Master of Science Thesis

Transient Modelling and Analysis of the OP16 Gas Turbine in GSP

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Summary

With the current trend towards flexible engine operation with high efficiency and low emissions, dynamic modelling of gas turbines has become critical to ensure safe and acceptable engine performance. Dynamic models are essential for design and development of control systems and for analysis of transient manoeuvres that are impractical to test.

This M.Sc. thesis project is carried out in collaboration with OPRA Turbines B.V. The aim of the project is to develop a dynamic model of the OP16 gas turbine. OP16 is a single-shaft all-radial industrial gas turbine rated at 1.9 MW, manufactured by OPRA Turbines. The resulting model can be used to simulate and analyse transient performance of the OP16 engine during manoeuvres of interest to OPRA Turbines. The model can also be used for designing initial fuel control strategies.

The Dutch National Aerospace Laboratory's Gas turbine Simulation Program (GSP) is used as the modelling platform. GSP is a 0-D, component based modelling environment that allows for steady-state and transient performance simulation of any gas turbine configuration. Since transient simulation in GSP is follows the quasi-steady-state approach, the steady-state model of OP16 is first developed. In order to evaluate the influence of steady-state heat loss on engine performance, thermal network modelling is also included. The steady-state model is verified against measured data across the entire operational envelope of OP16. The model is found to accurately simulate the steady-state performance of OP16 near the full load operating point.

The steady-state model is extended to simulate transient performance of OP16 by implementing engine-specific details. The transient model is verified by comparing model simulations to the measured data for a load step near the full-load operating point. The transient effects including rotor inertia, heat soakage and volume dynamics are analysed to determine their influence and importance for the transient behavior. Rotor inertia is found to dominate the transient behaviour of OP16 while heat soakage effects and volume dynamics remain negligible in comparison.

The transient model of OP16 is used to simulate engine behaviour when performing load sheds. The influence of fuel heating value, fuel valve closing time and combustor volume is analysed. One important aspect to guarantee safe operation is to be able to shut-off the fuel supply as soon as possible in case of an engine trip. From the simulations it is found that lower LHV fuels tend to result in higher rotor over-speeds in case of a load shed. For a LHV as low as 5.8 MJ/kg, the fuel valve closing time during a load shed should not exceed 0.7 seconds in order to maintain the rotor over-speed within acceptable limits, whereas for natural gas type of fuels the fuel valve closing can be up to 0.9 seconds.

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Nomenclature

Roman symbols

A	Area
C	Celsius
C_p	specific heat at constant pressure
deg	degree
h	specific enthalpy
h	heat transfer coefficient
I	moment of inertia
M	Mass
N	rotor speed, rpm
N_c	corrected speed
p	pressure
PW	Power
Q	Heat flux
R	specific gas constant [J/kg/K]
t	time
T	Temperature
u	internal energy
U	overall heat transfer coefficient
V	Volume
W,w	mass flow

Greek symbols

γ	ratio of specific heats
Δ	non-infinitesimal change
ω	angular velocity [rad/s]

Subscripts

abs	absorbed
amb	ambient
c	corrected
cond	conductive (heat transfer)
comp	component
conv	convection (heat transfer)
del	delivered
g	gas
hs	heat sink
p	pressure

rad	radiation (heat transfer)
V	Volume

Abbreviations

ANN	Artificial Neural Networks
GSP	Gas turbine Simulation Program
LHV	Lower Heating Value of fuel
NARX	Non-linear Auto-Regressive network with exogenous inputs
NARMAX	Non-linear Auto-Regressive Moving Average model with exogenous inputs
NGV	Nozzle Guide Vanes
NLR	Nationaal Lucht-en Ruimtevaart laboratorium (National Aerospace Laboratory, The Netherlands)
OEM	Original Engine Manufacturer
PID	Proportional Integral Differential (control)
TOT	Turbine Outlet Temperature

1 Introduction

Modeling and simulation has always been a significant part of design and development of a gas turbine engine. It serves to be an economical and time-effective alternative to experimental testing of expensive prototypes. Earlier, the use of such models was restricted to calculation of engine thermodynamic cycle, and prediction of component performance and thermal and structural loads. Development of new modeling techniques combined with increasing computing power has led to a growing extension of modeling scope. As observed by Asgari, et al. (2014, p.1) in [3], modeling and simulation of gas turbines has developed into a standard means for optimization of design, manufacturing, performance, and trouble shooting.

