result in small beneficial effects during acceleration”.

By the eighties simulation methods used to predict gas turbine behaviour were already accurately representative of the actual system. As the digital computer got better and faster, these codes were used to model the control systems of the engine. Torella[44] proposes a package of numerical codes that could be used as tool for study by students, engineers and even aircraft pilots. The code incorporated different control laws for several engine types and the user could understand the physical concept of engine transients though the use of graphical images. To generate the visuals, pre-programmed engine outputs with selective control were packed into a vast database and then depending upon the choice of user selected control factors and factors chosen by the code, if satisfactory similarity is obtained by the codes it calculate the performance of the engine under new laws by interpolation. In case of a major difference between the user and database option, the code is able to predict the output by the usual means of digital simulation. Hung[24] in 1991 developed an electronic/hydraulic governor control for a gas turbine-generator by monitoring the gas generator and power turbine speed and the exhaust gas temperature. The signals received from the engine were fed into a governor control schedule limiting their values within the range of operation below surge margins, over-temperature zone and above extinction zone. To comprehend the dynamics of such a system, the governor was “developed by deriving transfer functions from the corresponding electric circuits and the measured hydraulic response characteristics”; while the gas turbine was “represented by a linear model which was devised assuming that the engine can be represented as a collection of multi-variable functions, which can be linearised by writing them in total differential form”.

Gas turbine based propulsion systems had reached high landmarks by the nineties and the need for simulation of real time transients so as to develop a Full Authority Digital Electronic Control (FADEC) engine systems control was becoming all the more necessary. The time to run one transient on an average computer was nowhere near to the real time transient manoeuvre. MacInnis[45] at the Allison Engine Company developed a different matching method and compared its results to the production deck and horsepower extraction models that were used in the company. By using “the original component performance data and converting that into block data to be input only once and then applying iterative constraint method to determine flow scalers, the modified simple cycle-matching transient simulation was able to achieve a simulation time-to-transient time ratio of 1.8:1”. These results can be considered realistic if reached in the current project as the development process of AMEPS is in its infant stages.

The trend of compressor behaviour is slightly different at very low power settings than at speeds from idle to maximum rated. During the start-up an engine can be highly unstable as the stage loadings of the compressor may be close to surge and stall. To overcome these effects and other effects like hot start, whereby a failure in power change from negative to positive will result in very high temperatures of the turbine, the simulation of transients at start-up generally requires the scheduling of some kind of variable geometry in addition to typical fuel control in the engine to safely attain idle speed. Kim, Song et al[46] present a “simulation program for transient analysis of the startup procedure of heavy duty gas turbine for power generation”. Their approach is constant with the previous work in the idea to represent a GT. Their approach groups “the compressor stages in three categories (front, middle, rear) whereby the characteristics of each part are stacked by a modified stage-stacking method to estimate the operation of the compressor from zero to full speed”. Their results showed that “the modulation of variable inlet guide vanes is very important for the stable operation of the compressor and how a wrong schedule estimate can severely affect the startup characteristics estimate”.

In summary, the literature survey done helps to determine the following aspects of gas turbine transient simulation:

1. The method of intercomponent volume is the popular and accurate choice of transient performance prediction. The complex nature of analysis by this method can be heavy on computing power and unnecessary at the early stages of development.
2. The method of constant mass flow can also accurately represent the engine dynamics for large shaft transients. The method follows a procedure similar to off-design methodology.
3. For either of the methods chosen it is important to develop a fuel control system based upon engine configuration. For a simple gas turbine this is generally dependant on the shaft speed. However, with the addition of secondary control variables the transients can be achieved faster.
4. Heat transfer and seal clearances significantly affect the operational behaviour of the gas turbine therefore it is necessary to take into account these factors for any transient studies.
5. To perform parametric studies or for further modifications, the simulation code should be flexible in accommodating any additions without major changes. For this it is best to stick to the component modelling approach of steady state.

5.2. Method of Constant Mass Flow

The engine steady state point is defined by work balance between a compressor and its corresponding turbine. This work balance is achieved when the gas generator spool speed is fixed by the steady state operating point. Furthermore, the compressor work in general is a function of the pressure ratio, temperature, mass flow, rotational speed etc. (neglecting any external effects designed for mechanical stability) and the turbine work is a function of the same variables plus the value of fuel flow. Now a transient state arises when the work balance is hindered due to an externally invoked disturbance like change in fuel flow or some internal effect like change in inlet temperature due to ambient variation. The study of transient modelling involves the prediction of the running line from a certain point A to point B on the characteristic map (Figure 5.1) of the components.

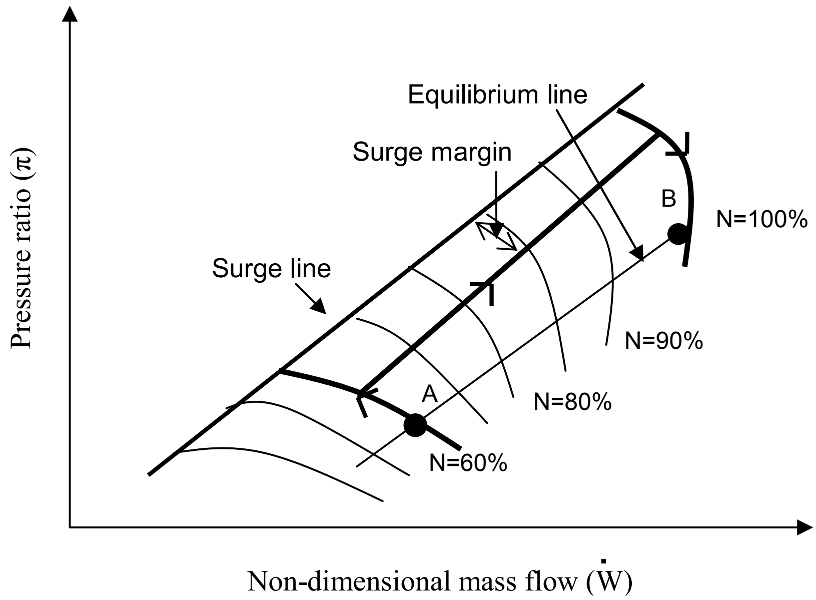


Figure 5.1 Possible trajectory of acceleration

Also, transient performance simulates the response of engine parameters in time. As far as steady-state operation, which are points A and B on the above map, is concerned, the rates of change for all engine parameters are nil. The, power imbalance on a shaft causes an acceleration shift in point A in the direction of the arrow. For the cases of the gas-generator and the power turbine shaft of a two-shaft engine, the following relationships can be expressed, respectively:

$$I_{PS} \cdot \omega_{PS} \cdot \frac{d\omega_{PS}}{dt} = \eta_{mech} \cdot P_{PT} - P_{EG} - P_{AUX} \tag{5.1}$$

$$I_{GG} \cdot \omega_{GG} \cdot \frac{d\omega_{GG}}{dt} = \eta_{mech} \cdot P_{TR} - P_{CMP} - P_{AUX} \tag{5.2}$$

According to the right hand side of eqn. 5.1, the useful power is expressed as the difference between the power delivered by the power turbine minus any parasitic losses and power, which is used to drive any auxiliary systems. For the gas generator, from eqn. 5.2 the rate of change of angular velocity depends upon the power excess between the compressor and turbine. Assuming adiabatic processes in the turbomachinery components, their power relationship which is similar to work done can be expressed as:

$$P_{CMP/TB} = W_{CMP/TB} = \dot{m} \cdot \Delta H_{CMP/TB} \tag{5.3}$$

This leads to the objective of transient simulation, which is to evaluate the angular acceleration the engine will experience, when work done by the turbine will be in excess or in deficit of the work required by the compressor (assuming auxiliary work is constant). To evaluate this acceleration means to find a rotational speed at $t+dt$. Of the several techniques to find a new speed Euler's method is of the simplest from, where it is assumed that the acceleration will remain unchanged during the time interval for which it has been calculated (eqn. 5.4).

$$\omega_{t+dt} = \omega_t + \left(\frac{d\omega}{dt}\right)_t \cdot \Delta t \tag{5.4}$$

5.3. Transient Calculation Methodology

The first transient simulation was carried out for the two-shaft gas turbine. To solve the aforementioned eqn. 5.4, so as to compute the time required to achieve steady-state the principle of conservation of mass flow is assumed. This means that the calculation takes form of an off-design modelling. At time $t=0$, with knowledge of inlet conditions the state vectors can be derived at the exit of inlet. By defining the rotational speed and fuel flow, set by demand of power, the point on compressor map that satisfies the continuity of mass flow is fixed. To find this point an arbitrary value for compressor pressure ratio is chosen and the non-dimensional mass flow and isentropic efficiency is gathered from the map. The temperature rise at the exit of compressor is derived form the isentropic efficiency, pressure is obtained from the chosen pressure ratio and mass flow is deduced from the non-dimensional value. This fixes the compressor exit properties. The combustor outlet temperature is determined by the fuel flow into the combustion chamber, the mass flow from compressor and the lower calorific value of the fuel. Pressure at combustor exit is taken as a percentage of its efficiency times the pressure discharged by the compressor. Continuing the calculation in the same fashion like the off-design steady-state modelling, choose another arbitrary turbine exit pressure. From the calculated turbine entry non-dimensional mass flow, find efficiency and establish the exit conditions of compressor turbine. The final guess of power turbine expansion ratio will result in an output by which all the important variables of the engine will be defined according to the individual characteristics or plain mathematics. In general, continuity of mass flow will not be satisfied from the first guesses. Hence iteration needs to be performed to find the right operating condition whereby the non-dimensional mass flow at certain stations can be replaced by the equation:

$$\left(\frac{\dot{m}\sqrt{T_t}}{p_t}\cdot\sqrt{\frac{R}{\gamma}}\right)_i=\left(\frac{\dot{m}\sqrt{T_t}}{p_t}\cdot\sqrt{\frac{R}{\gamma}}\right)_j\cdot\frac{\dot{m}_i}{\dot{m}_j}\cdot\sqrt{\frac{T_{t,i}}{T_{t,j}}}\cdot\frac{p_{t,j}}{p_{t,i}}\cdot\sqrt{\frac{R_i}{R_j}\cdot\frac{\gamma_j}{\gamma_i}} \tag{5.5}$$

In the GTPS code these steps to attain compatibility are carried out in the subroutine ODMFLW and are absolutely similar to the off-design analysis. However, the code was updated to compute the transients and fuel schedules and VSV schedules were added to perform controlled transients.

In the two-shaft gas turbine, after one pass was made through the subroutine, the transient subroutine was initiated. Selection of an appropriate time step is necessary to accurately represent the transient. A large step converges quickly failing to indicate the gas dynamics, while a very small step takes very long to process without significant changes actually occurring in the engine. The final step size selected was based upon intelligent guess and trial and error.

The transient is started by a change in the fuel flow. This value along with the power turbine rotational speed is fed into the code. A mismatch is created between the power of compressor and power of turbine that can be expressed as follows:

$$\Delta P = P_{TR} - P_{CMP} \tag{5.6}$$

This results in an accelerating torque where angular velocity at time t is given by eqn. 5.7 and net torque is eqn. 5.8

$$\omega_t = \frac{2\pi \cdot N_t}{60} \tag{5.7}$$

$$\tau = \frac{\Delta P}{I \cdot \omega} \tag{5.8}$$

The shaft reacts to the net torque and angular acceleration occurs that can be represented as;

$$\Delta\omega = \tau \cdot \Delta t \tag{5.9}$$

The new shaft speed can then be calculated as;

$$N_{t+dt} = N_{t+dt} + \Delta\omega \cdot \frac{60}{2\pi} \tag{5.10}$$

Or from eqns. 5.7 – 5.10 the rotational speed at time $t+dt$ is;

$$N_{t+dt} = N_t + \frac{\Delta P}{I \cdot N_t} \cdot \Delta t \cdot \left(\frac{30}{\pi}\right)^2 \tag{5.11}$$

The final equation 5.11 is the one which has been implemented in the transient code to calculate power at a certain point of time for the two-shaft gas turbine. This routine was created to set up the GT-EN interface by which the fuel demand and rotational speed of the power turbine is input through Simulink and output is supplied back so as to analyze the IFEP system behaviour. The speed N is the gas generator rotational speed. There is a control on the fuel set within the two-shaft engine to manage the stability limits on fuel input.

The single shaft simulation considered the gearbox ratio to calculate the rotational speed of the power turbine. An attempt was made to simulate the transients of the engine with this ratio. This involved carrying out the performance prediction as per the two-shaft engine and then modifying the shaft speed of power turbine from the eqn. 5.12

$$N_2 = \frac{N_1}{GBXTR} \tag{5.12}$$

Though results were achieved they are not very convincing for standalone transients using only the code. In the case of AMEPS, the simulation did not cause any hindrances, as the interface required the solution of transient due to power imbalance within the environment of Simulink. This was easily facilitated with a simple steady state matching, to evaluate the output power which is fed back into the IFEP system tool. A secondary control was not needed in this type of engine and only maximum and minimum fuel limits were set for it. To terminate the transient internally for both engine types, a condition is set such that when the torque on the shafts reach zero, the code exits the transient mode.

Other additional subroutines required for the GT-EN interface was that of transient initialisation and design point mass flow correction. The engine was fixed at a design point value of the overall pressure ratio and the combustor outlet temperature. To these values a mass flow correction was made to get the required output of useful work as close as possible to the desired work at design point. This was performed by iterating the value of mass flow at design point with the difference in work to satisfy the tolerance of error difference. The iteration was carried out with the subroutine DPCALC as this was a design point calculation. Also, many of the IFEP system simulations needed to be carried out from a point which was not the maximum rated power. To reduce the time necessary for a simulation, and to begin the transients at the right point, a routine that corrected the fuel to match the value at the required power setting was input. This was a simple routine which iterated between fuel flow and useful work output till the error between the value of required work and work predicted was below an error tolerance of 10^{-5}. Both the subroutines formed a part of the initial conditions of transient simulation in the interface.

5.4. Control Schemes of the Two-Shaft Gas Turbine

The gas turbine engine control system limits the working conditions of the compressor by surge control, over-fuelling limits and temperature boundaries, and allows speedy steady state convergence ensuring health and safety of the machine. Furthermore the two shaft performance model was to be implemented in the GT-EN interface to research its behaviour from a sub system point of view. In IFEP systems a gas turbine engine is not controlled simply by its own laws, but the electrical network system decides for its operation. Electrical networks are typically simulated and governed by a specific type of controller, known as the Proportional-integral-derivative (PID) controller. The fundamental theory of a PID governor is that given a variable with a required value, the system will perform a process starting at a previous value. A PID controller then returns a proportional, integral and derivative value of the error between the initial and processed value and feeds it back to the process. This procedure is repeated until the desired value is achieved. This is a closed loop type of control and usually the derivative is ignored as it creates noise in the system. Thus only a PI governor controller was implemented in the electrical network and was based upon the frequency difference of the power turbine and generator. The PI governor does not guarantee an optimal control or stability. Furthermore as the turbines in the two-shaft engine configuration are not coupled together by physical means, which means that the frequency of the compressor turbine is independent of the power turbine frequency and the PI controller cannot actually stabilise the gas generator. So an internal control schedule based upon the control laws of the gas generator needed to be implemented. The basic requirements of a control scheme can be listed as follows:

- The top priority of a control scheme should be the time limit desired by the user to complete a transient. For AMEPS the time limit was not set thereby allowing the control scheme to be flexible to the choice of the consortium.
- A control algorithm should ensure stability limits of the turbomachinery elements. The turbine entry temperature (or the COT if there is cooling air mixing involved) is the first boundary condition for the control schedule. As the first element to deviate from its operating point is this temperature because of the change in fuel input.

- Compressor surge margin or some limiting conditions for stable compressor operation should be another factor involved in engine control. This can be realised by numerous techniques depending upon engine geometry.
- Finally the control algorithms should be optimized for it to be reproduced as an actual system capable of operation.

5.4.1. Scheme 1: Maximum and Minimum Fuel Level

When controlling transients, the first limit can be set on the maximum and minimum fuel input. Fuel from steady state part load performance should be noted and then based on the limit of COT and/or surge margin fuel schedules can be decided. Its value can be calculated from the polynomial which can be plotted by referred rate of change of speed (NDOT) versus referred speed. By using parameter NDOT for control the response times for acceleration and deceleration do not depend upon secondary laws of control. Its use has some disadvantages however, which are in the fact that surge margins are worse when there are bleed flow extractions from the compressor, any auxiliary work provided, change in ambient conditions or deterioration of engine has occurred.

For marine propulsion systems the polynomial can also be written for referred fuel flow versus referred speed. For a control strategy like this generally to reach the path of the polynomial a step up of fuel is provided after which the equation kicks in. The rate of fuel for deceleration is similarly designed where the valves are open for a short period of time for a step down until the level reaches the scheduled law. These techniques can be used to achieve a certain response time when there is a set of standard response times in case of emergencies.

For the current project there were three polynomials, which were generated from an excel spreadsheet were used for a fuel schedule (Figure 5.2), which are given by the following equations; to calculate maximum fuel input capacity:

For $(N_{GG} / \surd T_{AMB} \leq 503)$

$$\dot{m}_{FL} = (\dot{m}_{FL} + 1) \cdot 0.04877 \cdot \sqrt{T_{AMB}}$$

$$\dot{m}_{FL} = 7984.6x^6 - 440.11x^5 - 369.68x^4 + 23.427x^3 + 10.725x^2 + 3.0699x - 0.0487 \qquad (5.13)$$

Where, $x = N_{GG} / \sqrt{T_{AMB}} / 441.504 - 1$

For $(N_{GG} / \surd T_{AMB} > 503)$

$$\dot{m}_{FL} = (\dot{m}_{FL} + 1) \cdot 0.095 \cdot \sqrt{T_{AMB}}$$

$$\dot{m}_{FL} = 3076.8x^5 + 64.627x^4 - 56.607x^3 - 2.7699x^2 + 2.132x + 0.0085 \qquad (5.14)$$

Where, $x = N_{GG} / \sqrt{T_{AMB}} / 558.1916 - 1$

To calculate minimum fuel input capacity:

$$\dot{m}_{FL} = (\dot{m}_{FL} + 1) \cdot 0.0277 \cdot \sqrt{T_{AMB}}$$

$$\dot{m}_{FL} = 1487.7x^6 - 89.228x^5 - 224.68x^4 + 52.31x^3 + 7.853x^2 + 1.9678x - 0.0374 \qquad (5.15)$$

Where, $x = N_{GG} / \sqrt{T_{AMB}} / 551.5221 - 1$

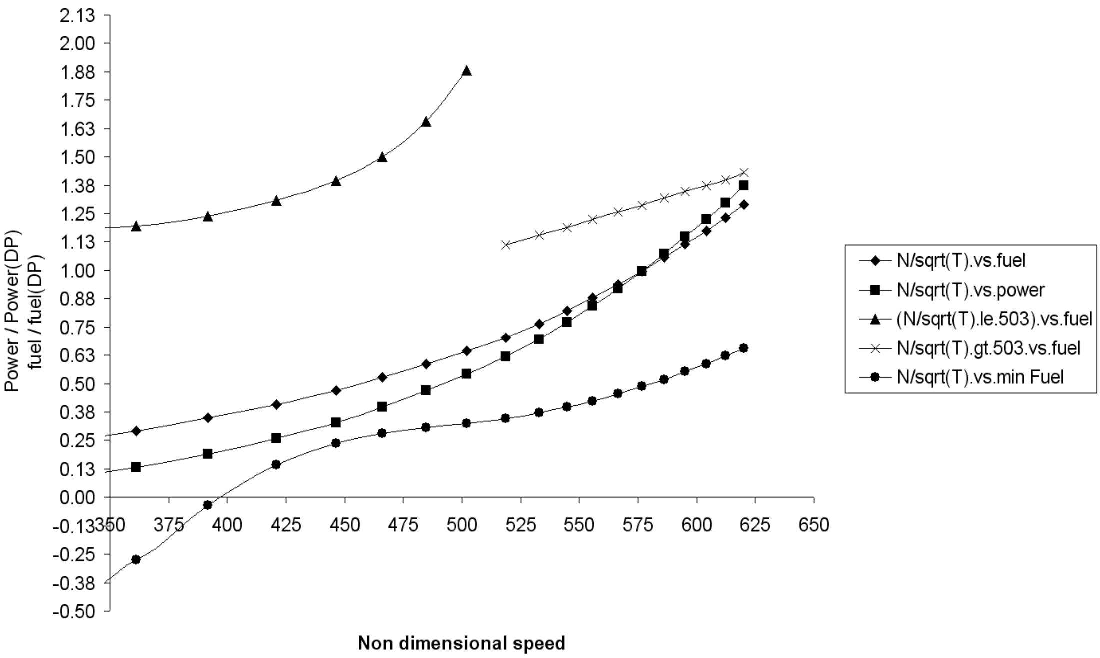


Figure 5.2 Fuel scheduling: Maximum and minimum level of fuel input

5.4.2. Scheme 2: Variable Stator Vane Angle Schedule

A gas turbine with variable geometry is highly desirable because of its use in improving transient control of the turbomachinery. This system model makes use of the stator angle schedule of the compressor to improve the surge margin of the running line during transient operation. Acceleration of an engine causes a fuel spike at the very beginning of time. During steady operation, in the compressor the first few rows of blades operate very close to surge, while the last rows are far from it if the engine is below idling. At high speeds the situation changes and a vice-versa phenomenon is noticed. If a ‘combined’ map of the possible stator angles is known, the effect observed on the map is a shift in the surge margin with respect to similar non-dimensional speed lines. The speed lines become tighter and closer to each other when the blades are closed which either push or bring nearer to the running line the surge line. There would be no change in the working line of the engine unless an effective control is developed.

Since the geometry of the engine has an effect on the part-load performance there were three laws developed based upon the component efficiencies. A variable geometry control is usually scheduled for referred speed values keeping the stacked map indigenous. The control design set (Figure 5.3) is between stator angle and gas generator spool speed. A condition of using a logical variable called ‘LVSVSCH’ was introduced here for law selection purposes. The second law is between stator angle and fuel flow and variable LSTEADY is the operator. Finally the assumption of LFFHND provides two options in the whole performance code. A value of ‘true’ signals that angle to be found w. r. t. the fuel flow and ‘false’ allows handling of COT. The solution at each point to find VSV from the three factors is given by eqns. 5.16 – 5.18 with a corresponding Figure 5.3 showing their trends. Five major combinations between the three logical variables are possible. They have been applied in the next section to find the behaviour of schedules on response time.