Modelling of transient behaviour of a gas turbine is crucial for the design and development of the control system [11]. Dynamic models are also needed for investigation of engine transient behaviour in extreme areas of the operational envelope and during manoeuvres that are expensive or impractical to test [26]. Owing to increasing demand for flexible engine operation with high efficiency and low emissions, transient modelling has become vital to ensure safe and acceptable gas turbine performance.

This work aims to develop a dynamic model of the OP16 that can be used for analysing its transient behaviour during manoeuvres of interest and for initial control system design purposes. The OP16 is a single-shaft all-radial industrial gas turbine rated at 1.9 MW, manufactured by OPRA Turbines B.V. Gas Turbine Simulation program, or GSP, is used as the modelling environment. GSP is a commercial gas turbine performance modeling tool developed by NLR, the Dutch National Aerospace Laboratory together with Delft University of Technology.

1.1 Problem definition and research objectives

Gas turbine transient operation presents specific operability concerns that do not arise at steady-state. Transient working lines of a compressor deviate considerably from their steady-state location. Other than rotor inertia, the transient phenomena such as heat soakage or volume dynamics might have an impact on the transient working line excursions [26].While rotor inertia alone is able to accurately predict transient performance of a gas turbine in most cases, heat soakage effects and volume dynamics may well be important, depending on engine size and design. Heat transfer from hot gas to the surrounding metal structure and the ambient can significantly affect the steady-state and transient performance in case of smaller gas turbines [16]. Furthermore, volume dynamic effects in larger components like certain combustors or recuperators may have a major impact on transient behavior of a gas turbine engine. With accurate system models these effects can be analyzed for OP16.

Besides the transient phenomena discussed above, fuel properties can also influence gas turbine behaviour. Fuels with low calorific values have a lower energy density. Therefore a higher fuel flow rate is required, which can impact the dynamic behavior of a gas turbine. Since the OP16 has the capability to burn a wide range of fuels including ultra-low calorific gaseous fuels, the influence of fuel LHV on engine performance is worth evaluating.

For power generation applications, *load shedding* is one of the transient manoeuvres of concern. Load shedding refers to a sudden loss of generator load which could occur, for example, because of an electrical failure. In case it happens, the fuel flow is required to be reduced quickly to avoid rotor over-speed. Load shedding is particularly critical when the engine is running at full load since it results in higher rotor speeds due to a larger power unbalance. The low calorific fuels, such as syngas, have typically a high temperature due to the process in which they are being produced. Fast acting fuel valves capable of handling high temperatures are scarcely available. In order to avoid customized and expensive valves it is important to know the maximum time required to safely close the valve in case of an engine trip.

The objective of this thesis assignment is to develop a dynamic model of the OP16 gas turbine using GSP including the thermal network modelling functionality. In this regard, the transient phenomena significant to the OP16 performance are investigated in order to have a better understanding of its transient operation and to accurately simulate its transient performance. Since the OP16 is a small gas turbine, it is essential to look into the associated thermal effects. The resulting model should be able to predict the generated mechanical shaft power and exhaust gas temperature as functions of measured variables and estimated parameters. The model intends to provide enough insight into the transient behaviour of OP16 to facilitate initial design of control strategies to regulate the delivered power and the exhaust gas temperature. The model can also be used to analyse transient manoeuvres of interest to OPRA Turbines.