When all three logical variables are true, For ($\dot{m}_{FL} \leq 1.3895$)

$$x = \dot{m}_{FL} / \sqrt{t_{AMB}} / 0.00479 - 1$$

$$\alpha = 0.3552x^6 - 1.9416x^5 + 0.257x^4 - 0.0681x^3 + 0.0673x^2 - 1.0895x - 0.0163 \quad (5.16)$$

$$\alpha = (\alpha + 1) \cdot 10.3202$$

When the condition LFFHND = 'FALSE' and For ($T_{TR.IN} \leq 1400$)

$$x = T_{TR.IN} / \sqrt{t_{AMB}} / 4.1232 - 1$$

$$\alpha = 6374.6x^6 - 3141.4x^5 - 136.54x^4 + 49.095x^3 - 1.8622x^2 - 4.1439x + 0.0169 \quad (5.17)$$

$$\alpha = (\alpha + 1) \cdot 10.3202$$

If LSTEADY = 'FALSE', For ($N_{GG} < 9800$)

$$x = \dot{m}_{FL} / \sqrt{t_{AMB}} / 0.00479 - 1$$

$$\alpha = -9257x^6 - 3150.4x^5 - 112.31x^4 + 10.055x^3 - 5.27x^2 + 4.4093x + 0.0752 \quad (5.18)$$

$$\alpha = (\alpha + 1) \cdot 10.3202$$

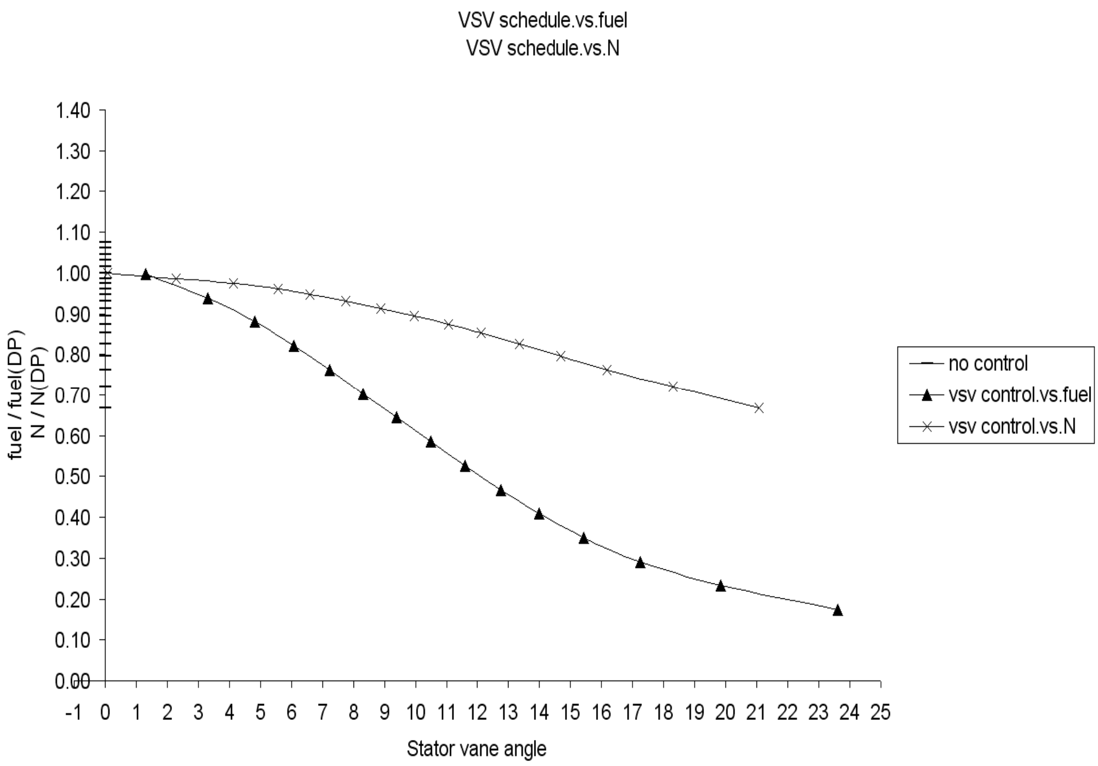


Figure 5.3 Variable geometry compressor scheduling

5.5. Transient Response Time Analyses

The idea of dynamically modelling a physical system is to check how it behaves over time. After analyses of such systems, recommendations can be usually given regarding its smooth and safe operation and the best possible ways to achieve the shortest time for simulation. In this section, based upon the noticed behaviour of the transient simulation code, the author suggests some factors which affect the simulation process and by which the response rate of the modelled gas turbines can be improved.

Considering the exceptional rotational speeds of the shafts, the response rate of any gas turbine engine, is the crucial result to be derived from dynamic prediction while monitoring the components behaviour for stability limits, to set safety standards of operation. "The response of a gas turbine is controlled by the rate at which excess fuel for acceleration of the rotor system can be introduced; this in turn is limited by the phenomena of compressor surge and temperature limitations in the turbine"[47]. The engine should be able to achieve a required time set by regulating authorities. The response time can be either limited as in the case of turbojet engines or can be monitored to understand the systems behaviour as is the case of this project.

Numerous factors can be attributed in influencing the time taken by the GT to either accelerate or decelerate. If there are no controls involved, then this solely depends on the inertia of the shaft. The smaller the inertia, the quicker will the engine respond to acceleration and vice-versa for deceleration. However, to run such an engine would mean taking incredible risks on operational costs and life cycle of the gas turbine. Though it is desirable to have high turbine entry temperatures, there is a certain limit placed on it in terms of manufacturing and technological levels. To exceed this limit would also mean a decrease in total engine life and can result in a catastrophe if the engine is producing high amounts of power.

Another factor that governs the response time is the rate of fuel flow even though this is not a direct function of time during transients. By supplying the combustion chamber high quantities of fuel, the turbine inlet temperature can be reached quicker and thus acceleration can occur quicker. Besides the criteria for turbine mentioned in the paragraph above, the rate of fuel flow is also limited by combustor design. All combustors have a region of stability where the fuel can be ignited and the flame maintained (Figure 5.4). By increasing the fuel flow without control, the working process of the combustor comes out of this region. If the fuel input is more than necessary then the point occurs in the rich limit and the flame cannot be maintained that will lead to a *'blow out'*.

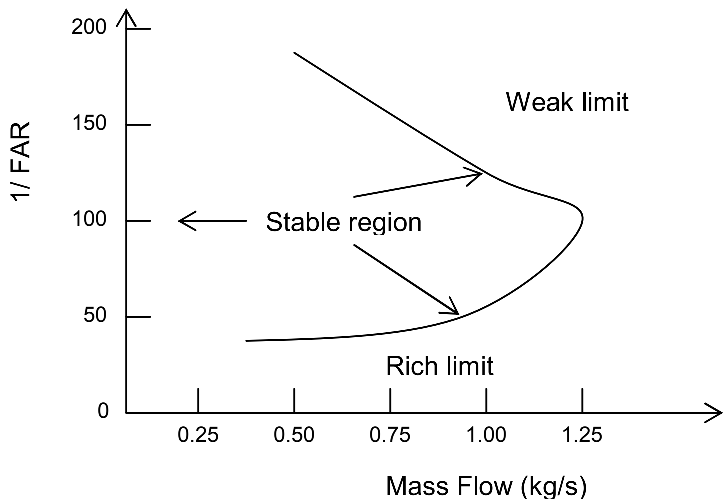


Figure 5.4 Combustion chamber stability loop[10]

Based upon the control techniques applied in the two-shaft code, Table 5-1 and Table 5-2 show the effects of variable stator vanes and fuel schedules on the response time. To compare the effects of stator vane angles and fuel schedules the engine was accelerated from three different settings to maximum power and the corresponding times were noted. The starting point was first obtained by steady state matching of the fuel flow and shaft speeds. Arbitrary values of 40%, 60% and 80% of design point fuel flows are taken. As an indicator, the computational time is provided in the last column of the tables. Although the two times are not identical, there is a pretty close difference, which indicates that the method of constant mass flow can be used in real-time simulation. Five combinations between three laws will satisfy the analyses to evaluate the important factors that affect the response time and also facilitates to understand the GT behaviour. The condition of *'true'* implies that the control law is used and *'false'* means control is ignored. The fourth scenario with 'not applicable' results is because when fuel flow initialises transients, 'handle=false' signifies the control to find the fuel in combustor and the code becomes unstable.

It was noticed that the fastest response time achieved was when the compressor stator vanes angles were modulated based upon the control schedule to prevent surge. The next quickest response time attained was by applying a schedule combination of variable geometry together with fuel flow schedule. By completely ignoring the schedules, it took the dynamic code the maximum amount of time to reach equilibrium.

Table 5-1 Acceleration times for two-shaft gas turbine

Acceleration					
FUEL (% of DP)	VSV	STEADY	HANDLE	TRANSIENT TIME (seconds)	CPU TIME (seconds)
40	FALSE	FALSE	FALSE	45.683	46.703
60	FALSE	FALSE	FALSE	43.266	34.282
80	FALSE	FALSE	FALSE	38.88	24.172
40	TRUE	FALSE	FALSE	31.654	39.094
60	TRUE	FALSE	FALSE	28.125	23.406
80	TRUE	FALSE	FALSE	26.402	17.203
40	TRUE	FALSE	TRUE	31.654	56.109
60	TRUE	FALSE	TRUE	28.125	33.640
80	TRUE	FALSE	TRUE	26.402	25.594
40	TRUE	TRUE	FALSE	not applicable	not applicable
60	TRUE	TRUE	FALSE	not applicable	not applicable
80	TRUE	TRUE	FALSE	not applicable	not applicable
40	TRUE	TRUE	TRUE	42.949	49.016
60	TRUE	TRUE	TRUE	38	29.375
80	TRUE	TRUE	TRUE	34.465	20.781

Table 5-2 Deceleration times for two-shaft gas turbine

Deceleration					
FUEL (% of DP)	VSV	STEADY	HANDLE	TRANSIENT TIME (seconds)	CPU TIME (seconds)
80	FALSE	FALSE	FALSE	42.025	25.891
60	FALSE	FALSE	FALSE	58.51	43.75
40	FALSE	FALSE	FALSE	81.319	74.375
80	TRUE	FALSE	FALSE	25.494	16.328
60	TRUE	FALSE	FALSE	37.583	27.859
40	TRUE	FALSE	FALSE	51.602	46.766
80	TRUE	FALSE	TRUE	25.494	16.234
60	TRUE	FALSE	TRUE	37.583	28.281
40	TRUE	FALSE	TRUE	51.602	45.688
80	TRUE	TRUE	FALSE	not applicable	not applicable
60	TRUE	TRUE	FALSE	not applicable	not applicable
40	TRUE	TRUE	FALSE	not applicable	not applicable
80	TRUE	TRUE	TRUE	36.975	22.109
60	TRUE	TRUE	TRUE	49.119	36.609
40	TRUE	TRUE	TRUE	69.792	54.328

For deceleration response time, the same analogy as acceleration was used. Starting at the maximum power the engine was gradually decelerated by reducing the fuel flow to similar levels as that of acceleration so that any differences could be spotted. The corresponding speed setting would be approximately 93%, 85% and 72% of design value. The step size of fuel flow was again chosen to compare the two transient times against one another. It is noticed that between the two, acceleration occurred faster than deceleration. The computing time was also noted and has no physical relevance except that it depends upon the computer's hardware configuration. The Figure 5.5and Figure 5.6 show the difference between the time taken to accelerate and decelerate for the same scene. Because the scheduling is not created as a function of NDOT, the time variation occurs.

In Figure 5.7 and Figure 5.8 the running line on the compressor map is plotted. When there is no fuel control or scheduling (Figure 5.7) the engine passes the surge line to accelerate. The line is flatter and as the pressure ratio increases more mass flow enters the engine and the line starts moving away from surge. The deceleration also occurs for a longer time in this case. When control of the engine is kicked in (Figure 5.8) even if it seems that surge is occurring, actually the stator vanes open to allow more mass flow (see compressor map plotted as the dotted line). Though the time of acceleration is not affected it happens in a safer environment. As for deceleration, it is much quicker because there is a big surge margin available and fuel is cut off according to the demand. Therefore deceleration occurs much quicker (approx. 12 sec. lesser)

For the single-shaft gas turbine comparisons were not made because it is the author's opinion that the link between the compressor turbine and power turbine being a ratio given by the gearbox formula, it will not effect the response time and the representation would be far from realistic. A comparative study of total time was made however to check the overall response rate.

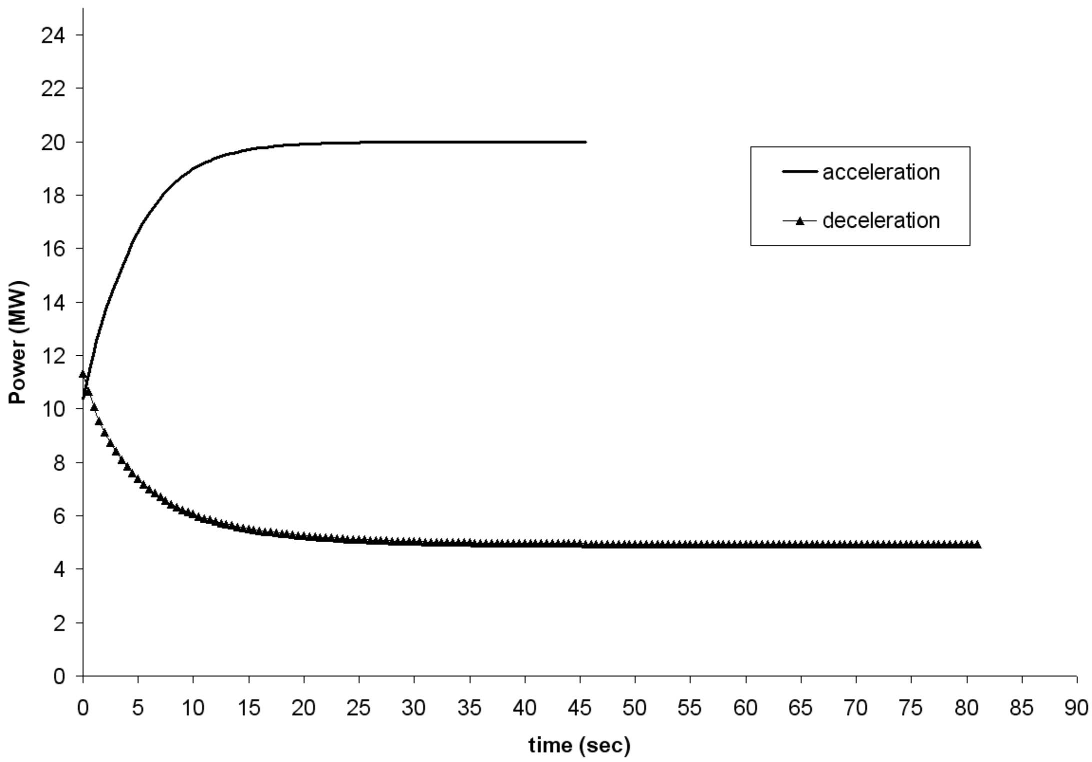


Figure 5.5 Response time for two-shaft gas turbine without control

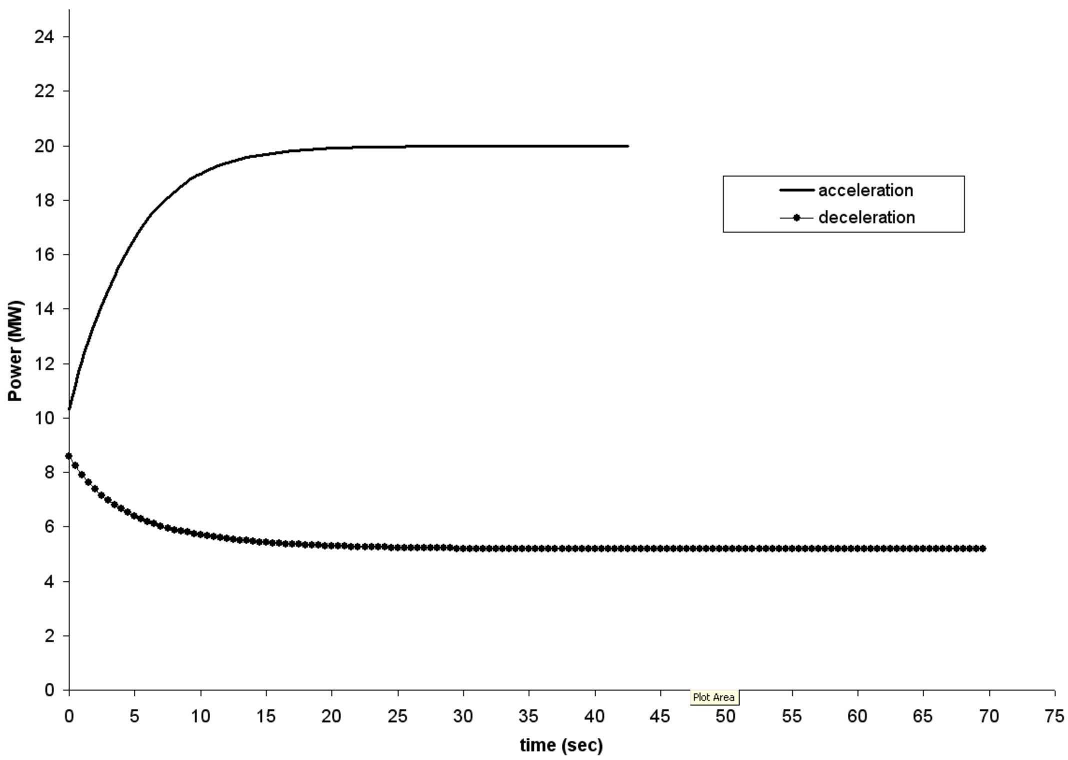


Figure 5.6 Response time for two-shaft gas turbine with full control

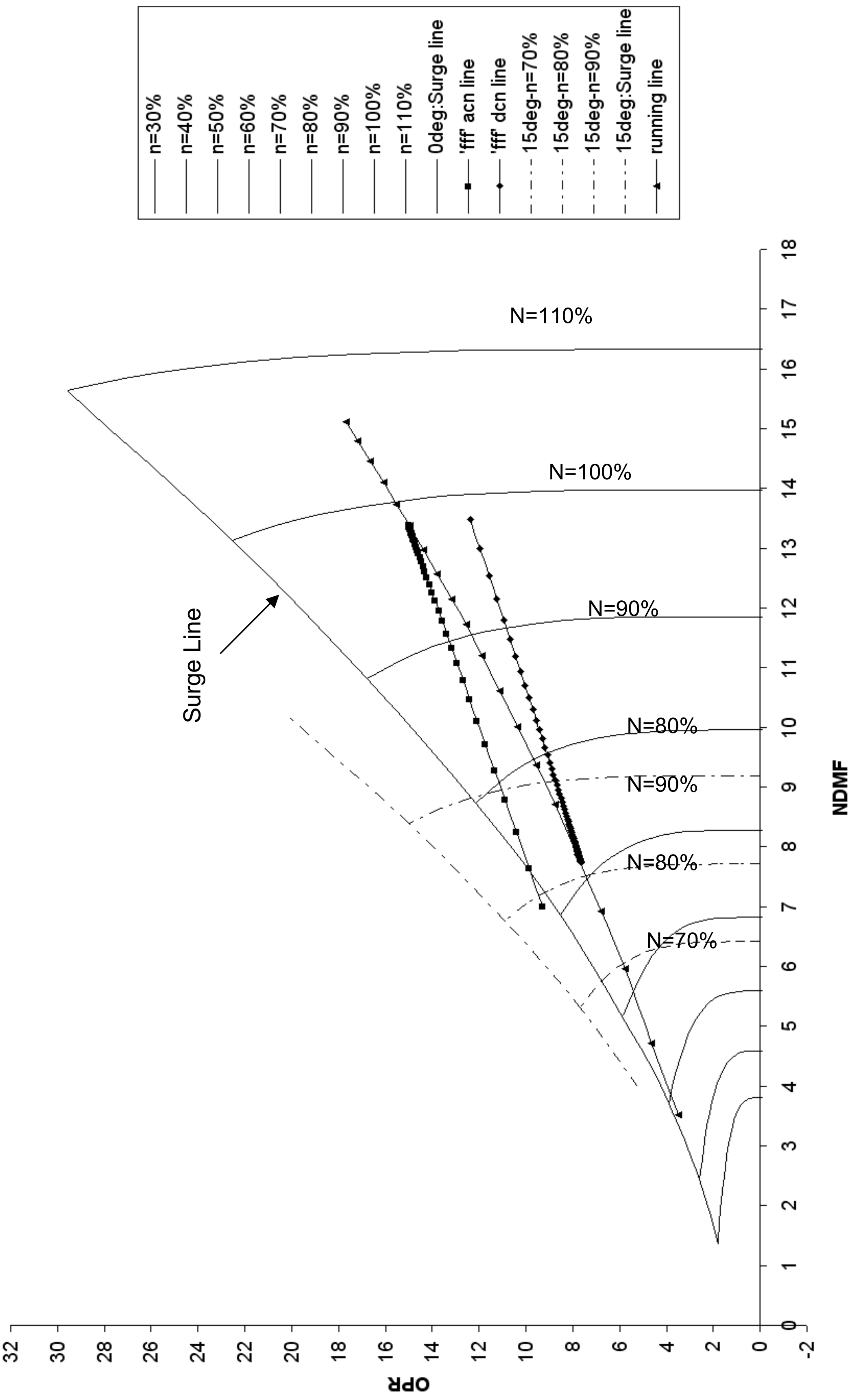


Figure 5.7 Transient lines without control schedules

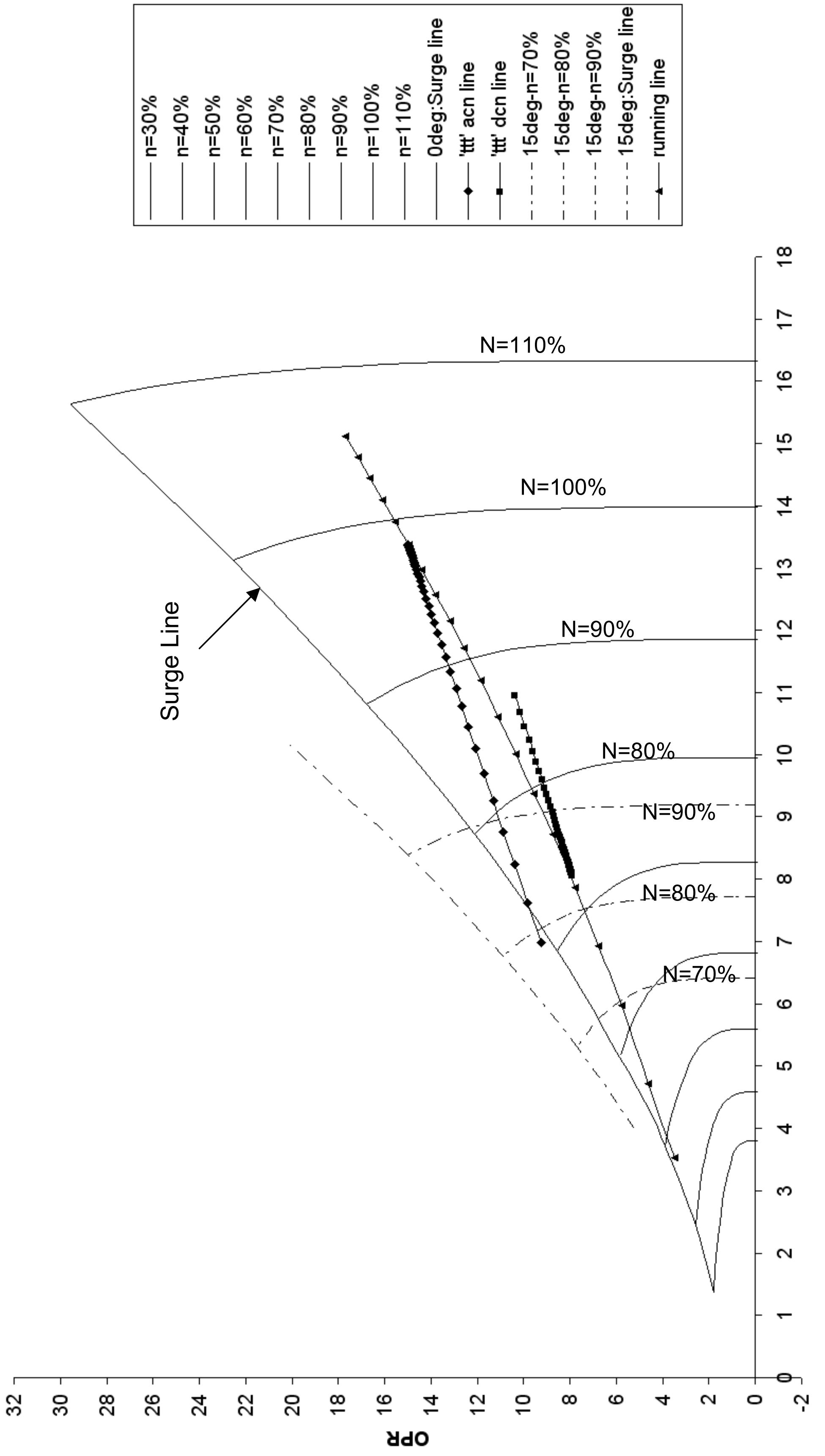


Figure 5.8 Transient lines with variable stator vanes and fuel schedules operating

5.6. Intercomponent Volumes for Transient Modelling

It is well understood that any physical model should be able to accurately represent the phenomenon of the system. Thermodynamics constitutes to the branch of physics and is the basis of all physical models that represent the gas turbine. Fundamental law of an ideal gas states the relationship between pressure, volume and temperature which is well-known. In the method of constant mass flow, the volume of the system is secondary to shaft dynamics. In the method of constant mass flow the assumption that the volume of mass between components is smaller than the mass flow may be true in certain cases, but by ignoring volume from gas dynamics makes the physical model only an approximation in certain scenarios. To the aspect of transient modelling of gas turbine, the method of intercomponent volumes provides a closer look into its thermodynamics. Therefore an implementation of volumetric analysis makes it possible:

- to consider the effects of heat transfer in transient prediction
- to design better control systems and algorithms
- to predict transient behaviour irrespective of the amount of rotational speed change
- to robustly model the gas turbine

5.6.1. Conservation laws

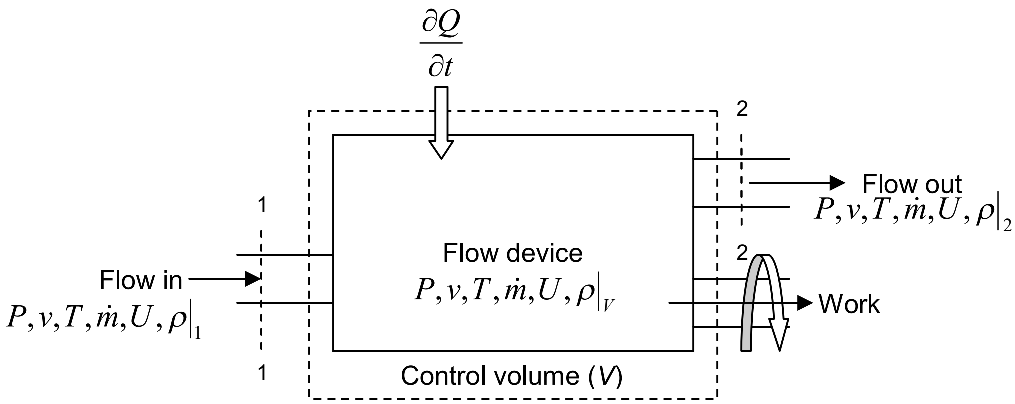


Figure 5.9 One-dimensional flow process diagram for ICV analysis

Consider the system (Figure 5.9) surrounded by a control volume in which one stream of fluid enters at (1) and the same stream leaves at (2). If $v_1 = v_2$ and there is no accumulation of mass or energy within the control volume, and the properties at any location within the control volume are steady at time then the system is in a state of steady flow. The principle of conservation of mass balance and energy balance are true at the time $t=0$ and are usually given in their differential form as[48];

Continuity equation (rate of change of mass flow)

$$\frac{\partial m}{\partial t} = \dot{m}_1 - \dot{m}_2 = \Delta \dot{m} \tag{5.19}$$

Energy equation (rate of change of energy)

$$\frac{\partial U}{\partial t} = \dot{m}_1 H_1 - \dot{m}_2 H_2 + \frac{\partial Q}{\partial t} \tag{5.20}$$

Or,

$$\frac{\partial U}{\partial t} = \Delta(\dot{m} \cdot H) + \dot{Q} \tag{5.21}$$

Momentum equation (rate of change of momentum)

$$\frac{\partial(\dot{m})}{\partial t} = \frac{A\Delta(kP)}{L} - \frac{R_x}{L} \tag{5.22}$$

When the system is in a steady-state all its station vectors are described by the design point. If a transient is initiated, for example by a change in fuel flow, it is known that a mismatch in work and flow is created. To calculate the rotor dynamics, work mismatch can be applied to the rotor dynamics to calculate speed (in a similar fashion as the method of constant mass flow) while the intercomponent volume along with gas laws (eqns. 5.19 - 5.20) can be utilised to find the thermodynamic station vectors of the engine. There are a few things to note about this method;

(a) The size of the volume chosen is usually much bigger than its physical size, because the shaft dynamics are not concerned with pressure changes and making the volume size equal to the physical size may create instabilities.