As an application of the developed model, this work analyses the rotor speed during load shedding for the OP16 gas turbine. In order to achieve acceptable engine performance when running on non-conventional fuels, the influence of fuel LHV on OP16 transient response when performing load sheds is analysed. The results from the investigation are used to derive the maximum allowed closing time of the fuel supply valves as a function of the fuel LHV.

1.2 Structure of the report

Chapter 2 explores the modelling techniques currently used in the industry to model gas turbine performance. Both steady-state and transient modelling methods are discussed since transient modelling of a gas turbine is often based on its steady-state characteristics. The physical phenomena associated with gas turbine transients are explained. Suitability of GSP as a modelling environment for OP16 is then determined.

Chapter 3 describes the engine configuration of OP16 along with the most important specifications of the gas turbine.

Chapter 4 tackles the steady-state performance modelling of the OP16 gas turbine in GSP. Application of thermal network modelling capability (available in GSP) to model steady-state heat losses is presented. The results from the model validation using measured data across the OP16 operating envelope are shown.

Chapter 5 deals with the transient performance modelling of OP16 in GSP. The rotor inertia, combustor volume dynamic effects and heat soakage are analyzed to determine their significance for the OP16 transient behavior. The results of the model validation against measured data are then discussed.

Chapter 6 analyzes the OP16 rotor speed during load shedding. The influence of the fuel LHV and combustor volume dynamics on the rotor over-speed are investigated. The fuel valve closing time required for safe operation in the event of a load shed, when running on low LHV fuels, is subsequently determined.

The conclusions and recommendations are presented in Chapter 7.

2 Literature review

This chapter reviews the off-design and transient modelling methods prevalent in the industry for gas turbine performance simulation and analysis.

2.1 Design point calculation

Gas turbine performance simulation requires design point performance of the engine to be defined before performance at any other operating condition can be simulated.

It is a common practice to choose the design point at the operating point where an engine would run the most. Design point analysis is done to optimize the thermodynamic cycle at the chosen design point by selecting the right engine configuration, number and type of compressor and turbine stages, overall pressure ratio and turbine inlet temperature. This is usually done by iteratively calculating the engine cycle based on fundamental equations of gas turbine thermodynamics by varying the design point input parameters [18]. The important design parameters are the overall pressure ratio, stator outlet temperature and component performance parameters in terms of efficiencies, flow capacities etc. Sample calculation examples suited to digital computer programs are discussed by Walsh and Fletcher (1998) in [26]. The design point calculation fixes the geometry of the engine. Any engine, however, runs within an operating envelope and not just at one design point and hence there is a need for off-design performance optimization. Off-design modeling is thus essential to meet the required steady state and transient performance, exhaust gas emissions, noise levels across the complete operating range.

2.2 Off-design modelling

Depending on the level of details sought for, the off-design models can be 0-D, 1-D, 2-D or 3-D. 0-D models calculate the properties of the working fluid at discrete points, usually at the entry and exit of a component. The component itself is treated as a black-box. Higher dimensional models (1-D, 2-D and 3-D) progressively introduce continuity in the calculations. 1-D models calculate the fluid properties along the gas path center line (and not just at the exit/entry of a component) while 2-D models extend the computation to the radial dimension (assuming circumferential symmetry) while 3-D models use the complete set of equations of conservation.

0-D models do not require a detailed description of the engine geometry and are easy to compute since the number of unknowns is fewer than their higher dimensional counter parts. Their simplicity allows them to run on small personal computers and doesn't necessarily call for a computer cluster. This is the main reason 0-D models are widely used for gas turbine performance simulations. Higher order models on the other hand are likely to be more accurate. Besides, they also allow for detailed modeling of the underlying physics. But this comes with a requirement for higher computational power. Despite its simplistic nature, 0-D modeling technique can result in good predictions depending on the accuracy of the component performance characteristics.