(b) The time step should be chosen appropriately because a very small step causes oscillations and a large step causes smoothing of the transient path.

(c) The calculation time required to predict transient performance may be higher but with the use of robust mathematical methods to solve the set of non-linear equations may make real time simulation of GT performance permissible using ICV method.

6. HEAT EXCHANGER MODELLING

The aim of a gas turbine based engine designer is to provide a machine in which the cycle efficiency (eqn. 6.1) is as close as possible to unity. The objective of the manufacturer of gas turbine engine would be then to create such a machine in which all the individual components perform their work with maximum efficiency and minimum losses. In a gas turbine, hypothetically this means if such there exists a medium whose ratio of specific heats is high, then the cycle can provide useful work with minimum compression and minimum fuel. At the current technological levels, a simple cycle efficiency for an aero-derivative engine hovers around the value of 0.40 with the isentropic efficiencies of 0.85 – 0.90 for the turbomachinery.

$$\eta = \frac{Net\ Work\ Output}{Heat\ Input} = 1 - \left(\frac{1}{\pi_{CMP}}\right)^{\frac{\gamma-1}{\gamma}} \tag{6.1}$$

From the above analogy, the efficiency of the gas turbine is a factor of the pressure ratio of the gas turbine, if the properties of gas are maintained as a constant. If the pressure ratio decreases the overall efficiency falls because as the engine is levered back in power demand, the ratio π_{CMP} is lower. Therefore a simple cycle gas turbine is best operated at full power settings. For a marine vessel, the cruise speed is not the maximum speed thereby the demand for its propulsive power is lower. If the gas turbine installation is selected as a prime mover, its inefficiency at part load would make it economically non-feasible at low speeds. To overcome the part load performance reductions, alternate forms of gas turbine configurations are considered. One cycle of the discussed group of advanced cycles (section 1.4.3) has been selected to power the type 45 destroyer of the United Kingdom Royal Navy, viz. the Intercooled-Recuperated Engine WR-21. Such an engine employs the use of two additional components to a gas generator, an intercooler and a recuperator to add value in terms of performance. Pressure rise in a single compressor to a certain limit depends on the maximum capacity of temperature rise it can get. The use of just an intercooler between two sections of a compressor or between two whole compressors allows the engine to achieve higher pressure ratios. This augment of pressure also results in increased mass flow, which with lower temperatures has a net result of decrease in the amount of turbine work needed for compression. At part power the compressor pressure ratio and turbine entry temperature is lower than the maximum power setting and the efficiency of the turbomachinery is also less than its peak value. A recuperator adds to the benefits offered by cooled compression and reduces the amount of heat energy required to normally operate the gas turbine by reheating the mass flow entering the burner through the exhaust gas temperature of the power turbine. It is well established that the exhaust gas temperature of an engine is much higher than ambient temperature. The recuperator of the ICR recovers some of this heat to raise the value of temperature entering the combustion zone. Therefore the amount of fuel required; to achieve desired value of the turbine entry temperature by the combustion process is reduced, effectively reducing the heat input of the main cycle but obtaining the same amount of useful work. Summing up the two effects helps to gain a significant higher value of the cycle efficiency at normal and part load operation.

6.1. Overview of heat exchanger types

The other novel cycles suggested too involve the use of some form of heat addition or deduction to the conventional gas turbine. This phenomenon happens outside the standard path of air flow in the gas turbine, in a separate module or component. The component is called a heat exchanger, irrespective of the result it produces. The concept of heat transfer occurring in the heat exchanger is similar to the concept of heat effects of the mass flow within the gas turbine. However, the performance output due to the same concept is radically different in a simple gas generator engine and an engine with the same core and an added heat exchanger.

Energy can be exchanged by two fluids in various forms. If the difference of energy is a change in temperature of the fluid, then it is said to undergo a process of heat transfer. For this exchange to occur the two fluids have to be at different temperatures. Then the problem of determining the rate at which the mediums exchange energy to form a thermal equilibrium is associated to the solution provided by heat transfer analyses. To reach a steady state of thermal equality heat is transferred in the natural forms of convection, conduction and radiation. If these processes are stimulated by an external medium then the solution lies in the determination of forced heat transfer. To design an effective heat exchanger, the physics and physical arguments of heat transfer should be familiar to the designer. There are numerous textbooks available which describe the physics of heat transfer and its effects. The author found the explanation of the authors Welty et al[49] and Cengel[50] precise in understanding the phenomenon and problems associated with heat transfer.

The novel cycles of gas turbine engines use additional heat exchangers to either cool or to heat the working fluid. If the application of the heat exchanger is to cool the air so that it can be compressed to higher pressure ratios then it is known as an ‘intercooler’. If the mass flow is heated before combustion the type of heat exchanger is a ‘regenerator’. Precisely, when the exhaust gas of a simple gas turbine is employed in the same cycle the module is a ‘recuperator’, but if the energy is supplied by an external source then it is a ’reheater’. The process of energy exchange in a heat exchanger can occur in various forms. When the energy exchange between two fluids occurs by heat transfer through a separating wall, by heat passing from one fluid through the wall to another fluid the exchanger is a direct transfer type. If there is mixing of the two fluids involved, it is known as an indirect transfer type. The classification of heat exchangers can be based on numerous factors like the number of fluids, geometrical construction of the exchanger, chemical and physical processes occurring in the exchangers, etc. Figure 6.1 illustrates the wide variety of classes in exchanger types.

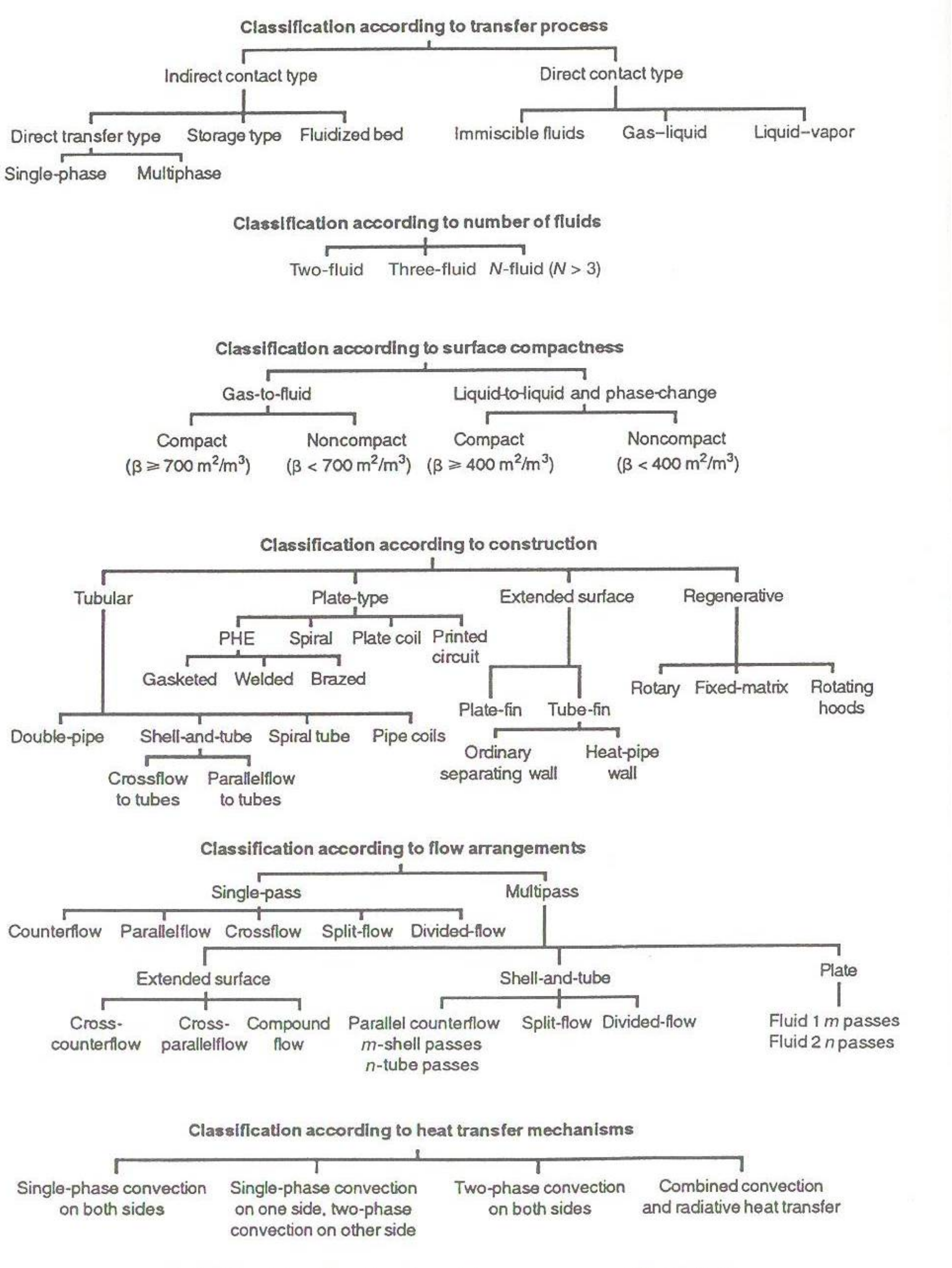


Figure 6.1 Type of heat exchangers[51]

6.2. Definition of Heat Exchanger Modelling

The definition of problem for heat exchanger modelling is complicated and the choice is one of many. Based on its type of application, available space for installation, possible flow arrangements together with number of fluids involved in the heat transfer process, boundary conditions of operation and the need for investigation of a particular problem or phenomenon stretches outside the scope of investigation of one project. Therefore based on certain assumptions and the basic requirements to design a heat exchanger was deduced as follows;

- To create a model of heat exchanger best suited for both the type of heat exchangers in an intercooled-recuperated gas turbine.
- To create a single model for performance of that heat exchanger in steady state and transient conditions
- To provide the results of sizing and rating based on the assumptions and comprehend the nature of heat transfer
- To design a heat exchanger in order to simulate an ICR, to be incorporated in the GT-EN interface of AMEPS

By selecting one single configuration which will suit multiple types of heat exchange applications, analytical and numerical solution techniques can be investigated more comprehensively. When the use of the heat exchanger is for gas turbine performance investigation, the true dynamics of the heat transfer process are of second nature to the performance output. A certain level of tolerability in the heat exchanger results will achieve accurate results of the performance because its prediction requires only a few thermodynamic properties of the fluids to gain the values of total work output and the thermal efficiency of the cycle. This assumption also provides flexibility in the GTPS model and various configurations of heat exchange cycles can be analysed. The drawback in the theory is that of possible execution of the obtained design as a practical application may not be possible.

To narrow the choice of types of heat exchanger that would fit the installation on marine vessels of a certain class, some literature background was researched. The WR-21 engine was selected as the frame of reference for the development of the heat exchanger model. This engine is designed with a recuperator enclosed with the gas generator and an intercooler system of saltwater heat exchanger to fit the engine room space of the USS Arleigh

Burke[52]. Although the geometrical dimensions of the heat exchanger are not a major factor for selection in this project, emphasis on minimum weight of gas turbine installations is always applicable. For engines used on mobile systems the power to weight ratio is a major factor of design of the propulsion system. The ratio translated in terms of specific power for shaft cycles, is a first order estimate of the total engine weight, volume and frontal area. This analogy if adapted to the dimensions of the base engine, assist in narrowing the types of heat exchanger designs for the ICR. But here are a lot of unknowns at the beginning of the heat exchange model and fundamentally it would be right to start with a simple design and progress towards more complex structures. Heat exchanger like the shell-and-tube (Figure 6.2) can be split into a single section unit depending upon the flow arrangement and by making some simple assumptions; the problem of modelling can be converted to the one of finding the heat transfer effects in a single pass heat exchanger with a counterflow or a parallelflow plan.

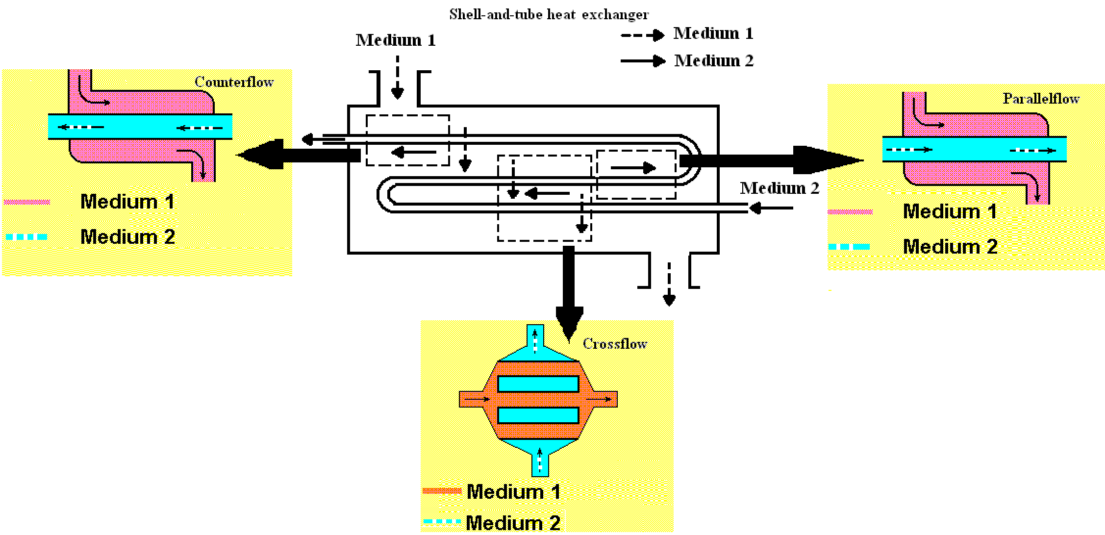


Figure 6.2 Shell-and-tube and single flow heat exchangers

There are assumptions required to model a complicated system like the heat exchanger. Some of them made are classical, whereby after conducting numerous studies to the effect of the assumptions they have been proved to be valid. These stand true in most of the cases, but may need to be reconsidered if the problem definition is changed or altered. To fit the solution of an ICR design point performance modelling a few other variables have been considered as constants. These constants are the thermodynamic properties of air and the cooling liquid, which is water in this case for the heat exchangers. The specific heat capacity of air at constant pressure is a function of temperature. For ambient conditions of temperature, its value can be considered constant. Its value at the face of the passage is fixed and kept constant through the flow. This assumption is true for an intercooler as the influence of the specific heat at constant pressure of water which is about 4 times higher is the deciding factor in the relationship for the rate of heat transfer. In the case of a recuperator the dependency is altered. The assumption of heat capacity rate is dealt as a constant should be applied cautiously.

To calculate the heat transfer effects, fluid flows in one passage follow one and only one direction and do not change the flow path or phase. The water when heated is bound to start evaporating. This phenomenon should be considered separately outside the boundary of the intercooler design and the condition could be set here, that the change of phase occurs at a constant temperature for any one fluid at constant pressure. All the accumulation of mass flow is considered in a single pass and the heat transfer is evenly distributed along each side of the fluid. The temperature in a passage is considered uniform and there is no fluid mal-distribution along the cross section[53]. The fluid flow rate is constant across its channel during steady state. In the transient phase, this assumption is relaxed and flow rate is calculated. However no flow stratification; in a direction parallel or counter to the flow direction of the fluid; flow bypassing and leakages occur in any state. There is a complete balance of thermal energy and no storage or sinks occur in the fluids or walls of the heat exchanger. If the outside walls of the heat exchanger are adiabatic, heat transfer then happens only across the channels and longitudinal heat conduction effect is negligible for the fluids in a parallelflow heat exchanger, leaving it from the general methodology of modelling assists in formulating a procedure. To create a single model by which the temperatures at the exit of the heat exchangers can be calculated, keeping the individual heat transfer coefficients independent of time, position and temperature; which is maintained uniformly; a solution can be provided for the heat exchanger modelling problem.

The design methodology of any heat exchanger is typically split into three major parts. To begin the process based upon the customer or user needs, a formulation of the problem is needed. This leads to a process specification that involves specifying the operating conditions, type of heat exchanger, flow arrangement, materials selection any use of emerging technologies etc. In parallel the process of costing and manufacturing methods and trends are carried out to investigate the feasibility of the chosen design by engineering logic. For simulation studies and research purposes, the latter part is usually omitted from the process specification. Then a hydraulic and/or thermal design of the exchanger needs to be performed. This is the *rating* problem. Given in the case of application or assumed in the case of research the operating terms, rating a heat exchanger is to evaluate the heat transfer rates, exchanger effectiveness and the outlet fluid properties like the temperatures, pressures and flow rates. A successful output follows with a mechanical design process. Determining the geometry of the exchanger, and evaluating the core to achieve the temperatures and effectiveness predicted by rating, the mechanical design and thermal analyses are two process which have to be performed in parallel or at least in progression starting by steady state modelling followed by sizing and concluding in an off-design or dynamic modelling. When an optimized solution is achieved, the heat exchanger can go into production and with systems optimization approach the end product is reached.

The following part of the chapter deals with the thermal design and geometry prediction using theoretical solutions of first order accuracy. Three aspects of heat exchanger thermal design modelling are considered. The modelling methodologies are based on the heat exchanger problem formulation aforementioned and solution for steady state is provided from the literature review conducted while the transient model was developed for an intercooler by Mr. Pellicer Ausias and for a recuperator by Mr. Pastor Francesc; who were pursuing their Master of Science degree at Cranfield University. The author of this thesis was a point of support for technical advice and had to implement the algorithms to simulate an intercooled-recuperated gas turbine.

6.3. Steady state modelling algorithms

Several established techniques of steady state modelling are practised in the industry to analyse the thermal design of the heat exchanger. "The applicability of the data in the design and performance evaluation of heat exchangers can be assessed from the list of assumptions in the previous section. It must be recognised that, even when the method is applied to single-phase fluids, the assumptions are unlikely to be met completely. Analyses of designs of heat exchangers where small variations of overall heat transfer coefficient or specific heat capacity occur, show that methods can be applied using average values of the varying parameters without introducing significant errors"[54].

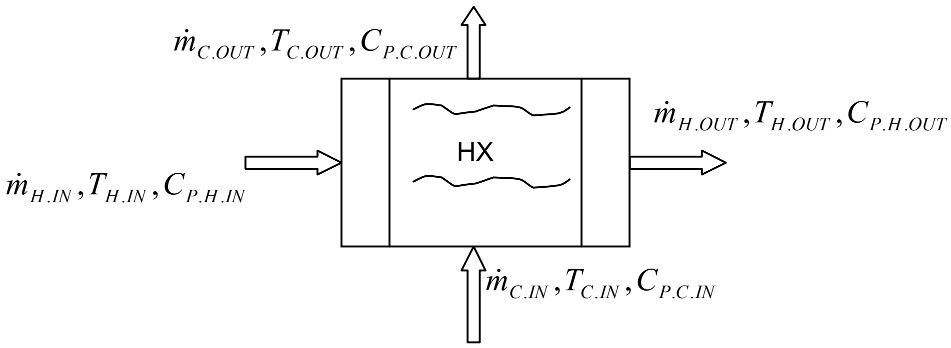


Figure 6.3 Simplified single-phase single-pass heat exchanger

Consider a simple heat exchanger as shown in Figure 6.3, the thermal design process lies in the evaluation of the fluid outlet temperatures of the two streams ($T_{C.OUT}$ and $T_{H.OUT}$), effectiveness of heat transfer ε and rate heat transfer Q. This figure could represent any heat exchanger and if the effectiveness of the exchanger is known then each of the other values can be calculated using any one of the few methods mentioned below:

6.3.1. The ε-NTU Method for Heat Exchanger Design

The solution of this method lies in the definition of three dimensionless parameters. These parameters together signify the overall working potential of the exchanger and depending on the flow arrangements they provide correlations between different types of heat exchangers. The first parameter of the group is heat exchanger effectiveness ε;

$$\varepsilon = \frac{\text{actual rate of heat transfer}}{\text{maximum rate of heat transfer}} = \frac{Q}{Q_{MAX}} \tag{6.2}$$

It can be considered as a measure of thermal performance for the exchanger. A more accurate meaning of effectiveness would be that the first law of thermodynamics is satisfied at each point along the heat exchanger. This interpretation is useful to develop an analogy for the solution of heat exchanger transient problem.

Number of transfer units or NTU is a heat transfer size parameter of the exchanger. This ratio can be derived to suit the problem at hand, but physically it is;

$$NTU = \frac{UA}{C_{MIN}} \tag{6.3}$$

It is an indication of surface area which may be required to achieve a transfer rate of heat energy. NTU is therefore a design parameter and also a magnitude of heat transfer.

The third value in the group is the ratio of the smaller heat capacity fluid to the higher heat capacity fluid C^*.

$$C^* = \frac{C_{MIN}}{C_{MAX}} = \frac{(\dot{m}C_P)_{MIN}}{(\dot{m}C_P)_{MAX}} \tag{6.4}$$

This is an operating parameter which indicates that a fluid with a lower specific heat at constant pressure will experience a smaller temperature change as compared to the second fluid. In an intercooler, it is thus essential to select a fluid with high values of specific heat for a quicker cooling effect.