2.2.1 Thermodynamic matching method

Steady state off-design behavior can be calculated once the engine configuration and geometry is fixed based on design point analysis. This is almost universally done by a method called 'thermodynamic matching'. As explained in [26], for a fixed engine design, the performance at varying operating conditions may be defined by matching the component characteristics. These component characteristics (also called maps) typically define the interdependence of mass flow, pressure ratio and speed, and these parameters are usually expressed as dimensionless quantities. Fixing the component geometry fixes the component map. Thus, once the design point is fixed, the component maps are fixed too. For any operating condition, there can be multiple possible operating points on each component map. Since the conservation equations (of mass, energy and momentum) along the gas path need to be satisfied, the operating point on each component map is dependent on the maps of the components it is connected to. Matching of these operating points results in a unique component operating point at every operating condition. For a one spool turboshaft engine, matching of components translates into compatibility of work and flow between the compressor and the turbine. This matching process is usually highly iterative.

Most of the literature on modeling the steady-state off design behavior of gas turbines is based on thermodynamic matching model of the 0-D type. For the majority of these models, the flow through the gas turbine is assumed to be adiabatic, meaning; there is no internal or external heat transfer. Since the temperature of the gas is higher than the air outside, there is always some heat transfer from the hot gas to the casing and to the ambient. Also, there is some heat transfer between adjacent components. Neglecting it is usually a valid assumption for large engines but not necessarily for smaller engines depending on surface to volume ratio [16]. [9] explains the effect of internal and external heat transfer on gas turbine steady-state performance. It was observed that this heat transfer changes the operating point of the gas turbine which induces a drop in performance and more fuel is required to produce the same shaft power.

2.3 Transient modelling

Transient modeling is more complex than steady-state modeling and it is essential to understand the transient behavior of a gas turbine engine to be able to simulate it using physical laws. Transient performance means that the cycle parameters are changing with time. Examples of such operating regimes are engine start-up, shut-down, accelerations, decelerations etc.

Transient performance and control system design go hand in hand since the engine responds to ‘controlled variables’ like fuel flow schedules, bleed flow variations etc. [26]. For a single spool engine for electrical generation, the following could be an example: if there is a step increase in load, the spool speed would initially decrease. The control system would then increase the fuel flow to increase the spool speed back to the synchronous value. The transient behavior of the engine performance parameters in this exercise would depend on two things: the physical processes going on in the gas generator during this time and the control system dynamics that works behind regulating the fuel flow to get back to the required speed during the same time. Thus, to obtain meaningful results, it is very difficult to investigate the physical processes going inside an engine without including the control system dynamics.

2.3.1 Transient performance phenomena

The major physical phenomena associated with transient performance of a gas turbine engine are rotor inertia effects, heat soakage effects, volume dynamics, changes in tip clearance, combustion delay, gas dynamics and control system lags [15,20].

Rotor inertia effects deal with shaft accelerations (or decelerations) owing to any power unbalance introduced onto the shaft. The power unbalance occurs whenever the engine operating conditions are changed. The associated transient phenomenon is called rotor inertia effects and is highly dependent on the inertia of all elements connected to the shaft.

Heat soakage effects refer to heat transfer rates between the working fluid and the metal parts around it whenever there is a change in the temperature difference between the two. While it could be negligible for steady-state operation (subject to the size of the gas turbine), it is significant during transient operation. Heat soakage effects are largest for components having large surface areas and large thermal masses subjected to high temperature changes, for example, the combustion chamber.

Volume dynamics means that the mass flow going in is not equal to that coming out of a component because pressure, temperature and thus density of the fluid vary with time. Each volume acts as storage of mass and energy resulting in a rate of change in fluid properties within the volume. This effect is also called gas dynamics and is higher for components with higher volume.

Combustion delays refer to the time delay between fuel injection and release of thermal energy after combustion. Control system lags refer to time taken by the control system hardware to react to any input.

It is important to identify which of the transient effects are dominant and which can be neglected for a particular application to efficiently model the transient behavior of any gas turbine engine. Fundamental equations representing the physical transient phenomena discussed in [26].

2.3.2 Quasi-steady-state approach to transient modelling

The 0-D type thermodynamic matching model described earlier in the report for steady-state off design modeling can be extended to model the transient behavior of a gas turbine by including dynamic equations representing the physics of the associated transient phenomena [17]. Except for during the primitive design phase it is common to include the control system model in the engine model to ensure meaningful scheduling of fuel flow etc. Such a model essentially contains the control program equations specifying the control variables (e.g. fuel flow, bleed flow etc.) to compliment the set of thermodynamic equations that represent the gas generator transient performance.

One of the most common ways of doing this is called 'Thermodynamic matching transient performance and control model' [26]. The thermodynamic part of this method is largely based on the steady-state model and hence it is often called the 'quasi-steady-state' approach to transient modelling. The model starts the calculation at time zero to calculate an initial steady-state point (exactly same as the steady state model) followed by incrementing time by a small value. The sub-routines that model the transient phenomena (such as heat soakage, volume dynamics etc.) are then activated. Also, time derivatives to the shaft power conservation equations are introduced in the model. The unbalance in power can then be calculated iteratively to match the fuel flow of the thermodynamic model to that set by the control algorithms. Owing to 0-D nature of the method, steady-state component performance characteristics are used for thermodynamic calculations. Suitable lags and delays can be applied to the control model routines to model a real controller.

Being an accurate representation of physical engine transient performance parameters, this type of model can be used for the analysis of engine transients. It can also be used for designing the schedules for fuel flows, bleed flows etc. to achieve engine transient requirements while maintaining proper surge margin. Another application could be in initial design of engine control laws. Other advantages of this model include prediction of engine transient behavior in extreme areas of the operational envelope where testing may not be feasible and investigation of maneuvers that are expensive or impractical to test.

A non-linear model for a low power single-shaft gas turbine based on engineering thermodynamic principles is discussed in [1]. Some assumptions were made to reduce the complexity of the mathematical model. These assumptions neglected some physical processes integral to gas turbine dynamics like volume dynamics and thermal effects. The model was used to investigate the dynamic response of the gas turbine to a pre-defined fuel flow step and didn't include a control system model. The results fulfilled the objective of initial control loop-shaping. A method to simulate transient behavior of a gas turbine engine (turbojet) based on thermodynamic laws is also described in [10]. The influence of fuel flow schedule and bleed flows has been investigated. Only rotor inertia effects on transient behavior have been taken into account in this work.

1-D transient models are used to model the physical transient processes in a greater detail. Such models typically try to extend 0-D models by including an additional dimension which requires detailed information of a component. This means the individual components cannot be treated as black boxes unlike in 0-D models. They are frequently used in cases where a specific component or a local area within an engine is required to be modeled with higher order of fidelity. Venturini (2005) discusses dynamic modeling of a compressor based on conservation of mass, energy, momentum, moment of momentum and heat balances using a 1-D approach in [21]. Each component is represented as one or more equivalent annular shaped cylinders whose dimensions have been calibrated using experimental data. The simulated results were found to be very close to the experimental values but at the cost of increased computation power and longer running times.

2.3.3 Black-box modelling

All the models discussed until now were based on the knowledge of physical processes governing the system behavior. It is not necessary that this information is always known. This is especially true for gas turbine operators and users who do not have access to sufficient engine data. This has led to the development of a modeling technique called 'Black-box modeling'. These models use measured or simulated data to construct a model that matches this data in the best way possible. This technique is primarily applied to create a dynamic model. Based on the availability of data and the objective of gas turbine modeling, a lower or a higher order model can be created.

One of the most important methods of black-box modeling found in literature is 'Artificial neural network' or ANN. Inspired by biological neurons, artificial neural network is a system of inter-connected computing nodes that take numerous inputs, apply some weight on each of these inputs and give some output(s) by non-linear summation of these weighted inputs. The applied weights are adapted to the data fed to the network during an iterative process called 'training'. The trained networks are then able to predict an output value from any set of input data.