Now, consider the single-pass single-phase exchanger as a counterflow heat exchanger, dividing the fluid flow over its length into smaller elements of length dx and applying the assumptions of uniform velocity $dv=v$, cross-sectional temperature distribution, surface area $dA=A$ (assumptions in the previous section) coupled with steady state flow, the Figure 6.3 can be represented as

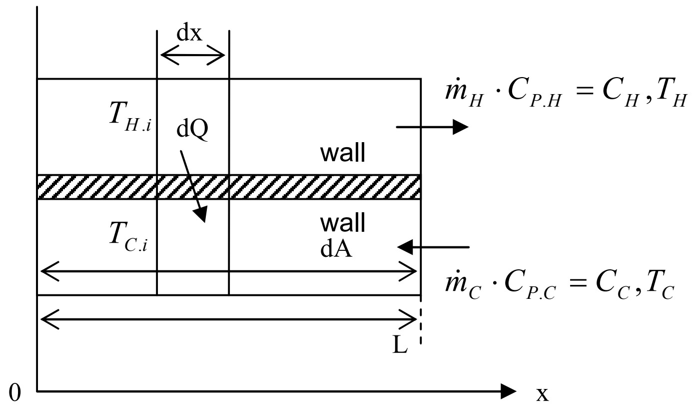


Figure 6.4 Differential element of a single-pass counterflow exchanger

By applying the rate equations for the fluids and wall to the element dx the enthalpy rate of the hot fluid, wall and cold fluid are presented by their differential solution as;

$$dQ_H = h \cdot A\big|_H \cdot \left(T_H - T_{H.WL}\right)\frac{dx}{L} \tag{6.5}$$

$$dQ_{WL} = \frac{k_{WL}}{\delta_{WL}} \cdot A_{WL} \cdot \left(T_H - T_{H.WL}\right)\frac{dx}{L} \tag{6.6}$$

$$dQ_C = h \cdot A\big|_C \cdot \left(T_{C.WL} - T_C\right)\frac{dx}{L} \tag{6.7}$$

The heat transfer from hot fluid to cold fluid is complete, without any lag or additional heat transfer occurring across the wall. Since the transfer rate is similar over the surface area, it can be deduced that,

$$dQ = U \cdot A \cdot \left(T_H - T_C\right)\frac{dx}{L} \tag{6.8}$$

Where dQ_H = dQ_{WL} = dQ_C = dQ in the eqn. 6.8, U is the local overall heat transfer and to find the total rate of heat transfer across a single fluid, boundary conditions are imposed that for $x=0$: $T_H=T_{H.IN}$ and for $x=L$: $T_C=T_{C.IN}$. Substituting these values in the above relation and solving for dq, such that the result is an expression to calculate the heat exchanger effectiveness;

$$\frac{d(T_H - T_C)}{T_H - T_C} = -\left(1 - \frac{C_{MIN}}{C_{MAX}}\right) \cdot \frac{U \cdot A}{C_{MIN}} \frac{dx}{L} \tag{6.9}$$

The heat transfer across the length of the heat exchanger is an integral of the above relation from $x=0$ to $x=L$ or

$$\int_{x=0}^{L} \frac{d(T_H - T_C)}{T_H - T_C} = -\left(1 - \frac{C_{MIN}}{C_{MAX}}\right) \cdot \frac{U \cdot A}{C_{MIN}} \int_{x=0}^{L} \frac{dx}{L} \tag{6.10}$$

Performing the integral and rearranging the mathematical terms it is deduced that,

$$T_{H.OUT} = \frac{\left(T_{H.IN} - T_{C.IN}\right) \cdot e^{\xi \cdot \frac{x}{L}} + T_{C.IN} - C^* \cdot T_{H.IN}}{1 - C^*} \tag{6.11}$$

Where $C^*=C_{MIN}/C_{MAX}$, which is the ratio of heat capacities given by eqn. 6.4 and $\xi = \frac{U \cdot A}{L}\left(\frac{1}{C_C} - \frac{1}{C_H}\right)$ is introduced here for simplification purposes. However it has a physical significance such that if the heat transfer variables are constant then the total temperature rise is a factor of geometrical dimensions of the exchanger. Until now three temperatures of fluids are known and to find the last, applying energy balance across the two mediums the final temperature can be given as,

$$T_{C.OUT} = T_{C.IN} - \frac{C_H}{C_C} \cdot \left(T_{H.IN} - T_{H.OUT}\right) \tag{6.12}$$

For the case of DP calculations in a GT or the rating of heat exchangers the effectiveness is not calculated but is either selected from manufacturer's data or assumed depending on technological level. If it needs to be compounded, then the temperatures at inlet and outlet should be known. The value of NTU

has yet to be found. That can be obtained from the specific heat and effectiveness. The calculation of NTU will then get the value for $U{\cdot}A$ and finally finding the area of the heat exchanger leads to the algorithm of sizing the exchanger. It will be seen ahead that sizing of an exchanger is much intuitive process as a numerical solution. If the same type of analogy is based for a parallel flow heat exchanger, similar equations can be deduced. These equations are available in various textbooks and the thesis of Pellicer[55].

The above method is a simplification of the heat exchanger geometry and flow conditions. The ε-NTU relationships for a direct contact and indirect contact heat exchanger are different. The model as a first order estimate provides good results, but complex heat exchanger modelling for stream asymmetries' in other exchangers like shell-and-tube exchangers is considered, limits this method to application of interpolation techniques for design of those exchangers. By introducing a temperature efficiency factor P in the ε-NTU relationships, the confusion is avoided when solving for stream asymmetrical heat machines. The P-NTU method has been widely used to design shell-and-tube heat exchangers since early years in the last century[51]. This method is better suited to the application of heat transfers in the ICR, but requires some optimization to select the number of tubes to achieve maximum possible efficiency. A simple trial and error solving technique to get an optimum of the configuration is sufficient for steady state modelling.

A third solution, known as the 'log mean temperature difference' is also available. The effectiveness ε, essentially is a factor of the temperature difference. With temperature curves varying for different flow arrangements and geometry of the exchanger, ε is the indicator of the performance. To determine the temperatures of a heat exchanger given the flow rates such that maximum performance (chosen) is always achieved, then unless the temperatures at the two planes of references in the exchanger are known, the solution will be a an approach either based on discretization with many assumptions (ε-NTU) and correcting them for different flow arrangements (P-

NTU). The solution provided by LMTD is based upon the fact that $\Delta T = \Delta T_{LM}$ which are the average values of the temperatures between the exit and inlet being equal to the logarithmic mean of the same difference. The solution of determining the temperatures then lies in applying the eqns. 6.13 and 6.14[56];

$$Q = U \cdot A \cdot F \cdot LMTD \tag{6.13}$$

$$LMTD = \ln \frac{\Delta T_1}{\Delta T_2} \tag{6.14}$$

Where, F is the correction factor for a certain type of heat exchanger and can be used from either available graphs or applying a numerical solution.

All three methods have advantages and disadvantages and the choice of method should be based upon the thermal design and sizing problem. And it cannot be said that one method suits a sample heat exchanger in all conditions because of the assumptions in all three theories.

6.4. Sizing methods for heat exchangers

The definition of sizing in the context of this thesis is the determination of the overall length, height, surface areas and volume of the two channels of an *extended surface exchanger* to gain the required rate of heat transfer and efficiencies so as to be able to predict the transient response of the exchanger. The actual procedure of sizing is one difficult process which requires tradeoffs between the parameters of thermal design and the selection and required values of sizes for individual components of the heat exchanger, such as tube length, diameters, thickness etc. for a shell-and-tube heat exchanger or the plate and fin geometrical sizes for the plate-fine heat exchanger. The final value achieved in that case will be the best optimum for the type of application based on the flow configuration and other dimensionless physical variables.

Because it has been a theoretical development, the algorithms are true for only the assumed output of the design predicted values, which can be considered as a guessing point for a true design. The assumptions of the steady state modelling still count in the sizing process and some additional data is required to perform the sizing. These algorithms are based on theoretical definitions of dimensionless parameters and certain fundamental physical laws that define fluid motion. A problem in physics can be solved by experimental and numerical methods. A numerical procedure to find the heat exchanger dimensions is described further in this section. The ideology between heat transfer and 'fluid' flow is that if a medium (mass) is in motion then depending upon the type of the medium, laws of physics are applicable throughout. Correlations have been derived that show the influence of temperature, either directly (Nusselt number) or indirectly (Reynolds number) on the internal properties of fluids (viscosity, density, conductivity). If the fluid is in motion the internal variables vary considerably. If the heat exchanger is a recuperator the variation of temperature will depend only on the properties of gases, but for an intercooler because one medium is in a liquid state the temperature distribution along the height of the gas path will be substantially different. But because the interest is to find the dimensions of the overall heat exchanger and not the temperature profile, the effect of water viscosity is only bound by its total effect rather than positional effect. By the selection of certain values of the heat exchanger sizes the problem can be solved by iteration until tolerance is minimum.

Consider a heat exchanger (Figure 6.5) with two channels through which the fluid flows. The height, length and width of hot fluid channel is given by Z_H, L, Y and cold fluid channel by Z_C, L, Y

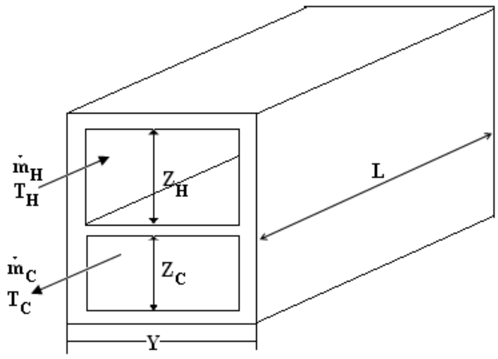


Figure 6.5 Total dimensions of heat exchanger

If the internal properties of the fluids are known or can be found then there exist various methods to find the dimensions of the heat exchanger. To elaborate the one used the following dimensionless parameters have to be defined,

Reynolds number, which signifies the type of flow occurring in the passage is given by,

$$\mathrm{Re} = \frac{\rho \cdot v \cdot D}{\mu} \tag{6.15}$$

The practical significance of the Reynolds number is important in this case for it tells about the effect of viscosity the flow will have on the heat transfer rate. If the number is high value then the heat transfer coefficient is high which is always desirable for the heat transfer process. However, it comes with a price for the power required to pump the fluid is higher.

The dynamic viscosity is a function of temperature and density, and the value for it is

$$\mu = \frac{1.5015 \cdot 10^{-6} \cdot T^{1.5}}{T + 120} \tag{6.16}$$

Prandtl number which is the ratio of molecular diffusivity of momentum to the molecular diffusivity of heat, given as,

$$\mathrm{Pr} = \frac{\mu \cdot C_P}{k} \tag{6.17}$$

Although dimensionless, this can be considered as a property of the medium. To find Pr the coefficient of conductivity has to be calculated. This can be found from reference[57];

$k=1.5207x10^{-11}T^3 + 4.48574x10^{-8}T^2 + 1.10184x10^{-4}T - 3.9333x10^{-4}$ *[watt-m/K]*

From the knowledge of Prandtl and Reynolds number, the third dimensionless quantity Nusselt number can be calculated. This number can be seen as the dimensionless convective heat transfer coefficient. There are various ways of determining this number, with updates typically made in references. The author has used the number found suggested by Pethukov[58].

$$Nu_{H/C} = \frac{f}{8} x \frac{(\mathrm{Re} - 1000)\cdot \mathrm{Pr}}{1 + (((12.7)\cdot(f/8)^{0.5})\cdot((\mathrm{Pr}^{2/3}) - 1)))} \qquad (6.18)$$

Where, $f = (0.79 \cdot \ln(\mathrm{Re}) - 1.64)^2$ is the friction factor in heat transfer.

In all the above equations the closest relation to the geometry of the heat exchanger is given by the Reynolds number. For an internal flow like the one illustrated in Figure 6.5, the value of

$$D_{H/C} = (2\cdot Y\cdot Z_{H/C})/(Y+Z_{H/C}) \qquad (6.19a)$$

To find the diameter one has to define the dimensions of the heat exchanger. Guess a value of Y here as the width is considered constant. If the fluids spend a minimum amount of time dt in the heat exchanger then the total amount of time spent by them would be,

$$t=dt\ x\ n \qquad (6.19b)$$

where n are the number of elements the heat exchanger is divided into. If the velocity of anyone fluid is known by the following formula, one is able to find the length of the heat exchanger.

$$L = t\ x\ v \qquad (6.19c)$$

The velocity of the hot gas in the heat exchanger is a function of Mach, and if the flow is considered incompressible and Mach is constant (initial assumption of exchanger modelling) then from the linearity of $v = M\cdot\sqrt{\gamma}\ R\ T$. velocity is known. At this point by applying the values of internal properties to the fluids (see section 3.7.6) find the area of the heat exchanger from the guessed value of width. Now if the area and length are known, find the dimensionless parameters in the order by solving eqns. 6.15-6.18.

Next proceed to find the overall heat transfer coefficient. For an effect of heat transfer between two mediums the overall coefficient is derived from the point of view that there are two ways of thermal energy exchange and it is natural for any material to resist heat transfer. The total overall heat transfer coefficient can be written as,

$$\frac{1}{U}=\frac{1}{h_H}+\frac{1}{h_C}+\frac{1}{h_{WL}} \tag{6.20}$$

To find U, use $h_H = (Nu_H \, x \, k_H)/D_H$ and $h_C = (Nu_C \, x \, k_C)/D_C$ and $h_{WL} = 0$ was made as that would be a steady state problem. If the product of $U*A$ is now found through actual multiplication of its corresponding values in the sizing algorithm; logic suggests that $U*A$ obtained from the sizing should be equal to $U*A$ at steady state. If there is a difference, then from the right hand terms in eqn. 6.20 it is visible there are no variables of dimensions, signifying that the value of $U*A$ now is just a function of area. Therefore, to find the right equality between the two $U*A$ the dimensions of the heat exchanger have to change to fit values of $U*A$ suggested from effectiveness in the steady state algorithm and values from sizing. The choice is only one, to change the area such that the desired effectiveness is obtained. Therefore iteration has to be performed from the point where the width Y of the exchanger was guessed until convergence is achieved. In practical applications the method may not hold ground but as learning tool for students or for further research the method provides an inset into the sizing problem and the amount of factors that could reflect the geometry of design. There are other methods which may be used to size an exchanger in better degree. In his thesis, Pellicer (2007)[59] provides a good explanation of the critiques of such an approach to design theory. The LMTD method of steady state analysis may be a better approximate if used as a background to develop a code for sizing.

6.5. Transient model algorithm

The transient modelling of any device is particularly needed if it has to be implemented in true situations. This also stands true for heat exchangers and if their application is of novel use then the dynamic analysis is all the more required to reduce not only costs, as for gas turbines, but technological risks too. Wolf[60] first tried to provide a general solution for a parallelflow multichannel heat exchanger from a systems point and provided a matrix solution for the exchanger with any number of fluids and any number of channels. Mishra et al studied the effect of temperature and flow perturbations for a crossflow heat exchanger[61]. Their analysis was for a multilayer plate-fin heat exchanger, which they discretized into a symmetrical module and studied a step and ramp change in the fluid flow rates and temperature difference. Their results showed that mean temperatures of either fluid were interdependent on their flow rates, while the variation of flows and temperatures had a relative effect, the effect of hot fluid being an increase and the effect of cold fluid being a decrease in mean temperatures. Abbasov et al suggest their procedures for the dynamic modelling of heat exchangers[62]. The heat exchanger under consideration was a counterflow type. Their models are derived from partial differentiation of the heat exchanger under examination and solving it for boundary conditions. A model similar to theirs is described further in this section which was to be used in the performance simulation of WR21. Though the paper proposes four models only one of them can be actually represent the dynamics of a heat exchanger for the other three depend upon the temperature difference of the flows to be known. The modelling of transient performance of a heat exchanger has been traditionally solved by applying the finite element method. The heat exchanger shown in (Figure 6.4) can be illustrated as follows along its length L

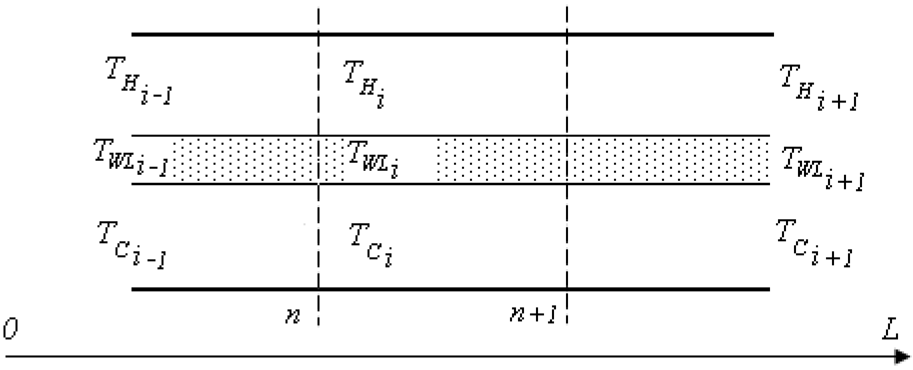


Figure 6.6 Section *dx* of single-pass counterflow exchanger

As the heat exchange is in a transient phase, fluid flow rate is not constant but known always. The wall of the heat exchanger is no more adiabatic but heat transfer takes place through the wall only by convection through mean values of temperature, the wall stores no heat and the coefficients of heat transfer on both fluid sides are known. To find the wall temperature the following eqn. 6.15 should to be solved in a forward calculation scheme.

$$\rho \cdot Y \cdot \Delta x \cdot e \cdot C_P \frac{\partial T}{\partial t} = Y \cdot \Delta x \cdot (h_H \overline{T}_H - (h_H + h_C)\overline{T}_{WL} + h_C \overline{T}_C) \tag{6.21}$$

In the above equation the terms on the left hand side and right hand side are the internal energy of the wall. The left hand side is deduced from the definition of internal energy, while the right hand side is obtained by comparing the eqns. 6.5 and 6.7 which is the rate of energy transfer through the wall in their total forms.

Solving the eqn. 6.21 for time Δt the wall temperature is then given as eqn. 6.22[55 59],

$$T_{WL-i}\big|_{t+\Delta t} = T_{WL-i} + \frac{\Delta t}{\rho \cdot e \cdot C_P}\left(h_H \overline{T}_H - (h_H + h_C)\overline{T}_{WL} + h_C \overline{T}_C\right) \tag{6.22}$$

The above equation calculates the temperature of a very thin wall of thickness e for its mean temperature at a node i for time $t+\Delta t$ along the heat exchanger of length L. This equation is valid only in the case of a fully developed flow. A separate analysis has to be performed if the variation along the boundary of the wall is required at a node. To perform this analysis, one can consider the case of the exchanger as having a multi stream pattern and using the *successive partitioning method* [63] a numerical solution may be derived.

To calculate the time for the hot and cold streams if one performs a analysis on the rate of heat transfer across the channel. With the help of the following set of four eqns. 6.23-6.26

$$dQ_{i,i+1}\Big|_{H,t+\Delta t} = \dot{m}_H C_{PH}(T_{H-i+1} - T_{H-i}) \tag{6.23}$$

$$dQ_{i,i+1}\Big|_{t+\Delta t} = h_H \Delta x Y\left(\frac{(T_{H-i+1} + T_{H-i})}{2} - \frac{(T_{WL-i+1} + T_{WL-i})}{2}\right)\Bigg|_{t+\Delta t} \tag{6.24}$$

$$dQ_{i,i+1}\Big|_{C,t+\Delta t} = \dot{m}_C C_{PC}(T_{C-i+1} - T_{C-i}) \tag{6.25}$$

$$dQ_{i,i+1}\Big|_{t+\Delta t} = h_C \Delta x Y\left(\frac{(T_{WL-i+1} + T_{W-i})}{2} - \frac{(T_{C-i+1} + T_{C-i})}{2}\right)\Bigg|_{t+\Delta t} \tag{6.26}$$

It can be seen that the first two equations and the next two equations are the same i.e. the rate of heat transfer for an infinitesimally small section across in the direction of fluid flow is equal between any two nodes n, $n+1$ (eqns. 6.23 & 6.24) and heat energy given by the hot fluid to the wall the across the wall at any time interval $t+\Delta t$ in any one node n. Solving the equations of finite elements for any one fluid by comparing the pair of eqns. which are similar, obtains the final equations that determine the temperatures of the fluids from the wall temperature calculated by the eqn. 6.22 when the exchanger's overall dimensions are known. These temperatures for the counterflow heat exchanger are calculated as[55 59],

$$T_{H/C}\Big|_{t+\Delta t} = \frac{1}{B}(A \cdot T_{H/C,i} + T_{WL,i+1} + T_{WL,i})\Big|_{t+\Delta t} \tag{6.27}$$

Where, $A=K-1$ and $B=K+1$ and $K = \dfrac{2 \cdot \dot{m}_{H/C} \cdot C_{P.H/C}}{h_{H/C} \cdot \Delta x \cdot Y}$

The above equation is true for hot and cold fluid sections of the heat exchanger because for certain efficiency the temperature difference between the hot and cold flow should be equal. Typically this is true only for the design model, while in off-design cases the efficiency of the heat exchanger changes because the temperature of the hot fluid changes.

7. RESULTS AND DISCUSSION

This chapter presents the results and discussion of the parametric studies conducted on the steady state and transient performance of two gas turbine engines. Each of the two gas turbines are first analysed separately for their performance results and then comparative results are presented. A separate section deals with the results of the prime-mover electrical network model results and explains the changes performed to create the models. A set of results from the transient behaviour of the GT are presented. Finally some results of heat exchanger thermal design are presented.

7.1. Design point performance of GT engines

Figure 7.1 show the effect of bleed extraction for three types of gas turbine engine. The operating conditions (mass flow, compressor pressure ratio and COT) are held constant. With an increase in bleed flow there is a significant decrease in the power output of the gas turbines. For a total decrease in 20% of mass flow the reaction of the two-shaft and single shaft engine is a decrease in approximately 6.1 MW of power output and decrease of approximately 6.8 MW power for the ICR.

Figure 7.2 shows the effect of bleed air extraction on the fuel input for a single shaft, two-shaft and ICR gas turbines. The difference is this case is more significant for the simple cycle engines with an approximate difference of 0.33 kg/sec of lesser fuel flow when bleed air is increased up to 20% of mass flow rate.