The advantage of black box modeling is that it can give accurate results even when there is a lack of physical information about the system to be modeled. This is particularly beneficial for the users who do not have access to enough data to create a thermodynamic model.

Asgari, Chen and Sainudiin (2013) discuss various black-box modeling techniques that have been researched so far, in [2]. ANN method of modeling was investigated by Bettochi, et al. (2004) in [7] to model a single-shaft gas turbine and was found to be useful for real time simulation of gas turbines especially in case of lack of enough information about the system dynamics. Similar modeling technique was used for design and off-design modeling of a single shaft industrial gas turbine by Lazzaretto and Toffolo (2001) in [14] and the resulting model was found to have good prediction accuracy. In another research work NARX model was applied for system identification of a heavy-duty power plant gas turbine [4]. Chiras, Evans and Rees (2002) describe the use of NARMAX structures to create a non-linear gas turbine model [8]. The work establishes the performance superiority of non-linear models over the linear models.

The drawback of black box modeling is that it cannot be used at the design phase of an engine since there is no data available and the only option is to create a thermodynamic model based on the knowledge of physical laws and processes governing the system. Besides, black-box modelling techniques do not allow derivation of meaningful sensitivities obviating any scope of physical analysis of engine behaviour.

2.4 GSP as a modelling environment

Gas Turbine Simulation Program (GSP) is a gas turbine modeling tool developed by NLR, the Dutch National Aerospace Laboratory together with Delft University of Technology. It provides a component based modeling environment where different component sub-models can be stacked together to represent a specific gas turbine configuration. The extensive component library available in GSP allows the user to virtually model any gas turbine engine. GSP primarily follows 0-D thermodynamic based approach to gas turbine modeling. The averaged values of fluid properties are calculated at the inlet and exit of each component. The simulations are based on conservation equations (of mass and energy) and component performance characteristics and the resulting system of non-linear differential equations is solved by using customary numerical methods (e.g. Newton Raphson).

Steady-state off-design performance calculations use the thermodynamic matching method as described earlier. Transient performance calculations are based on quasi-steady-state approach [24]. It is essentially same as the thermodynamic based transient modeling method described previously. GSP includes the following transient effects: rotor inertia, volume dynamics, heat soakage and control system dynamics in case control system models are added. GSP allows for addition of 1-D component models at specific locations in the system model.

Recently, detailed thermal effect models have been implemented in GPS, which are of great relevance for small gas turbines where thermal heat losses could be significant. This thermal network modeling capability can be used to simulate steady state heat loss as well as the transient heat soakage effects. The approach uses engine details corresponding to 1-D models but the true mathematical computation still remains 0-D [25].

Besides, GSP component models can simulate variable geometry, Reynolds effect and deterioration for the components which use performance maps. GSP also provides an option to customize the component models for a specific application in case detailed analysis is required. Other applications include real time simulation and gas path analysis. Visser (2015) extensively explains the working principles of GSP and its capabilities in [24].

2.4.1 Conservation laws

The conservation laws used by GSP to simulate gas turbine engine performance are discussed by Visser (2015) in [24]. The relevant equations are reproduced below.

The conservation of mass for flow through a component:

$$\frac{dM_v}{dt} = W_{in} - W_{out} \tag{2.1}$$

Assuming isentropic flow,

$$\frac{dM_v}{dt} = \frac{V_{comp}}{\gamma RT} \bullet \frac{dp}{dt} \tag{2.2}$$

The conservation of energy for a component:

$$\frac{dM_v}{dt} \cdot u + M_v \cdot \frac{du}{dt} - Q = w_{in} \cdot h_{in} - w_{out} \cdot h_{out} + PW_{abs} \tag{2.3}$$

Since GSP is a 0-D model, equation 2.3 can only be evaluated in steps for most of the components. For example, the calculation of the power delivered (or absorbed) by the turbine (or the compressor) is done using isentropic or polytropic efficiencies derived from a performance map, assuming no heat transfers (adiabatic process) and no volume dynamic effects. This implies that the terms on the left hand side of equation 2.3 are first assumed to be 0. Once PW_{abs} is calculated, the calculation of these terms is performed for inlet, exit or averaged conditions and used to correct power and component exit conditions.

The conservation of energy for a drive shaft:

$$I \cdot \frac{d\omega}{dt} \cdot \omega = PW_{abs} + PW_{del} \tag{2.4}$$

The equations for conservation of energy for a heat transfer between the gas (g) and a heat sink (hs) element in the thermal network component models (see section 4.3.1):

$$Q_{hs} = U \cdot A \cdot (T_{hs} - T_g) \tag{2.5}$$

$$M_{hs} \cdot C_{p_{hs}} \cdot \frac{dT_{hs}}{dt} = \sum Q_{hs} \tag{2.6}$$

2.4.2 Transient simulations

To simulate transient performance of a gas turbine, the time derivatives in the equations of conservation discussed above are integrated using the modified Euler method [24]. The corresponding equation is reproduced below:

$$x_{i+1} = x_i + \Delta t \cdot x'_{i+1} \tag{2.7}$$

At every time-step, the equations of conservation are iteratively solved using the Newton-Raphson method. Since the time derivative terms in the conservation equations are non-zero during transient simulations, equation 2.7 implicitly becomes a part of the system of equations. The transient effects not captured by the steady-state performance characteristics cannot be included in this type of transient simulation model and hence it is called, quasi-steady-state simulation. For further details on numerical solution methods used in GSP, the reader is referred to [24].

2.5 Creating a performance model using a gas turbine simulation tool

An efficient method to create a performance model using a simulation tool such as GSP is discussed by Kurzke (2005) in [12]. According to this method, a suitable design point is chosen and the model is adjusted to the data of this point by changing the design input parameters like compressor pressure ratios, engine mass flow, component efficiencies and burner exit temperature. It is important to note that the model design point need not be same as the actual design point of an engine during its development phase where there could be different design points for different components and different operating conditions.

The next step is to build or tune the component maps to match the available off-design data. One could start with the compressor map by adjusting the corrected flow-efficiency correlation to match the compressor efficiency data followed by tuning the corrected flow-corrected speed correlation since the latter has only minor effects on the formerly adjusted correlations. It is advisable to not concentrate on the spool speed data in the first step of creating an off-design model. Owing to the limited operating range of the turbine, the effect of a turbine map on simulation results is limited. Adjusting the efficiencies on the turbine map can most likely result in a satisfying match.

Based on reliability of data available, sophistication of the model can be increased by plugging in the finer details like specifics of internal air system, gearbox losses, thermal heat losses etc.

2.6 Conclusions

It is clear from the literature discussed so far that the dynamic modeling approach highly depends on the modeling objective and availability of data. The ultimate goal of the project is to model the transient performance of the OP16 to be able to simulate and analyze transient behaviour during manoeuvres of interest. The model also intends to be used for initial control system design purposes. 0-D models can adequately serve this purpose if accurate component performance maps are used. Since OPRA Turbines is the manufacturer of the OP16, availability of reliable component performance data is not a problem and, hence, 0-D modeling technique can result in accurate predictions.

The physics associated with transient phenomena is generic and with the help of specific engine data of OP16, the 0-D thermodynamic modeling can be easily extended to simulate its transient performance. Because physical modeling of transient phenomena also allows the derivation of meaningful sensitivities, thermodynamic transient modeling is preferred over black-box modeling. Owing to its comparatively smaller size, transient modeling of the OP16 should take into consideration the associated thermal effects. Also, to achieve meaningful transient simulation and for model validation against measured data, control system dynamics needs to be considered.

The resulting dynamic model of OP16 would provide an insight into the steady-state and transient behavior of a small industrial gas turbine in the range of 2 MW shaft power and the physical parameters that govern this behavior.