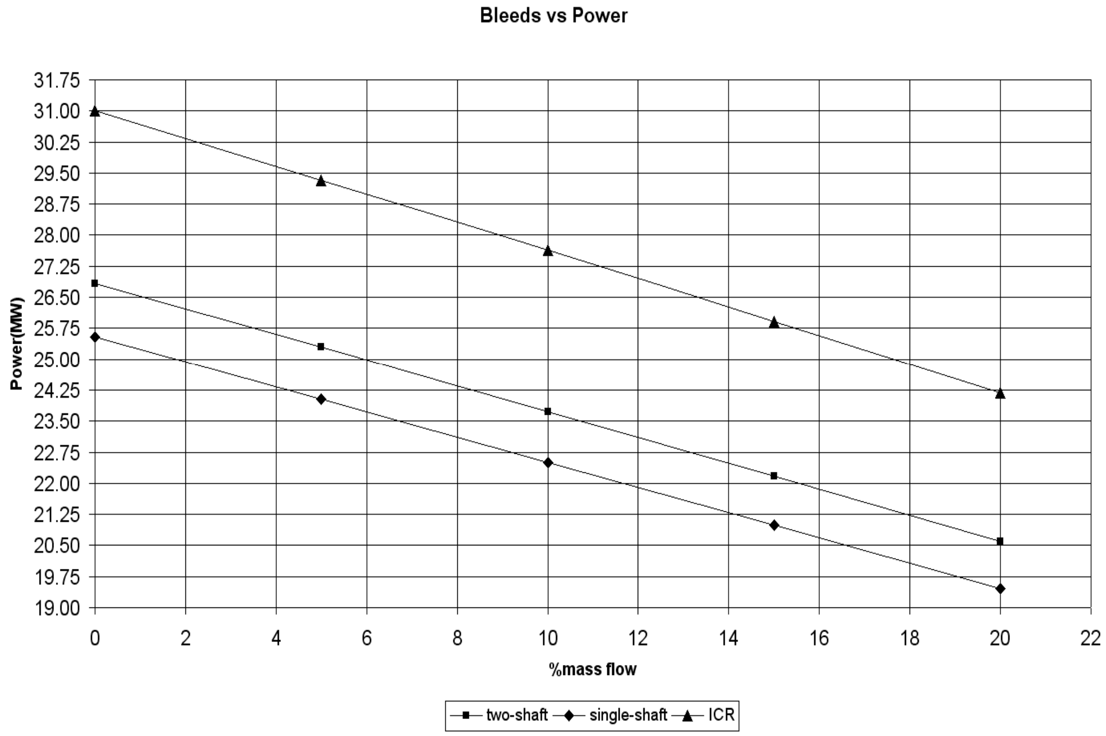


Figure 7.1 Effect of bleeds on power output at DP

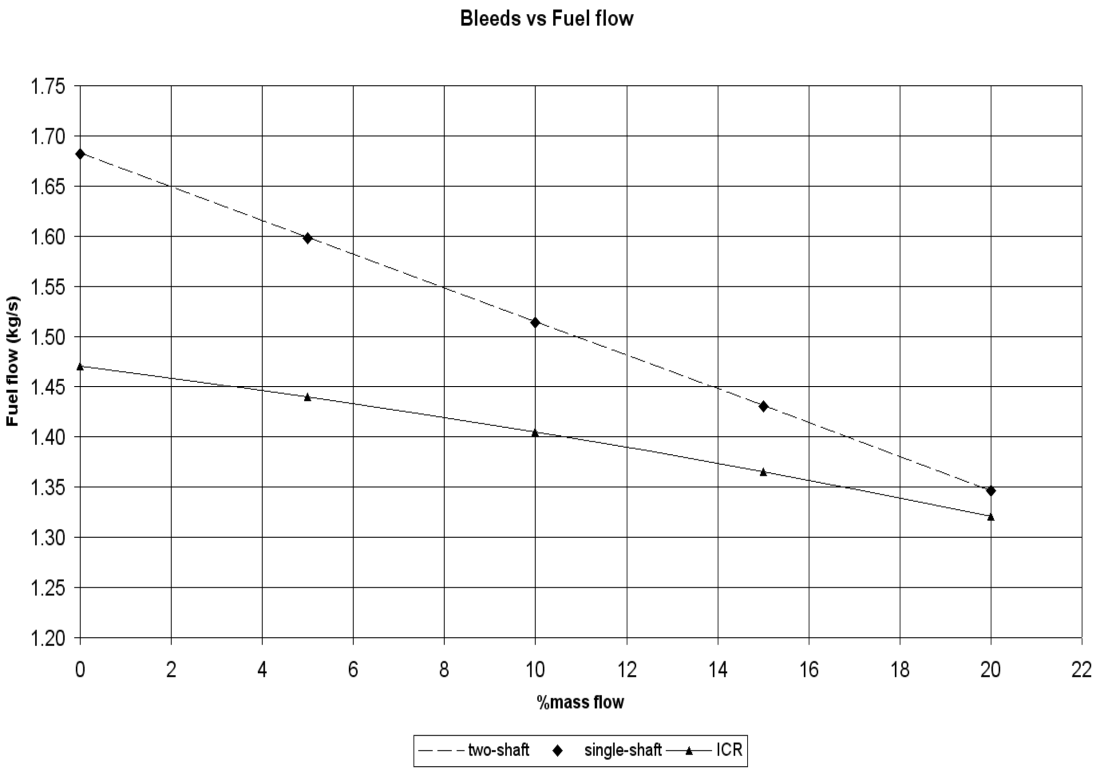


Figure 7.2 Effect of bleeds on fuel flow

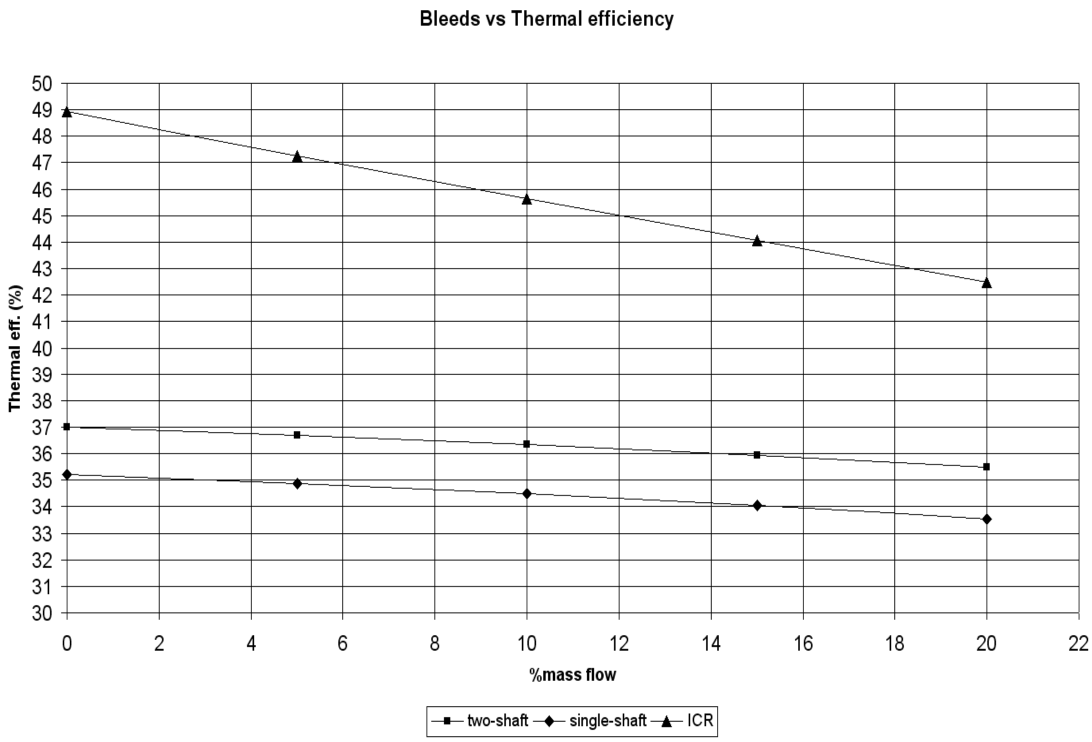


Figure 7.3 Effect of bleeds on thermal efficiency for gas turbines

Figure 7.3 shows the variation of thermal efficiency when compared to bleed extractions. The intercooled-recuperated engine efficiency is significantly affected as mass flow to the heat exchanger decreases. Therefore the steep decline in the thermal efficiency of the ICR compared to the single and two-shaft engine. The effect of bleeds is very significant for the gas turbine engine. If an engine does not have any bleeds then it will have maximum performance. The operating conditions on the compressor map will be fixed for every point and the engine will behave in a singular manner. If bleed valves are present then, there is an option of engine control for emergency manoeuvres. They can be used for safe operation of the compressor during transients.

Table 7-1 Power outputs of gas turbine engines

% Bleed flow	Power (MW)		
	Single-shaft	Two-shaft	ICR
0	25.537	26.837	31.016
5	24.025	25.29	29.329
10	22.507	23.734	27.63
15	20.982	22.169	25.918
20	19.449	20.594	24.192

7.2. Prediction of OD performance of simple cycle GT

In the Figure 7.4 when the ambient temperature is high the compressor requires additional work and correspondingly the work of the turbine is more. Total work delivered by the engine is therefore lower which requires high fuel flow to maintain the COT. As the temperature decreases the effect is positive for the engine and the thermal efficiency is thus higher.

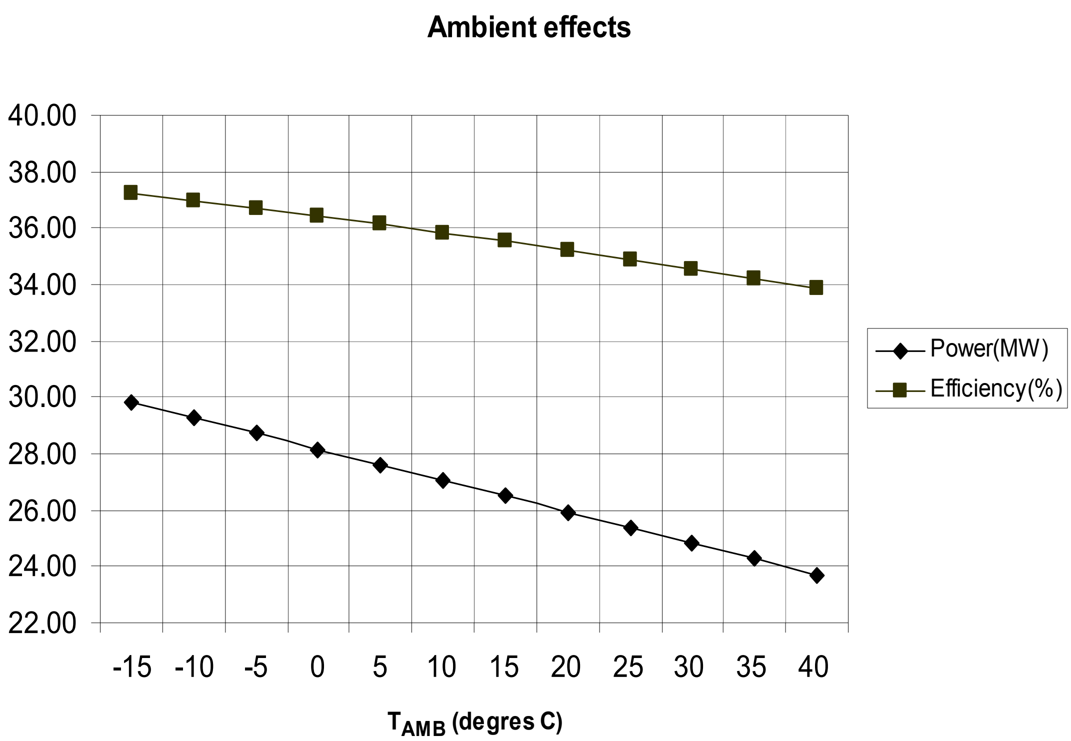


Figure 7.4 Effect of ambient temperature on Power and efficiency

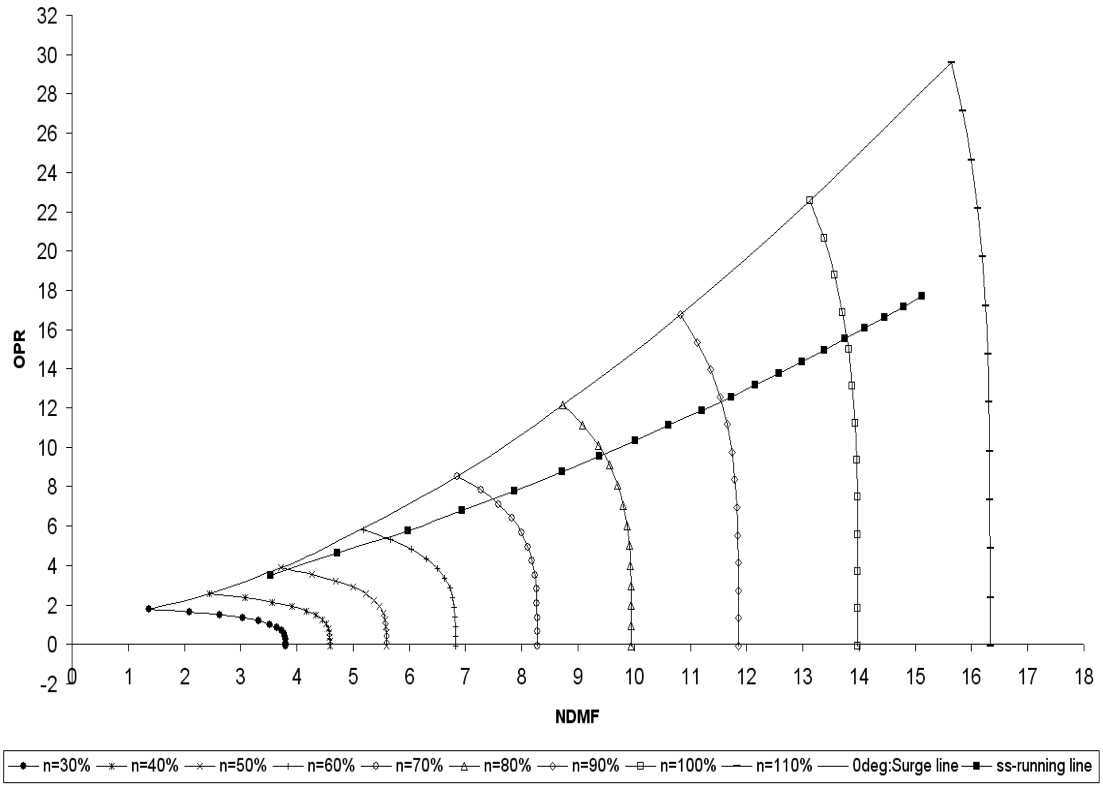


Figure 7.5 Running line with VSV schedule

Figure 7.5 shows the running line of the engine. As the engine is lowered in power setting the compressor starts to move towards surge because of the lack of air for compression. This is indicated by the flat lines at the low end of compressor and the low torque available for rotation. Typically to move out of this region variable area geometry is applied whereby the nozzle guide vanes or the stator vanes are closed.

7.3. GT transient performance

Figure 7.6 represents the time required to accelerate the engine and its corresponding COT is shown in Figure 7.7. The acceleration occurs from conditions of approximately 25%, 50% and 75% of total power (20MW) to max power of 20 MW. At time t =0 when fuel is steady the engine operates at its design temperature. This temperature is set by the conditions of the gas generator. When fuel is suddenly increased or decreased the turbine entry temperature changes. If not controlled the temperature can reach high values like 1950 K as shown in the Figure 7.7. The temperatures of the GT are thus controlled by implementing control schedules.

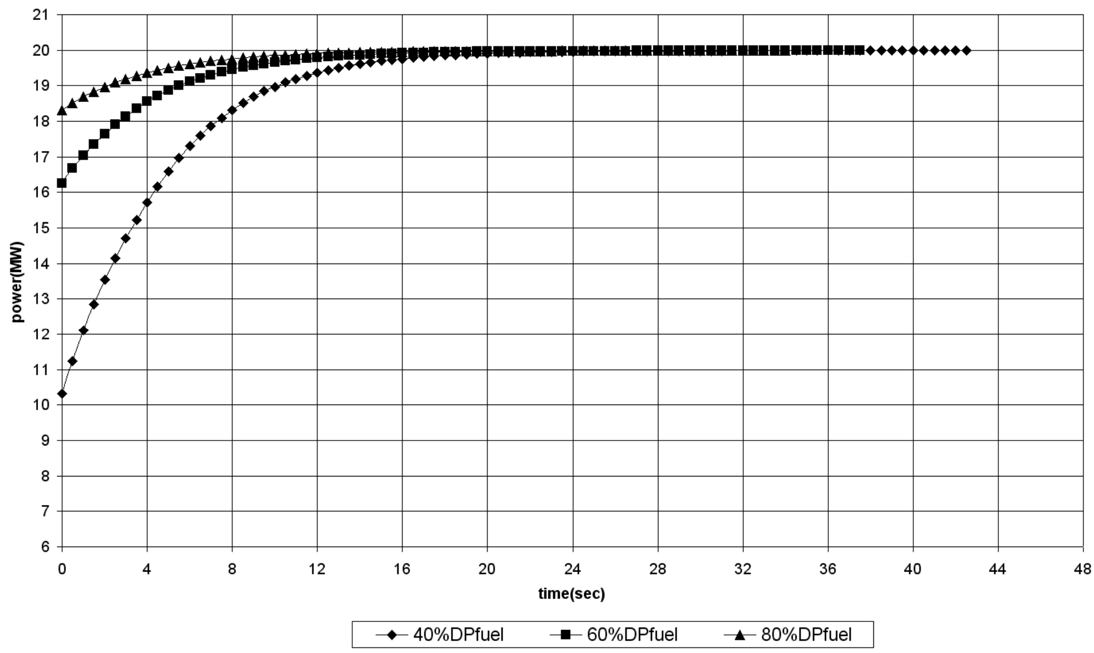


Figure 7.6 Response rate of two-shaft gas turbine

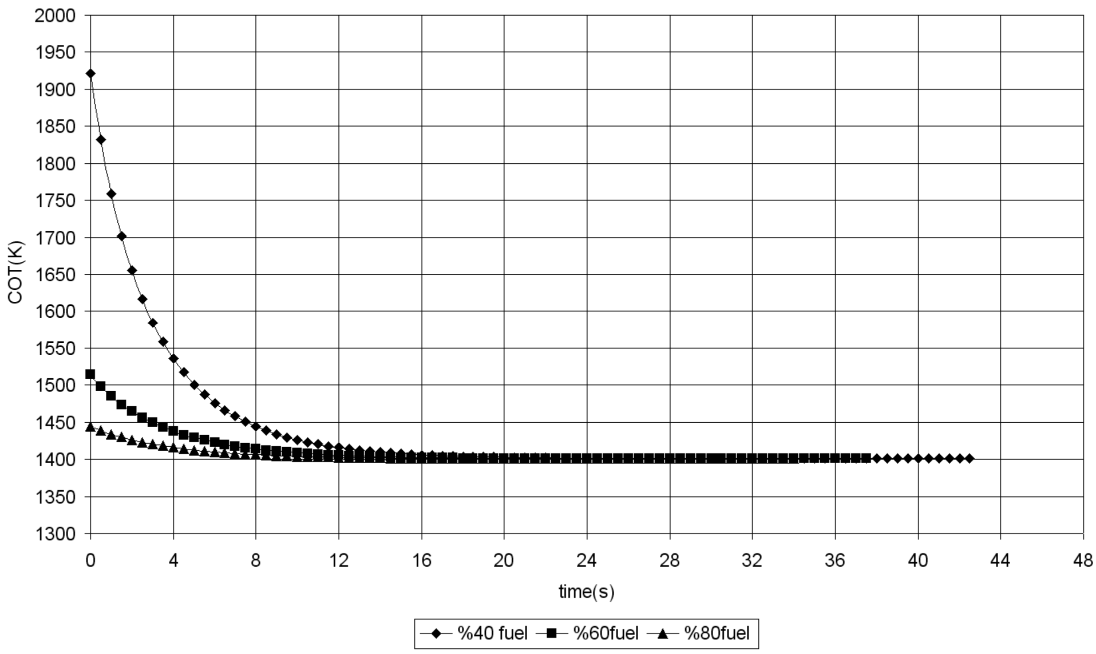


Figure 7.7 Change of COT during acceleration

The following studies on transient simulation were conducted at UoS for reasons mentioned before. It uses the FORTRAN gas turbine models to investigate different scenarios. Though the simulations may represent severe scenarios that may not occur often, it showed the capabilities of the integrated model.

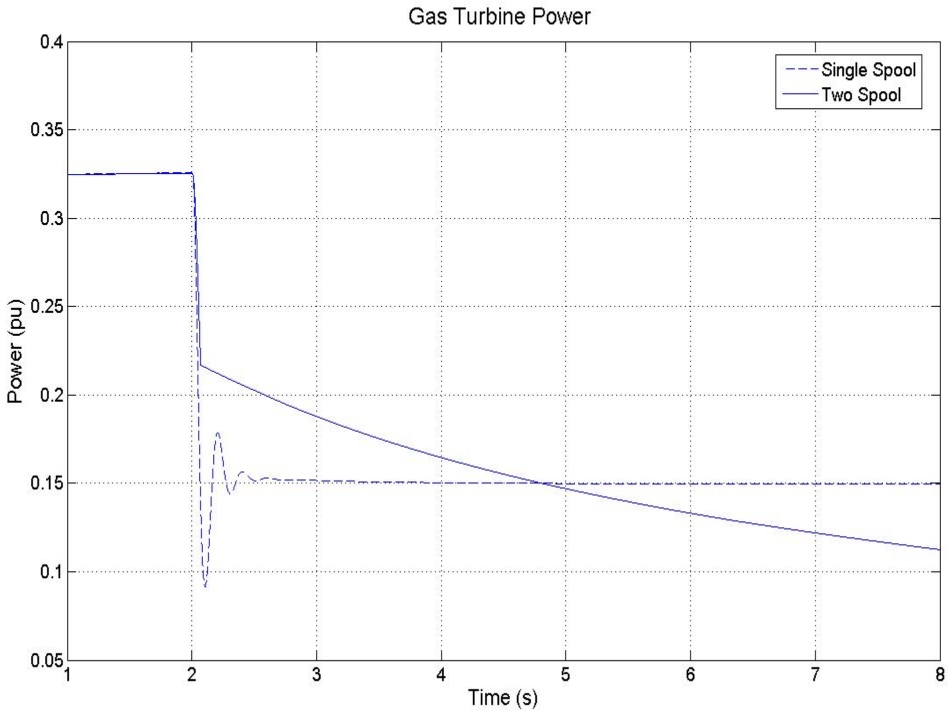


Figure 7.8 Gas turbine response to loss-of-propulsion load

In the Figure 7.8 the response of the two-shaft and the single shaft gas turbine are shown to a failure in the system. This meant that the fuel flow was cut down to a base load of around 3 MW. The power dimension is per unit of 23 MW. Although the response of the single shaft gas turbine had settled the two-shaft engine responded to the fuel demand set by the controller. This behaviour can be attributed to the fact that the single shaft was controlled by the response of the main control system, while the two-shaft gas turbine was controlled by two schedules, one of the system response and the second of the integrated logic in the code. In the second case study checked to see the response of the system it was chosen to load the gas turbine to almost 90% in order to see whether the gas turbine temperature operates in the safe operating range. The result of the case study is shown in Figure 7.9. An upper limit of the exhaust gas temperature was set to be displayed always (Figure 7.10). This temperature was always reached by the gas turbine. This can be explained by the phenomenon of the component efficiencies being not very high. The model was simulated for cyclic loading. This may not be a true scenario, but can be indicative of a case when the propeller of the ship comes out of water due to high sea roughness. Generally in all the cases some form of result was obtained.

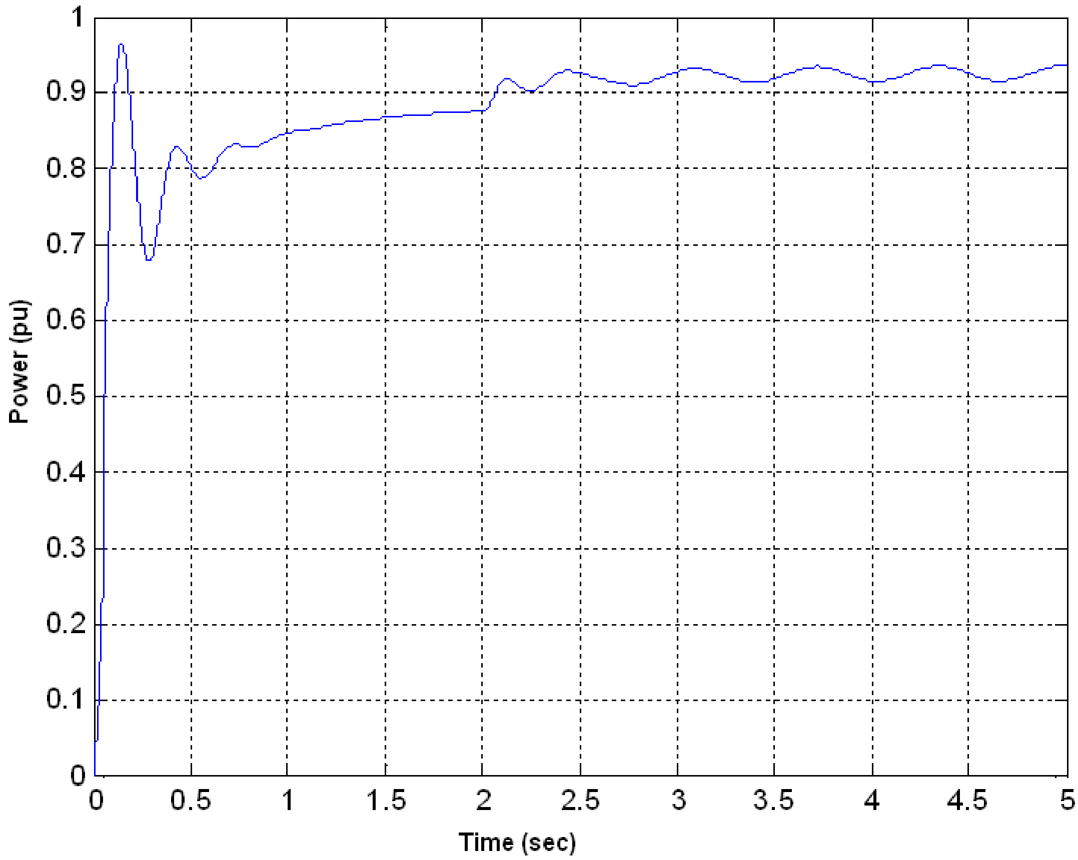


Figure 7.9 Gas turbine response to cyclic loading

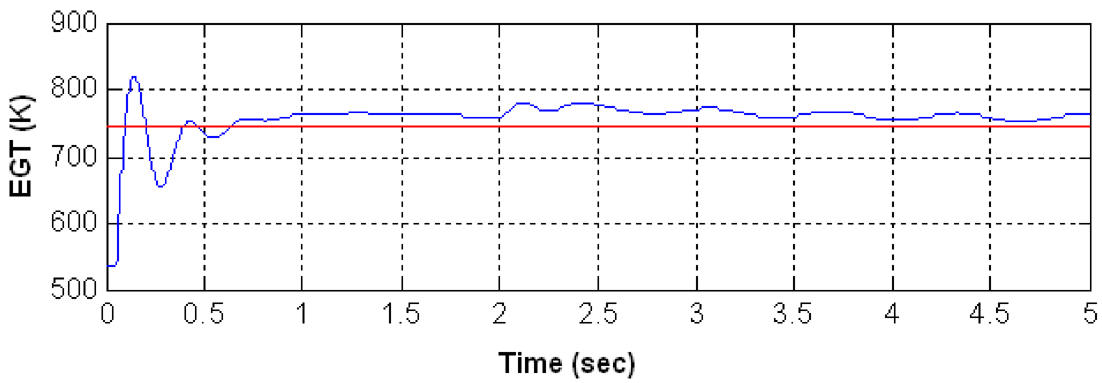


Figure 7.10 Variation of exhaust gas temperature to cyclic loading.

7.4. Heat exchanger model results

The results from the heat exchanger models are presented in the next set of figures and tables. Table 7-2 shows the results from a counterflow and parallelflow intercooler and Table 7-3 shows the results of a recuperator. The intercooler efficiency for the counterflow arrangement is 0.80 and the parallelflow arrangement is 0.53. These results were able to be achieved with a parallelflow design of recuperator only in the case of low heat exchanger efficiency. However the recuperator did not fail to produce results when counterflow pattern of exchangers was selected. This is something similar which has been noticed by operators of such machine. The reason could be that the two fluids are in the same state and very close in their temperatures that heat transfer may occur in an oscillating manner when the scheme of the exchanger is a parallelflow.

Table 7-2 Intercooler temperature distribution

Counterflow			Parallelflow		
T_H (K)	T_C (K)	ΔT (K)	T_H (K)	T_C (K)	ΔT (K)
408.5	309.3	99.2	408.5	288.2	120.3
391.7	305.6	86.1	398.7	302.9	95.8
377.1	302.4	74.7	390.0	303.6	86.4
364.5	299.6	64.9	382.0	304.2	77.8
353.5	297.2	56.3	374.9	304.8	70.1
344.0	295.1	48.9	368.5	305.3	63.2
335.7	293.3	42.4	362.7	305.7	57.0
328.5	291.7	36.8	357.5	306.1	51.3
322.3	290.4	31.9	352.8	306.5	46.3
316.9	289.2	27.7	348.5	306.8	41.7
312.2	288.2	24.1	344.7	307.1	37.6

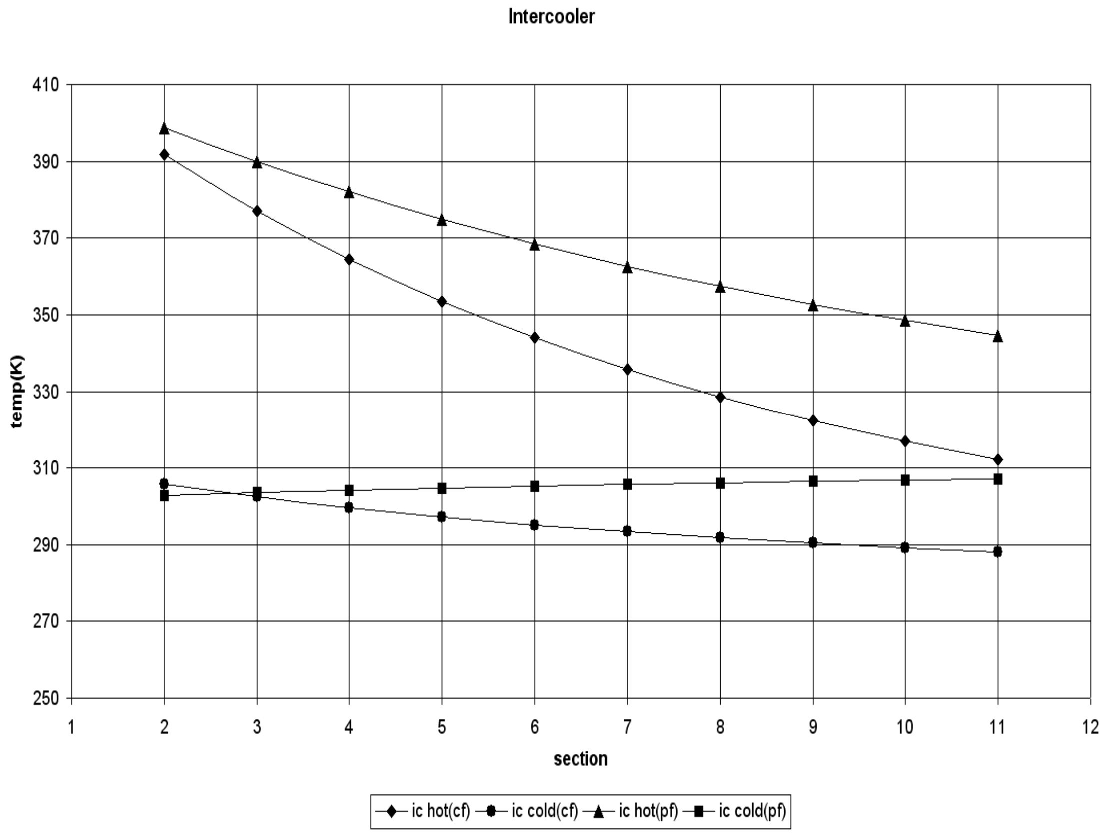


Figure 7.11 Temperature profile of intercooler (counterflow & parallelflow)

Table 7-3 Recuperator temperature distribution

Counterflow			Parallelflow		
T_H (K)	T_C (K)	ΔT (K)	T_H (K)	T_C (K)	ΔT (K)
735.1	691.4	43.7	735.1	691.4	43.7
725.0	678.5	46.5	721.3	679.6	41.7
714.3	664.7	49.6	707.0	675.9	31.1
702.9	650.1	52.8	696.3	673.1	23.2
690.7	634.5	56.3	688.3	671.0	17.3
677.8	617.8	60.0	682.4	669.5	12.9
664.0	600.1	63.9	678.0	668.3	9.6
649.3	581.2	68.1	674.6	667.4	7.2
633.6	561.0	72.5	672.2	666.8	5.4
616.9	539.6	77.3	670.3	666.3	4.0
599.1	516.7	82.4	669.0	666.0	3.0

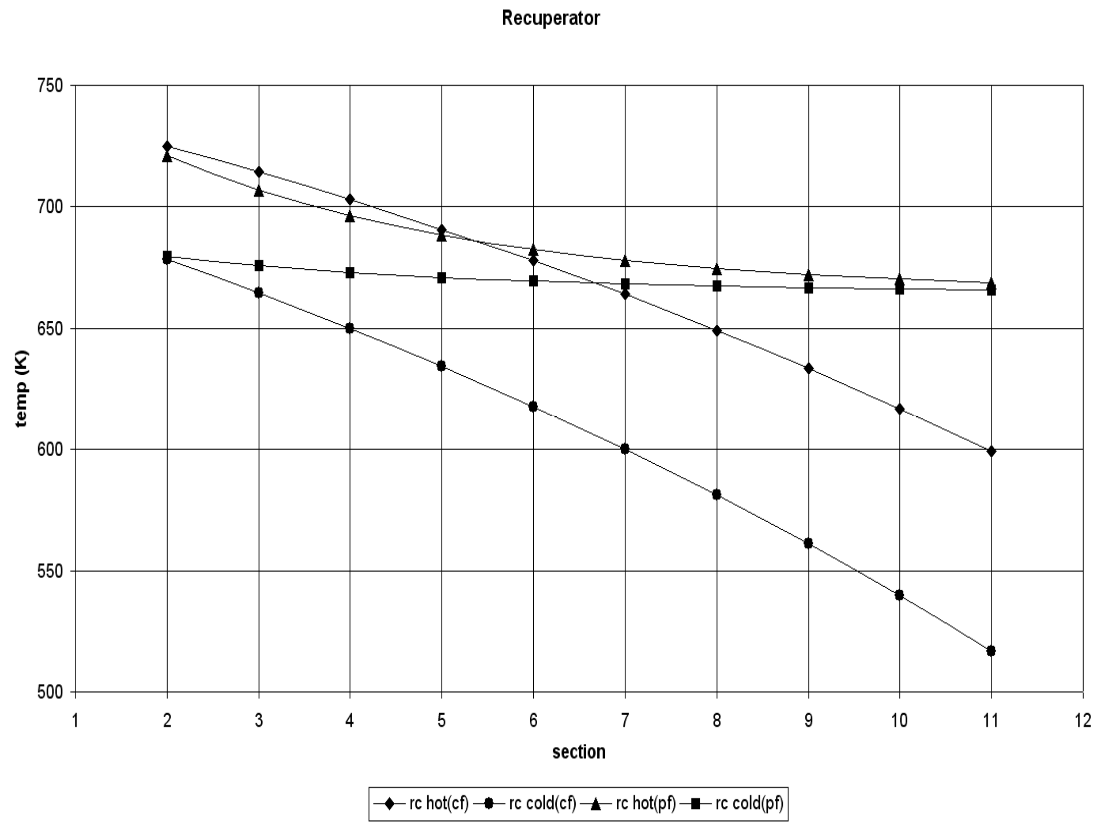


Figure 7.12 Temperature profile of recuperator (counterflow & parallelflow)

From Figure 7.11 and Figure 7.12 it can be noticed that the counterflow type of arrangement suit better for the intercooler and the recuperator. It was noticed that the thermal efficiency of the cycle is highest for a counterflow arrangement when the effectiveness of either exchangers is assumed to be similar. Maximum possible

Table 7-4 shows the power and the cycle thermal efficiency of an intercooled-recuperated engine for different configurations of heat exchanger and flow arrangement.

Table 7-4 ICR Power and cycle efficiency with maximum exchanger effectiveness

Heat exchanger type	ε	Power (kW)	η
Counterflow IC and RC	0.99 & 0.99	24546	0.458
Parallel flow IC and RC	0.53 & 0.53	22257	0.391
Counterflow IC and Parallelflow RC	0.99 & 0.53	24691	0.416
Parallelflow IC and Counterflow RC	0.81 & 0.99	23613	0.443

8. CONCLUSIONS AND RECOMMENDATIONS

Today gas turbine technology is well developed and the initial literature survey conducted confirmed the same. It also helped to realise the scope of its applicability and understand the current market and scope for further development. For the marine industry, which is one of the new customers, the benefit of the gas turbine engine coupled with their existing full electric propulsion system has an added advantage over traditional schemes. In this context the research carried out by the author has value not only for the industry but also for systems developers.

All along the project duration aims were set, objectives decided and results achieved. Some results were ordinary while some significant. The Gas Turbine-Electrical Network interface that was created was a major milestone for system simulation. With a high-fidelity model that can simulate the behaviour for an electro-mechanical propulsion system will help in reducing initial risk and development costs. With this tool system modelling is more accurate to the whole system than deriving implicit relationships between mathematical models. The adaptation of gas turbine design models to a transient model was successful and two configurations of engine dynamic parametric studies were accomplished. This coupled together with the previous study provided a good insight into the behaviour of complex systems. The control systems were developed for the two-shaft gas turbine and parametric studies were performed using all tools. Finally, the steady state and sizing model of the heat exchanger had been accomplished with the transient model close to completion.

8.1. Conclusions

The results of the gas turbine design point showed good behaviour in terms of the theory of gas turbine performance. This behaviour was analysed for the effect of bleeds. It was seen that the single shaft gas turbine power increased higher with a decrease in bleed flow. Therefore it is best to have bleeds and control can be developed if required based on them. This control will be cheaper than variable geometry blades.

The high fidelity model developed showed good behaviour in the response trends with the electrical networks. The model is robust to various load fluctuations has increased capabilities of dual control strategies such that if one fails the other could be used to handle. Because there was no experimental data available the models were validated through various technical support persons from industry in the form of personal communication.

The heat exchanger models for the thermal design and sizing were created. They showed good convergence when the system of fluids was generally in the opposite direction i.e. counterflow. The schemes converged for parallelflow fluids when the efficiency was kept low. Sizing of the heat exchanger was performed, and it may be concluded that any transient simulation performed should be with some accurate data of the exchanger.

8.2. Recommendations

From the research undertaken to simulate gas turbine steady state and transient performance the following recommendations can be made

- There is plenty of scope for the applications of the gas turbines. For this novel configurations have to be researched and developed. Simulation studies can be performed for these gas turbines similar to the one carried out in the case of the two gas turbines selected. This if compared to other research indicates good convergence then full models could be developed for the engines.
- A few novel cycles that should be researched are the steam injected gas turbines. Though the machinery required to purify the water is not cheap, the source is available in plenty for marine vessels. If an analysis is made to understand the minimum grade of water required

for such engines then a together with a techno-economic risk analysis the option may provide an efficient source of power for marine vessels.

- The prime mover-electrical network can be updated with new engine configurations for types like ICR. It has been analysed that the ICR has better part load efficiency..
- From a completely different point of view, gas turbine simulation platforms for academic purposes are being developed by many universities. Cranfield University has its own performance deck 'TURBOMATCH' and 'PYTHIA' which are extremely sophisticated. The development of this code to provide a GUI may prove to be a useful starting point for such a program.
- The steady state models that were used for the study were based on constant mass flow. For the aim of this project where the main study was to have a transient model to work with an electrical network, the transients mainly analysed were of high frequencies (shaft transients). Therefore a model with the intercomponent volume method should be added and then a comparative study should be undertaken to estimate the behaviour of the system as whole.
- The single shaft transient model need further development of control laws, which then should be combined into the system as a backup when simulating network interferences.
- Finally the author recommends continuing the work on heat exchanger modelling and a complete set of gas turbine and heat exchanger models can then be used in the high fidelity model.

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APPENDIX A

Program for linking gas turbine and electrical network.

- Fortran file to convert any available source code to a direct link library
- Sample data file supporting the fortran file (Two-shaft and single shaft gas turbine)

```
      SUBROUTINE SIZES(SIZE)
C     .. Array arguments ..
      INTEGER*4        SIZE(*)
C     .. Parameters ..
      INTEGER*4       NSIZES
      PARAMETER       (NSIZES=6)

      SIZE(1) = 0
      SIZE(2) = 1
      SIZE(3) = 3
      SIZE(4) = 2
      SIZE(5) = 0
      SIZE(6) = 1

      RETURN
      END

C
C=====================================================
C
C     Function:  OUTPUT
C
C     Abstract:
C       Perform output calculations for continuous
C       signals.
C
C=====================================================
C     .. Parameters ..
      SUBROUTINE OUTPUT(T, X, U, Y)

      REAL*8           T(*)
      REAL*8           X(*), U(*), Y(*)
C --- Nothing to do.
      RETURN
      END

C
C=====================================================
C
C     Stubs for unused functions.
C
C=====================================================

      SUBROUTINE INITCOND(X0)
      IMPLICIT NONE
      DOUBLE PRECISION SV,BF,BFT,FCVAL,FFLOW,CWRK,RCC,TWRK,
      2AWRKS1,AWRKS2,RPM1,RPM2,CNRT,TCNRT,TPNRT,BETAC,VSVC,
      3DP150,DP31,DP445,DP495,PAMB,TAMB,FMNO,ALT,DTISA,ETACC,
      4BETACT,BETAPT,TCPRAT,TPPRAT,ETATC,ETAPT,VNGVCT,VNGVPT,
      5TCWRK,TPWRK,TCNDFL,TPNDFL,TOL,GGSWRK,UWRK,ETATH,OPR,FUELNC,
```

```
     6RPM1NC,ETAC, egpwr, pwcnt, A1,DPFLOW,REQWRK, tetod,inipwr
      INTEGER NYMPH, iexe
      REAL*8           X0(*)
      COMMON/ODP/
SV(12,200),BF(20),BFT(20),FCVAL,FFLOW,CWRK,RCC,TCWRK,
      2AWRKS1,AWRKS2,RPM1,RPM2,CNRT,TCNRT,TPNRT,BETAC,VSVC,ETACC,TPWRK
,
      3DP150,DP31,DP445,DP495,PAMB,TAMB,FMNO,ALT,DTISA,VNGVCT,VNGVPT,
      4BETACT,BETAPT,TCPRAT,TPPRAT,ETATC,ETAPT,TCNDFL,TPNDFL,TOL,GGSWR
K
     5,UWRK,ETATH,OPR,ETAC,FUELNC(50),RPM1NC(50),
     6 egpwr, pwcnt, A1, tetod
      COMMON/MASSFL/DPFLOW,REQWRK
      common/initsspt/inipwr

C---------  FIRST, DO THE DP CALCULATIONS ----------------
      OPEN (8, FILE='ZOUTDGN8.DAT')
      OPEN (10, FILE='ZOUTDGN10.DAT')
      OPEN (11,ACTION='READ', FILE='YDPt11TS.DAT')
      OPEN (12, FILE='ZOUTGEN12.DAT')
c     OPEN (14, FILE='ZOUTPLT14.DAT')
c     WRITE (14,8100)
c     open (98,file='sspt.txt')
      open(90,file='ZRESULT_TS.TXT')
      write(90,6000)

      DPFLOW = 80.0
      call dpmflw             !DP mass flow calculated

      CALL DPCALC
c     OPEN (13,ACTION='READ', FILE='YODP13.DAT')
c     call odcarpet
c     write (98,*) uwrk,fflow,sv(7,14),betac
      call transient_initialization       !Starting point initialised

      close(8)
      close(10)
      close(11)
      close(12)
c     close(13)
c     close(14)
c     close(98)

      return

6000  FORMAT('  time    PAMB-kPa   TAMB-K     ALT-M    P2-kPa   ',
      1' NDFlow2   M_FLOW    NDFlow3    P3-kPa     T3-K      T4-K   ',
      2' P4-kPa    T41-K   NDFlow41   P41-kPa     T43-K   P43-kPa   ',
      3' T45-K     P45-kPa  NDFlow45    T47-K    P47-kPa   CWRK-kW  ',
      4'    UWRK-kW    acFuel    ETATH    GGRPM     PTRPM',
      5'     vsvc    betac   etac    etatc   etapt')

      end

      SUBROUTINE DERIVS(T, X, U, DX)
      REAL*8           T, X(*), U(*), DX(*)
C --- Nothing to do.
      RETURN
      END
```

```
      SUBROUTINE DSTATES(T, X, U, XNEW)
      REAL*8           T, X(*), U(*), XNEW(*)
C --- Nothing to do.
      RETURN
      END

      SUBROUTINE DOUTPUT(T, X, U, Y)
      IMPLICIT NONE
      REAL*8           T, X(*), U(*), Y(*)

      DOUBLE PRECISION SV,BF,BFT,FCVAL,FFLOW,CWRK,RCC,TCWRK,
      2AWRKS1,AWRKS2,RPM1,RPM2,CNRT,TCNRT,TPNRT,BETAC,VSVC,ETACC,TPWRK
,
      3DP150,DP31,DP445,DP495,PAMB,TAMB,FMNO,ALT,DTISA,VNGVCT,VNGVPT,
      4BETACT,BETAPT,TCPRAT,TPPRAT,ETATC,ETAPT,TCNDFL,TPNDFL,TOL,
      5WERR,GGSWRK,AST,A1,UWRK,ETATH,OPR,FUELNC,RPM1NC,ETAC, werr2
      6 ,egpwr, pwcnt, tetod
      INTEGER LOOPW,NCALC,NYMPH,J,I, looppt
      COMMON/ODP/
SV(12,200),BF(20),BFT(20),FCVAL,FFLOW,CWRK,RCC,TCWRK,
      2AWRKS1,AWRKS2,RPM1,RPM2,CNRT,TCNRT,TPNRT,BETAC,VSVC,ETACC,TPWRK
,
      3DP150,DP31,DP445,DP495,PAMB,TAMB,FMNO,ALT,DTISA,VNGVCT,VNGVPT,
      4BETACT,BETAPT,TCPRAT,TPPRAT,ETATC,ETAPT,TCNDFL,TPNDFL,TOL,GGSWR
K
     5,UWRK,ETATH,OPR,ETAC,FUELNC(50),RPM1NC(50),
     6 egpwr, pwcnt, A1, tetod

      double precision fuelfl, rpmpt, powerout

      double precision  inertia1, inertia2, time, dtime, cnst, pi,
     2  drpm1, drpm2
      integer icount, igenpower

      common/transientr/ inertia1, inertia2, time, dtime, cnst,
      2 drpm1, drpm2, pi
      common/transienti/ icount, igenpower

c     OPEN(UNIT=98,FILE='tempin.txt')
c     WRITE(98,*)u(1),u(2)
c     CLOSE(98)
      fflow = u(1)
      rpm2 = u(2)
      CALL trptstep(fflow, rpm2, powerout)
      y(1) = powerout
      y(2) = fflow
      y(3) = sv(7,31)
c     open (99,file='tempout.txt')
c     write (99,*) y(1),y(2),fflow,rpm2
c     close (99)
C --- Nothing to do.

      RETURN
      END

      SUBROUTINE TSAMPL(T, X, U, TS, OFFSET)
      IMPLICIT NONE
      REAL*8           T,TS,OFFSET,X(*),U(*)
```

```
      double precision fuelfl, rpmpt, powerout

      double precision  inertia1, inertia2, time, dtime, cnst, pi,
     2  drpm1, drpm2
      integer icount, igenpower

      common/transientr/ inertia1, inertia2, time, dtime, cnst,
      2 drpm1, drpm2, pi
      common/transienti/ icount, igenpower
      ts = dtime
      offset = 0
      RETURN
      END

      SUBROUTINE SINGUL(T, X, U, SING)
      REAL*8           T, X(*), U(*), SING(*)
C --- Nothing to do.
      RETURN
      END
```

Data file of program

```
15  1400                                        !L1    DESIGN PT DATA: -RC-TET
0.88  0.999 0.88  0.89 9800   3600              !L2      DESIGN  PT  DATA:  flow-ETAC-ETACC-ETATC-ETATP-RPM1-
RPM2
2  288  101.3   000  0.0   0                    !L3    NAMB  Tamb, Pamb, ALT, MN, DTISA
1     0     9     0.00001                       !L4    NINT, NYMPH,NBF,TOL
1.0  1.0   0                                    !L5    COMP DESIGN POINT AT N= ; BETA = ; A= ;
1.0  1.0   0                                    !L6    C-TURB DESIGN POINT AT N= ; BETA = ; A= ;
1.0 1.0   0                                     !L6    P-TURB DESIGN POINT AT N= ; BETA = ; A= ;
0.005 0.01 0.003 0.01 0.002 0.01 0.01 0.10 0.04 !L7    9 Bleed Flows, See Below
0.01  0.05  0.01 0.01   0.01  0.0               !L8    6 Pressure losses
43100   400  200                                !L9   fuel  Cal  Val  (MJ/kg)  +  Power  extractions  from  each
                                                shaft in kW
0.2 0.5 0.1 0.3 0.5 0.09 0.01                   !L10   Mach Numbers in Engine
150. 100.                                       !L11   Inertias of shafts gas generator and power turbine-
                                                electrical generator
1.e-3   20.e3   20.e3                           !L12    Time step for transient calculations, DP power, OD
                                                power for transient initialization.
Design Point Data file:  When starting, only lines 1-3 need to be specified.
                   The others can be used as defaults, Copied from this file

Line1-      OPR and COT

Line2-  Flow,  Comp + Turb Isentropic Efficiencies + Rot Speed of Two Shafts

Line3-  Chice of Namb determines the DP ambient conditions.
      NAMB = 1:   Tamb and Pamb are taken into account
      NAMB = 2:   ALT (in m) and DTISA are taken into account

Line4-NINT:   Interpolation type: 1-Linear  and 2-Parabollic
      NYMPH:  Diagnostic Prints - 0 No diagnostic prints
      NBF:  No of Bleed Flows (This must agree with line 7)

Line5-  This point on the Compressor map will be scaled with DP data
```

```
Line8- DPs DP inlet,  DP comb. DP i/turb, DP exhaust, DP other 2

Line9-      fuel Cal Val  (MJ/kg)  Power extractions from each shaft in kW

Line10-     Various Mach Numbers in engine

Line11- The shaft inertias for the gas-generator and for the power turbine-electrical
        generator shaft in (kg m^2)

Line12- Time step for transient calculations
```

APPENDIX B

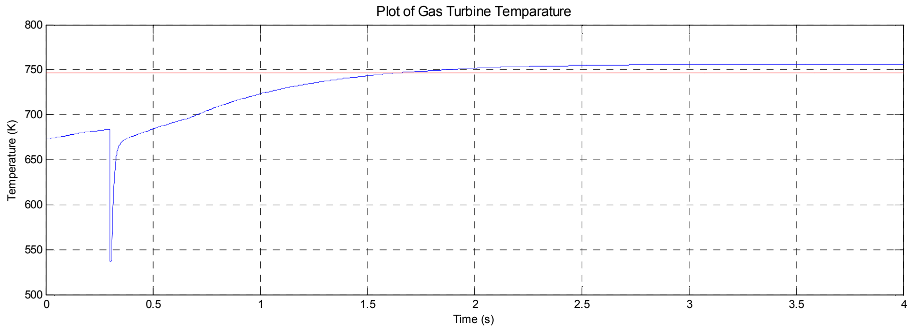
Comparison of case studies

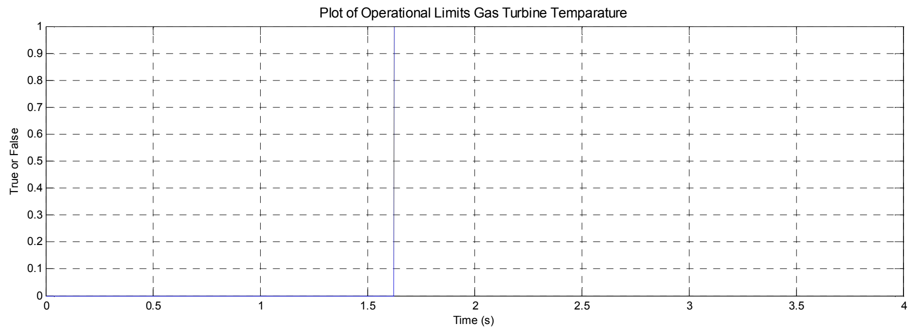
The simulation took roughly 200-350s to run (a 3s simulation).

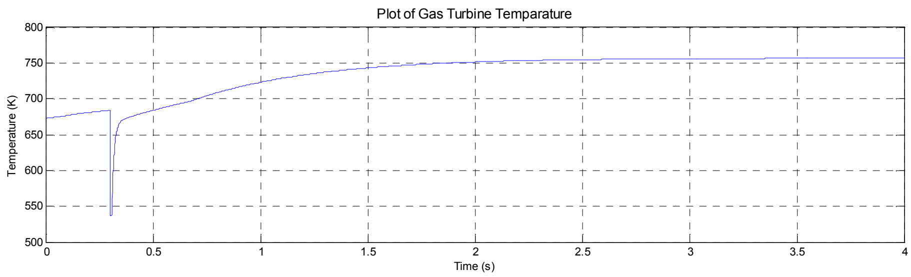
Case Study 1

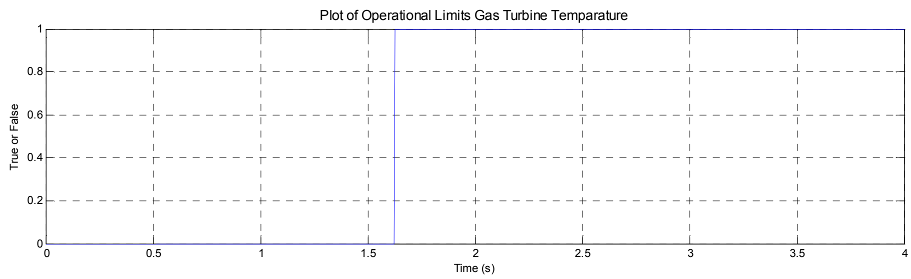
This first case has been aimed towards the following initial condition:

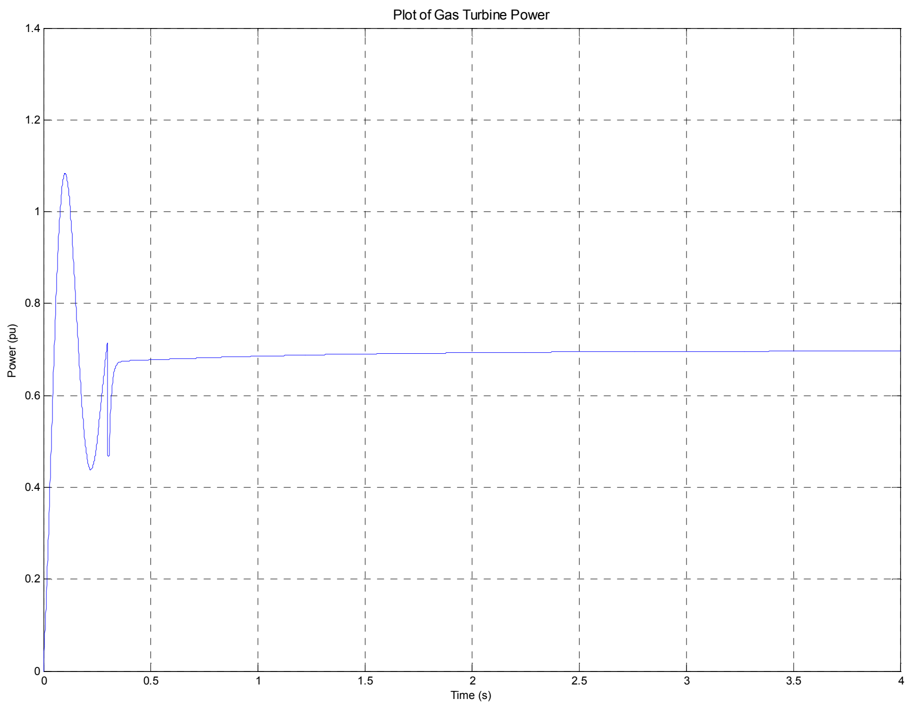
- Initial aimed power: 16MW (as the fixed resistive load is 16MW); corresponding to 1.1086kg/s of fuel (initial fuel valve condition)
- Power base is according to the generator; 23.33MW

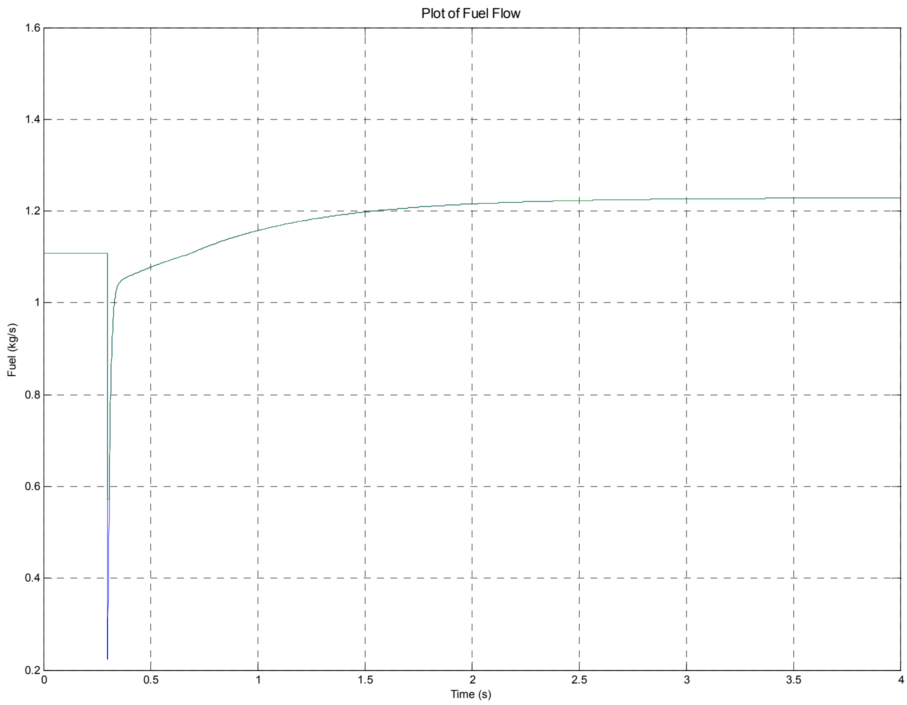






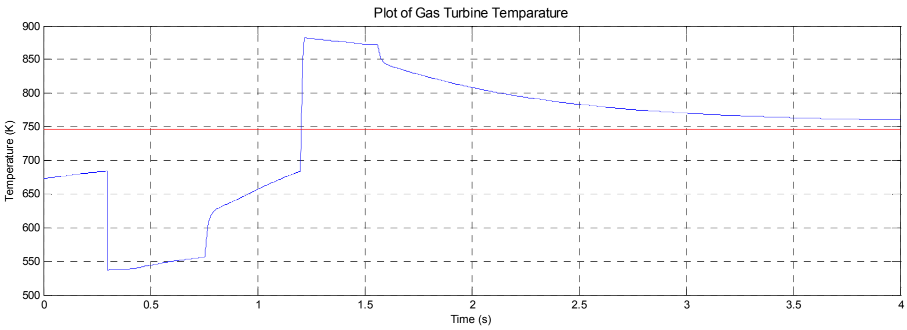


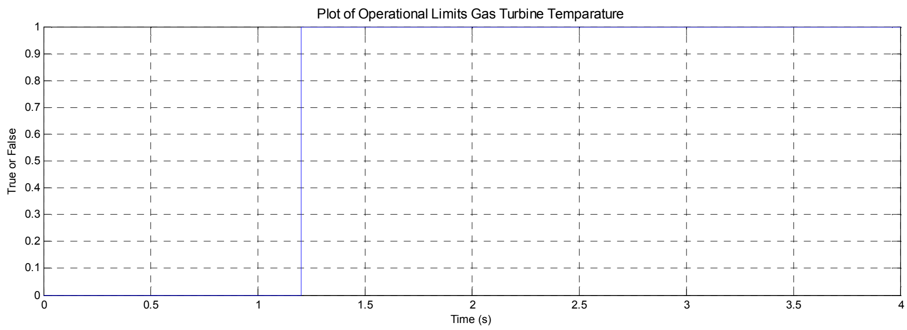


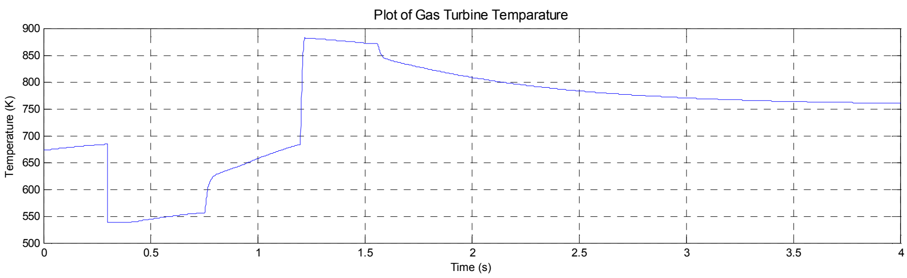


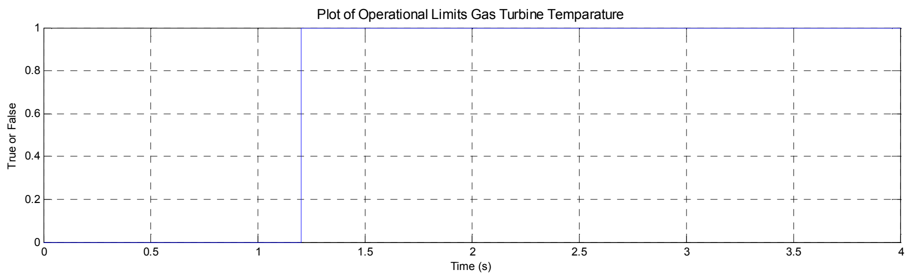
Case study 2

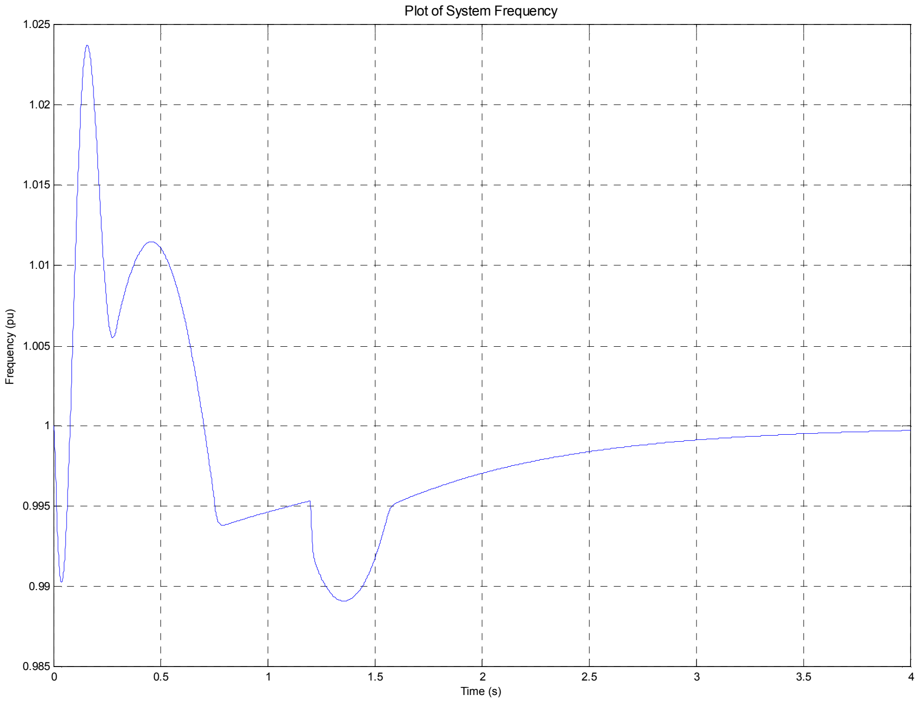
Case study 2is the same as case study 1 apart from the fact that the 16MW fixed resistive load has split up in two blocks. 10MW is connected from the very beginning, however 6MW will be switched in at t=1.2s. This could be considered equal to the scenario where a 6MW motor is used.

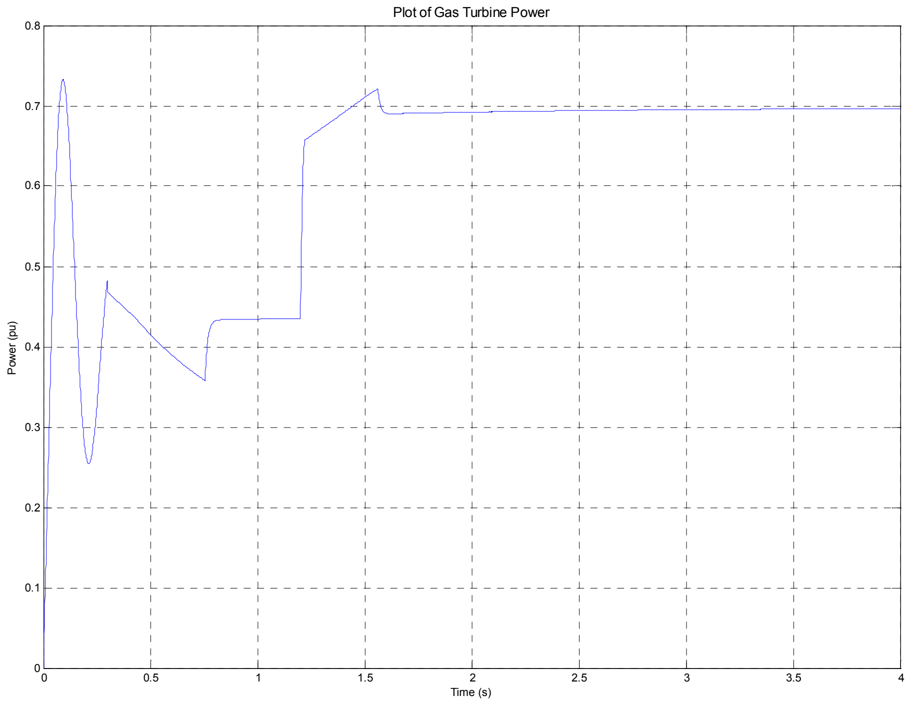


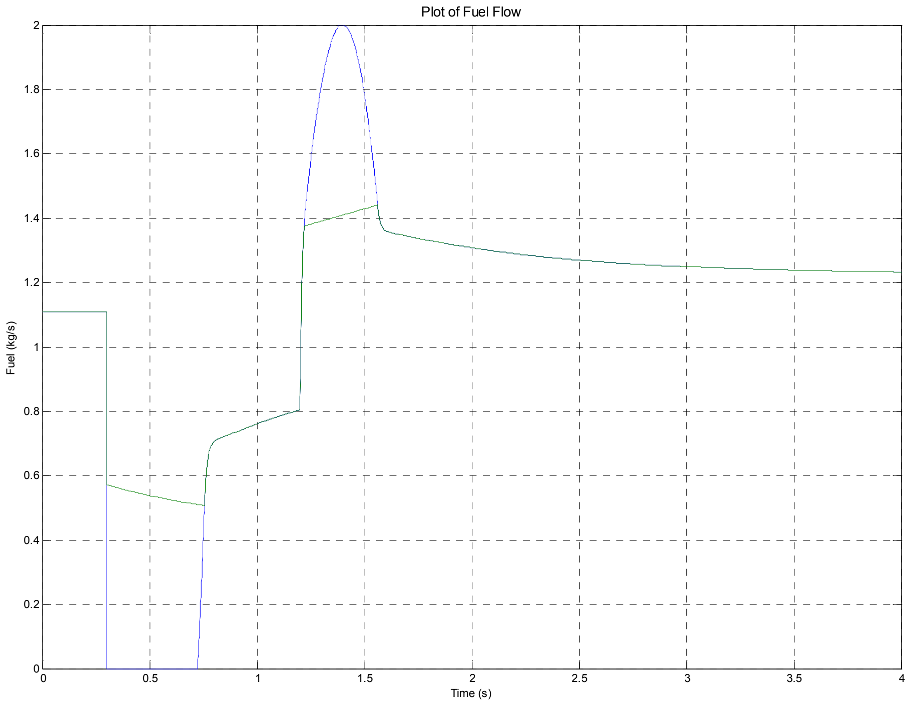


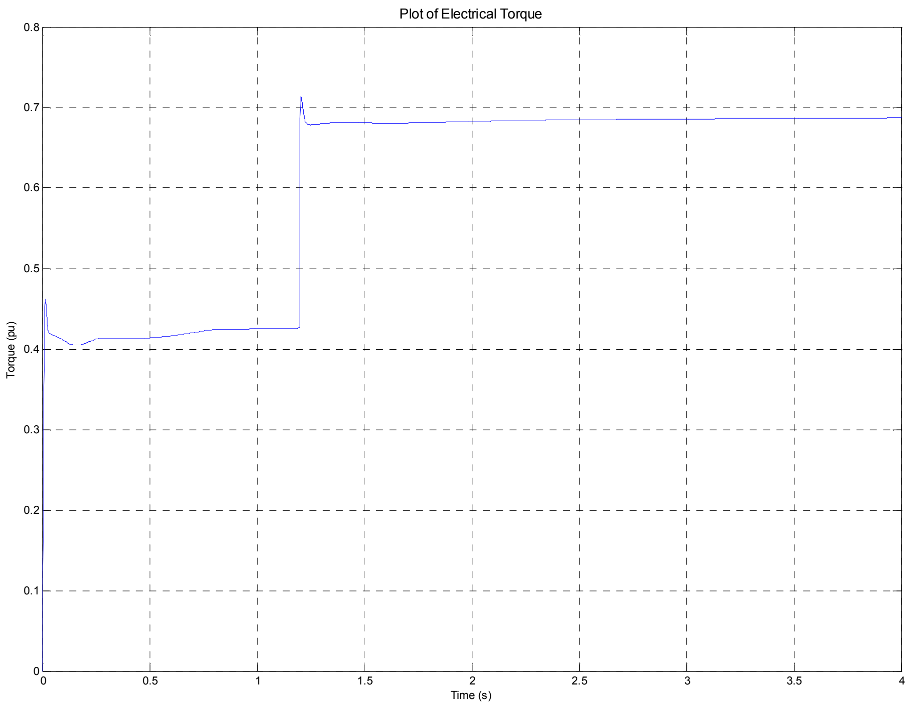





Plot of System Frequency
Frequency (pu)
Time (s)
1.025
1.02
1.015
1.01
1.005
1
0.995
0.99
0.985
0
0.5
1
1.5
2
2.5
3
3.5
4


Plot of Gas Turbine Power
Power (pu)
Time (s)
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
0
0.5
1
1.5
2
2.5
3
3.5
4

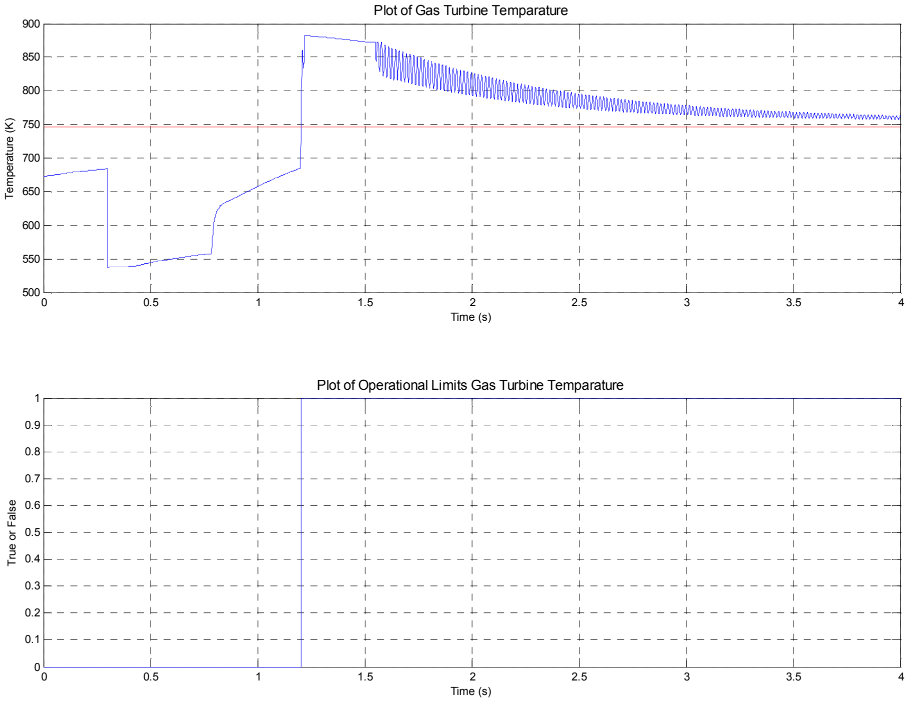


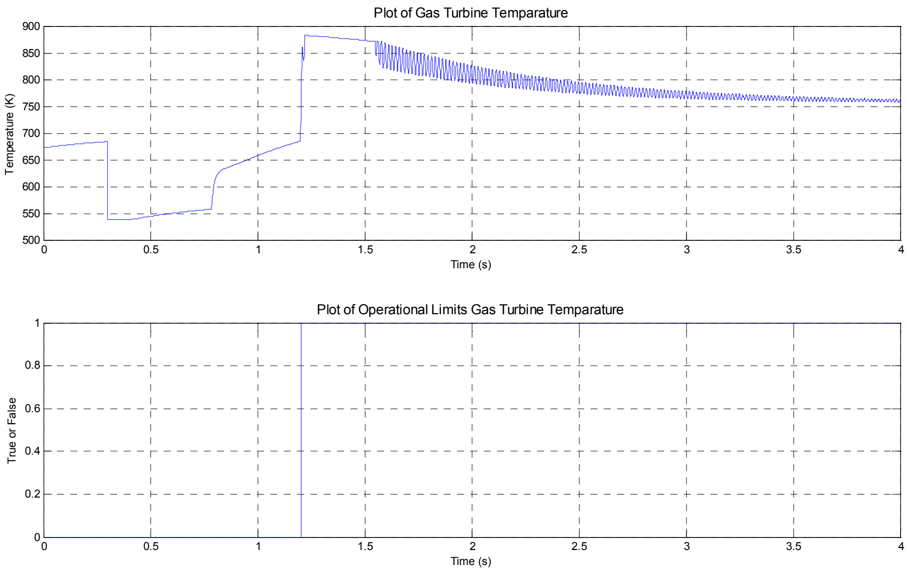


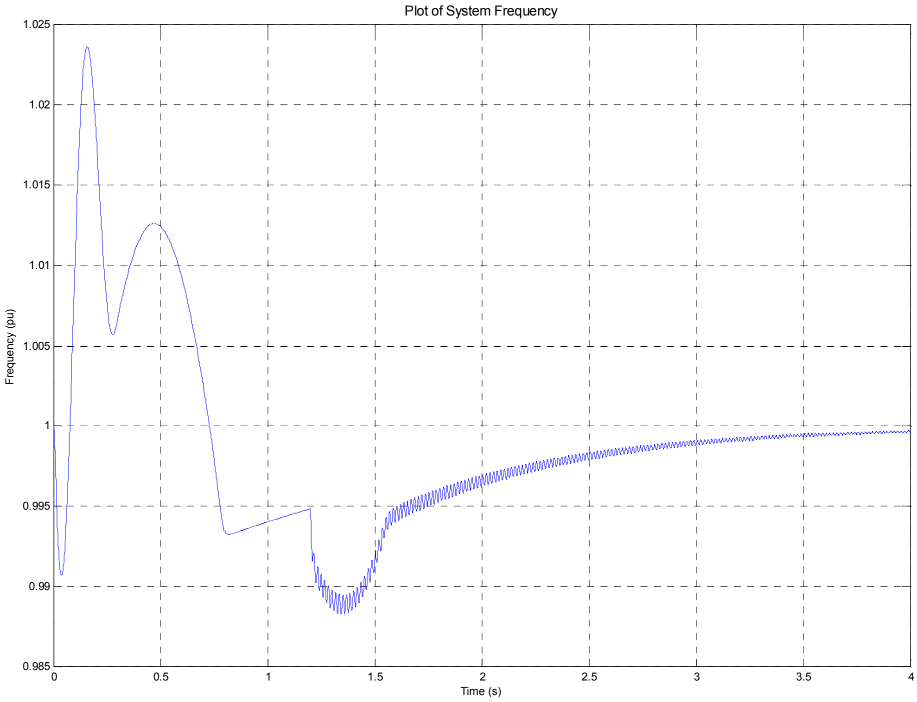
Case Study 3

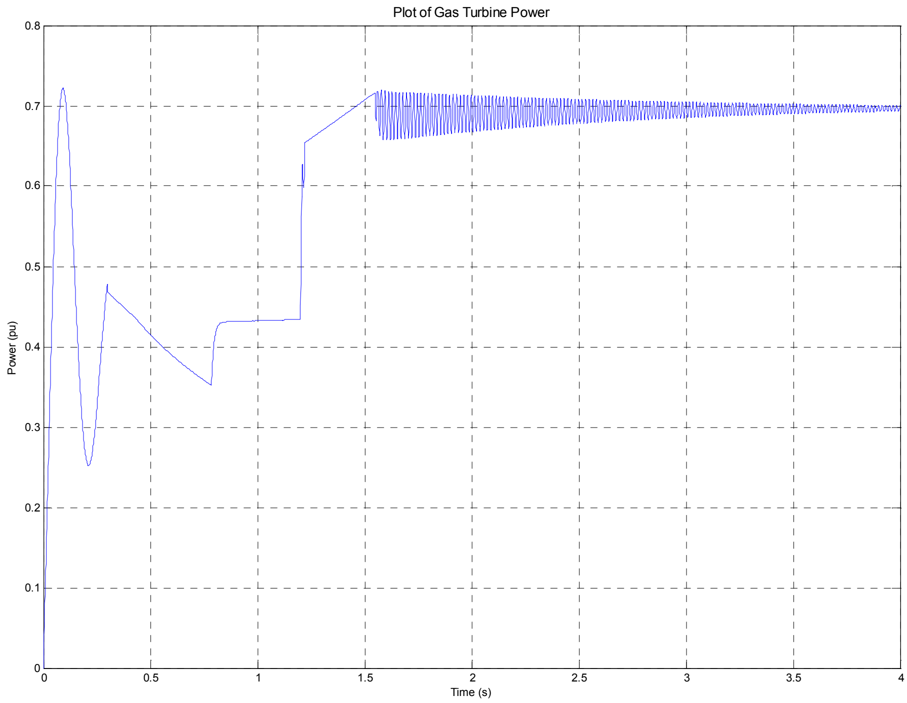
Case study 3 is the same is case study 2 apart from the fact that for both fixed loads a power factor of 0.8 has been introduced (parallel resistive-inductive). The frequency of the oscillation (that starts at t ≈1.55s, time that the required fuel flow is the as the actual fuel flow again) once the 6MW load has been switched in is 60Hz.

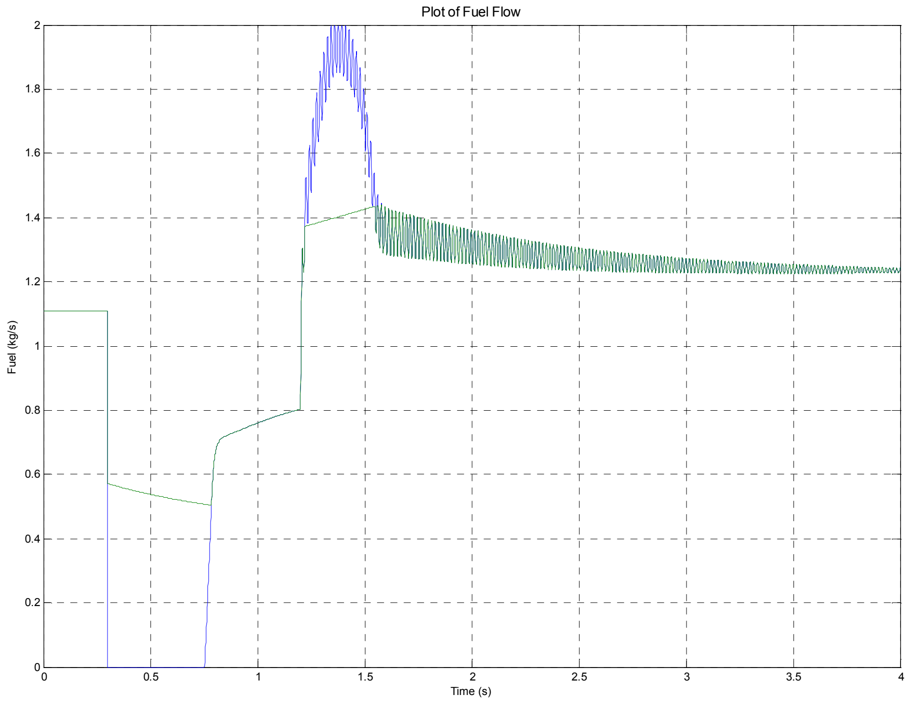
The same oscillation of 60Hz can be found on the shaft torque between the gas turbine and the generator.



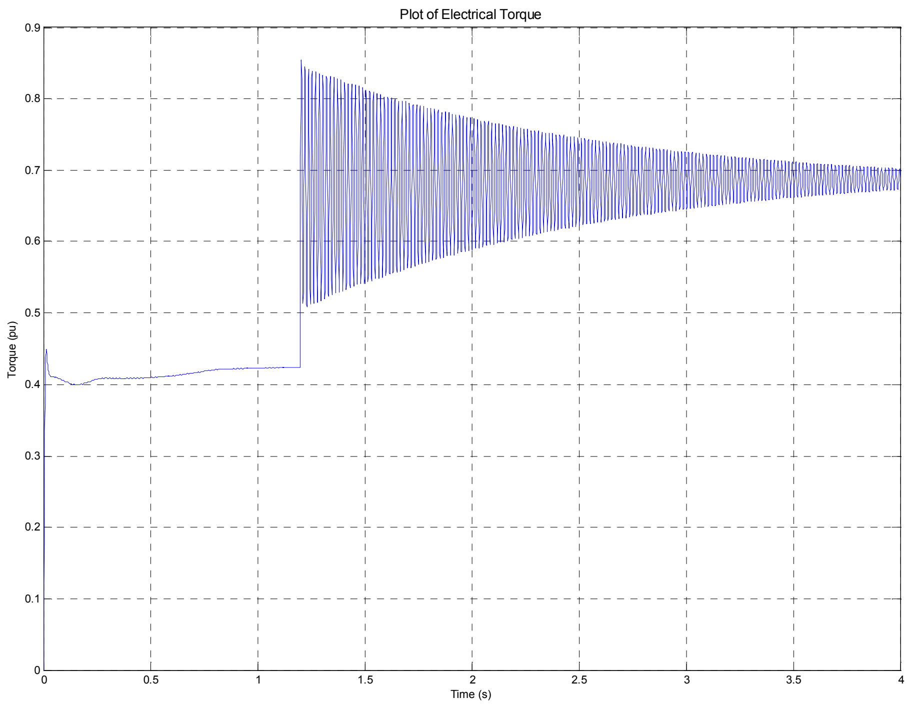



Plot of System Frequency
Frequency (pu)
Time (s)
1.025
1.02
1.015
1.01
1.005
1
0.995
0.99
0.985
0
0.5
1
1.5
2
2.5
3
3.5
4


Plot of Gas Turbine Power
Power (pu)
Time (s)
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
0
0.5
1
1.5
2
2.5
3
3.5
4



Plot of Fuel Flow
Fuel (kg/s)
Time (s)
0
0.2
0.4
0.6
0.8
1
1.2
1.4
1.6
1.8
2
0.5
1.5
2.5
3
3.5
4



Plot of Electrical Torque
Torque (pu)
Time (s)
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1
1.5
2
2.5
3
3.5
4

APPENDIX C

Fortran file for the thermal design of heat exchangers.

- Effectiveness-NTU method (part 1)
- Sizing scheme 1 based on part 1
- Sizing scheme 2 based on part 1
- Sample data file of program

```
      program intcol
      implicit none
      double precision tci,thi,tco,tho,ua
      call ic_dp

      call ic_sizing

      stop
      end program intcol

      subroutine ic_dp
      implicit none
      double                                                   precision
m1,m2,cp1,cp2,ua,tci,thi,tco,tho,c,expo,c1,c2,pi,
      1po,ntu,eff,q,ic_eff,cmin,gamma1,gamma2,mach1,dtime,tol,beta,
      2R,feff,vel2,vfac,deltap
      integer i,itype,itempr,imethod
      real th(0:10),tc(0:10),deltat(0:10)
      common/param/m1,cp1,gamma1,mach1,thi,tho,pi,m2,cp2,gamma2,tci,tc
o,
      1deltap,dtime,tol,beta,vel2,R,feff,vfac,ua,itype,itempr,imethod

      open(50,action='read',file='heat_exchanger.dat')
c     open(50,action='read',file='heat_exchanger_orig.dat')
      open(51,file='hx_results.dat')
      read(50,*) m1,cp1,gamma1,mach1,thi,tho,pi
      read(50,*) m2,cp2,gamma2,tci
      read(50,*) itype,itempr,imethod
      read(50,*) deltap,dtime,tol
      read(50,*) beta,vel2,R,feff,vfac,ic_eff

      c1 = m1*cp1
      c2 = m2*cp2
      if (c1.gt.c2) then
      c = c2/c1
      cmin = c2
      else
      c=c1/c2
      cmin = c1
      endif
c     c=1.0
      if (itempr.eq.0) then             !itempr  =  0,  select  tho  as
input
      q = c1*(thi-tho)                  !q=Qhot here
      tci = thi-q/(ic_eff*cmin)
      write(51,9000)
      elseif (itempr.ne.0) then         !itempr  =  non-zero,  select
tci as input
      q = ic_eff*cmin*(thi-tci)         !q=EFFECTIVENSS x Qmax
```

```
      tho = thi-q/c1
      write(51,9010)
      endif
      tco = tci+q/c2

      if (itype.eq.0) then
      write(51,9020)
      ntu = log((1-c*ic_eff)/(1-ic_eff))/(1-c)
      elseif (itype.ne.0) then
      write(51,9025)
      ntu = -log(1-ic_eff*(1+c))/(1+c)
      elseif      (c.eq.1.0) then
      ntu = ic_eff/(1-ic_eff)
      endif
c     print*,'c1,c2,po,c,itype,ntu'
c     print*,c1,c2,po,c,itype,ntu
      ua = ntu*cmin

      if (itype.eq.0) then
      expo = -(ua/c1)*(1-(c1/c2))
      else
      expo = -(ua/c1)*(1+(c1/c2))
      endif

c******temperature distribution calculation******
c***********************************************
      if (itype.eq.0) then                !counter-flow IC
      write(51,9030)
      tco = 1/(exp(expo)-c2/c1)*(thi*(exp(expo)-1)+(1-(c2/c1))*tci)
      th(0) = thi
      tc(0) = tco
      deltat(0) = th(0) - tc(0)
      do i=0,10
      th(i)           =           (1/(1-c1/c2))*(tco-(c1/c2)*thi+(thi-
tco)*exp(expo*i*0.1))
      deltat(i) = (thi-tco)* exp(expo*i*0.1)
      tc(i) = th(i)-deltat(i)
      write(51,9040) th(i),tc(i),deltat(i)
      enddo
      elseif (itype.ne.0) then            !parallel flow IC
      th(0)=thi
      tc(0)=tci
      deltat(0)=th(0)-tc(0)
      write(51,9030)
      write(51,9040) th(i),tc(i),deltat(i)
      do i=1,10
      th(i)= (1/(1+c1/c2))*(tci+(c1/c2)*thi+(thi-tci)*exp(expo*i*0.1))
      deltat(i) = (thi-tco)* exp(expo*i*0.1)
      tc(i) = th(i)-deltat(i)
      write(51,9040) th(i),tc(i),deltat(i)
      enddo
      endif

      print*,'thi,tci'
      print*,thi,tci
      print*,'tho,tco'
      print*,tho,tco
      print*,''
```

```
      return
9000  format('Tho - Hot fluid outlet temperature selected as input')
9010  format('Tci - Cold fluid inlet temperature selected as input')
9020  format('Heat Exchanger type : Counter-flow')
9025  format('Heat Exchanger type : Parallel-flow')
9030  format('   th(i)       tc(i)       deltat(i)')
9040  format(3f12.5)
      end

      subroutine ic_sizing
c  This  subroutine  does  the  calculations  for  the  sizing  of  the
intercooler.
c  The  values  required  by  the  subroutine  are  temperatures  of  the
fluids, UA.
c The outputs are the geometric dimensions and various non-dimesional
numbers,
c which define the flow characteristics
      implicit none
      double precision m1,m2,cp1,cp2,U,Area,UA,Tci,Thi,Tho,Tco,Y,L,w1,
     1w2,h1,h2,Mach1,R,p1,T1,T2,t1stat,Pi,Po,vel1,vel2,ht1,ht2,
     2ro1,ro2,D1,D2,A1,A2,dtime,deltax,mu1,mu2,k1,k2,Re1,Re2,Pr1,Pr2,
     3Nu1,Nu2,f1,f2,feff,vfac,A,B,C_0_water,tol,deltap,v1,p1stat,T0,
     4deltapini,vol,beta,error,gamma1,gamma2
      integer loop,imethod,nymph,itype,itempr
      common/param/m1,cp1,gamma1,mach1,thi,tho,pi,m2,cp2,gamma2,tci,tc
o,
      1deltap,dtime,tol,beta,vel2,R,feff,vfac,ua,itype,itempr,imethod

      if (imethod.eq.0) then
c*******************first sizing method begins*******************
c*******************fixed residence time method********************
!!!!!!Preliminary calculations
!!!!!!Properties calculations
!  The  properties  of  the  flow  will  be  evaluated  at  the  mean
temperatures
! in the heat exchanger

      po = pi*(1-(deltap/100))                              !air
pressure loss!
      p1 = (pi+po)/2
      !mean air pressure!
      t1 = (thi+tho)/2                                      !mean   air
temp!
      t2 = (tci+tco)/2                                      !mean
water temp!
      t1stat = t1/(1+((gamma1-1)/2)*mach1**2)
      p1stat= p1/(1+((gamma1-1)/2)*(mach1**2))**(gamma1/(gamma1-1))
      Ro1= p1*101325/(R*t1stat)
      Ro2= 1000                                          !! correlation
needed!!
      Vel1= mach1*SQRT(gamma1*R*t1stat)
      Vel2= Vel1/vfac
      A1= m1/(Ro1*Vel1)                                     !!!
Areas!!!
      A2= m2/(Ro2*Vel2)
      L= Vel1*dtime
      a=120.0
      b=524.07
      mu1=(0.01827*((a+(b*0.555))/(a+t1))*((1.8*t1/b)**1.5))/1000
```

```
      A= 2.414*1.e-5
      B= 247.8
      C_0_water= 140
      mu2= A*10**(B/(T2-C_0_water))
      k1=(1.5207*1.e-11)*T1**3-(4.8574*1.e-8)*T1**2+(1.0184*1.e-4)*T1
      1-3.93333*1.e-4
      k2=0.6

!!!!! Iteration process
      y=0.001
      loop=1
 1000 continue
      W1= A1/Y                !!! widths !!!
      W2= A2/Y
      D1= 2*W1*Y/(W1+Y) !!! hydraulic diameters !!!
      D2= 2*W2*Y/(W2+Y)

      Re1= Ro1*Vel1*D1/mu1
      Re2= Ro2*Vel2*D2/mu2
      Pr1= mu1*Cp1/K1
      Pr2= mu2*Cp2/K2

      F1= (0.79*log(Re1)-1.64)**-2
      F2= (0.79*log(Re2)-1.64)**-2

      nu1= (f1/8)*((Re1-1000)*Pr1)
      1/(1+(12.7*sqrt(f1/8))*((pr1**(0.666))-1))
      nu2= (f2/8)*((Re2-1000)*Pr2)
      1/(1+(12.7*sqrt(f2/8))*((pr2**(0.666))-1))

      h1= feff*(Nu1*k1/D1)
      h2= Nu2*k2/D2

      U=1/(1/h1+1/h2)
      Area=Y*L
      ht1=a1/y                !!! heights !!!
      ht2=a2/y
      vol=area*(ht1+ht2)
      beta=area/vol
      error=ua-u*area
      IF (ABS(error).LT.tol) GO TO 1100
      CALL ITER(1,4,LOOP,y,error)
      LOOP = LOOP + 1
      GO TO 1000
1100  continue

      print *,'re1,re2,pr1,pr2'
      print *,re1,re2,pr1,pr2
      print *,' '
      print *,'f1,f2,nu1,nu2'
      print *,f1,f2,nu1,nu2
      print *,''
      print *,'ua,u*a,error'
      print *,ua,u*area,error
      print *,''
      print *,'y,iterations'
      print *,y,loop

      endif
```

```
c*******************first sizing method ends*******************
1150  continue

c*******************second sizing method begins*******************
      if (imethod.ne.0) then

      po = pi*(1-(deltap/100))                                !air
pressure loss!
      p1 = (pi+po)/2
      !mean air pressure!
      t1 = (thi+tho)/2                                        !mean   air
temp!
      t2 = (tci+tco)/2                                        !mean
water temp!
      t1stat = t1/(1+((gamma1-1)/2)*mach1**2)
      p1stat= p1/(1+((gamma1-1)/2)*(mach1**2))**(gamma1/(gamma1-1))
      Ro1= p1*101325/(R*t1stat)
      Ro2= 1000                                            !! correlation
needed!!
      D1=0.005                               !     Assumption     of
hydraulic diameter
      D2=0.005
      a=120.0
      b=524.07
      mu1=(0.01827*((a+(b*0.555))/(a+t1))*((1.8*t1/b)**1.5))/1000
      A= 2.414*1.e-5
      B= 247.8
      C_0_water= 140
      mu2= A*10**(B/(T2-C_0_water))
      k1=(1.5207*1.e-11)*T1**3-(4.8574*1.e-8)*T1**2+(1.0184*1.e-4)*T1
      1-3.93333*1.e-4
      k2=0.6
      deltapini=pi*deltap*101325/100

!!!!! Iteration process
      vel1 = 2.0
      loop=0
1200  continue
      A1= m1/(Ro1*Vel1*0.85)
      A2= m2/(Ro2*Vel2)
      Re1= Ro1*Vel1*D1/mu1
      Re2= Ro2*Vel2*D2/mu2
      Pr1= mu1*Cp1/K1
      Pr2= mu2*Cp2/K2
      F1= (0.79*log(Re1)-1.64)**-2
      F2= (0.79*log(Re2)-1.64)**-2
      nu1= (f1/8)*((Re1-1000)*Pr1)
      1/(1+(12.7*sqrt(f1/8))*((pr1**(0.666))-1))
      nu2= (f2/8)*((Re2-1000)*Pr2)
      1/(1+(12.7*sqrt(f2/8))*((pr2**(0.666))-1))
      h1= Nu1*k1/D1
      h2= Nu2*k2/D2

      mach1=vel1/sqrt(gamma1*R*t1stat)
      U=1/((1/h1)+(1/h2))
      Area=UA/U
      VOL=Area/beta
      Ht1=VOL/Area
```

```
      Y=A1/Ht1
      Ht2=A2/Y
      L=VOL/A1
      deltaP = (f1*L*ro1*(vel1**2))/(D1*2)
      error=deltap-deltapini
      IF (ABS(error).LT.tol) GO TO 1250
      CALL ITER(1,2,LOOP,vel1,error)
      LOOP = LOOP + 1
      GO TO 1200
1250  continue

      print *,'re1,re2,pr1,pr2'
      print *,re1,re2,pr1,pr2
      print *,' '
      print *,'f1,f2,nu1,nu2'
      print *,f1,f2,nu1,nu2
      print *,''
      print *,'deltapini,deltap,error'
      print *,deltapini, deltap, error
      print *,''
      print *,'vel1,mach1,iterations'
      print *,vel1, mach1,loop

      endif
c*******************second sizing method ends*******************
      return
      end

      SUBROUTINE ITER (NYMPH,IVAR,LOOP,V1,V2)
      IMPLICIT NONE
      DOUBLE PRECISION ERR,VAR,DEDV,V1,V2,V3,DGRAD,DGR,X1,DV
      INTEGER NYMPH,IVAR,LOOP,JLP,ILP
      DIMENSION ERR(20,5),VAR(20,5),DEDV(20,5)
      IF (NYMPH.EQ.9) WRITE (10,8160) NYMPH,IVAR,LOOP,V1,V2
C     THIS ROUTINE CHANGES V1 SO V2 BECOMES ZERO
C------------------  V1 IS THE VARIABLE   V2 IS THE ERROR
      VAR(IVAR,1) = V1
      ERR(IVAR,1) = V2
      DO 1100 ILP = 1,4
      JLP = 6 - ILP
      VAR(IVAR,JLP) = VAR(IVAR,JLP-1)
      ERR(IVAR,JLP) = ERR(IVAR,JLP-1)
      DEDV(IVAR,JLP) = DEDV(IVAR,JLP-1)
C -----------------   FIND VARIABLE FOR THE FIRST LOOP
 1100 CONTINUE
      IF (LOOP.GT.1) GO TO 1200
      V1 = VAR(IVAR,2)*1.000001
      IF (NYMPH.EQ.9) WRITE (10,8100)
      1IVAR,LOOP,V1,VAR(IVAR,2),ERR(IVAR,2)
      DEDV(IVAR,1) = 10000
      RETURN
 1200 CONTINUE

C -----------------   FIND VARIABLE FOR THE 2ND AND LATER LOOPS
      DEDV(IVAR,1) = (ERR(IVAR,2)-ERR(IVAR,3))/
      1(VAR(IVAR,2)-VAR(IVAR,3))
      DGRAD = 2*(DEDV(IVAR,1)-DEDV(IVAR,2))/
      1(DEDV(IVAR,2)+ DEDV(IVAR,1))
      DV = ERR(IVAR,2)/DEDV(IVAR,1)
```

```
      X1 = 1
C-----CHECKS TO MODERATE
C     IF (ABS(DGRAD).GT.0.3) X1 = 2
      V2 = ABS(5*(VAR(IVAR,2)-VAR(IVAR,3)))
      V3 = ABS(DV)
      IF (V3.GT.V2) DV = V2*DV/V3
C-----END OF CHECK
      V1 = VAR(IVAR,2) - DV
      IF (NYMPH.EQ.9) WRITE (10,8100)
      1IVAR,LOOP,V1,VAR(IVAR,2),ERR(IVAR,2)
      IF (NYMPH.EQ.9) WRITE (10,8150) DEDV(IVAR,1),DGRAD
      RETURN
 8100 FORMAT (/'ITER VAR&LOOP:',2I3,'  VARN-VARO-ERR',3E14.6)
 8150 FORMAT ('    -GRADIENT + DGRADIENT',3E14.6)
 8160 FORMAT ('ENTER ITER NPH-IVAR-LP',3I3,'   VAR-ERR',2E12.4)
      END
```

Data file of program

```
78.52 1150.0      1.3   1.0   743.41      0.0   2.94
64.11 1005.0      1.4   622.59      0.0
1.0   1.0   0.0
5.0   0.5   0.00001
700.0 12.0  287.0 10.0  150.0 0.58

Line 1- Fluid 1 properties: m1; cp1; gamma1; mach1; thi(K); tho; pi

Line 2- Fluid 2 properties: m2; cp2; gamma2; tci

Line 3- HX design/sizing selectors: itype; itempr; imethod
      itype = 0   :Counter-flow HX ELSE Parallel-flow
      itempr = 0  :'tho' selected as input for rating purposes ELSE 'tci' selected
      imethod = 0       :sizing method is 'Fixed residence time' ELSE 'Pressure drop comparison'

Line 4- deltap(%); deltat:residence time; tolerance

Line 5- constants: beta; v2; R; feff; vfac; hx_eff
```