Following from the analysis of the research discussed above, GSP is concluded to be a suitable tool to model OP16 since it uses 0-D type thermodynamic based modeling techniques. GSP also allows for modeling any gas turbine configuration by providing a wide range of component sub-models which can be linked together thermodynamically to represent any possible gas turbine engine.

A steady-state model of the OP16 including the steady-state heat loss and finer details like specifics of internal cooling system would be developed in GSP. The model would be validated by tuning/calibrating the compressor and the turbine maps as well as the heat transfer parameters in order to match the measured data. The transient model including rotor-inertia, heat soakage effects and volume dynamics would be developed from the steady-state model, again on GSP, with the help of engine specific details like component volumes, rotor inertia, thermal mass of the components etc. For validation of the transient model against measured data, control system dynamics should be taken into account.

3 The OP16

The OP16 is a small single-shaft industrial gas turbine engine designed for power generation applications in the 2MW range. The all-radial configuration of OP16 consists of a single-stage centrifugal compressor closely coupled to a single-stage radial turbine resulting in a compact arrangement. Figure 3.1 shows the OP16 gas turbine and figure 3.2 shows a 3-D view of the engine from inside.

The major components of the OP16 are listed below:

- Reduction gearbox
- Shaft with bearings
- Compressor
- Combustion chambers (4x)
- Turbine and exhaust diffuser

The centrifugal compressor running at 26000 rpm compresses the inlet air and delivers it to the combustion chamber. The combustion system of the OP16 comprises of four combustor cans. The combustor cans have a reverse-flow configuration as can be seen in figure 3.2. Some combustor can variants, designed for burning low LHV fuels, are larger in volume to provide sufficient time for complete combustion of the low-calorific value fuels. After undergoing combustion, the high pressure, high temperature gas is directed to the single turbine stage through the nozzle guide vanes. The hot gas expands through the turbine and enters the exhaust diffuser where final expansion takes place before the gas exits at ambient pressure.

Owing to compact arrangement of the OP16 compressor and turbine, the rotor shaft is cantilevered to the bearing housing located at the cold engine end (compressor side). This adds to rotor robustness and compactness and eliminates the need for bearing lubrication at the hot end.

The speed reduction gear located next to the inlet housing reduces the shaft speed from 26000 rpm to 1800 rpm or 1500 rpm for 60 Hz or 50 Hz applications, respectively. Accessory drives for starting motor, fuel pressure pump and lubrication pump are incorporated into the gear box.

The main specifications of the OP16 engine are provided in table 3.1.

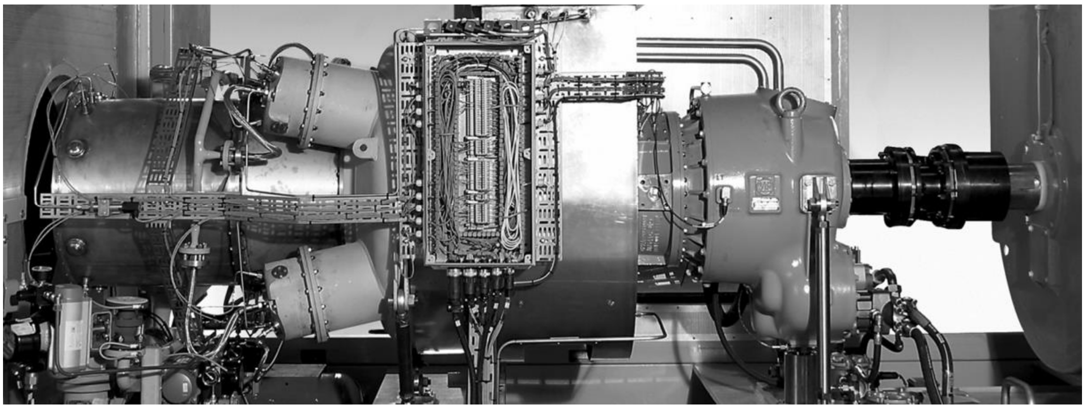

Figure 3.1 The OP16 gas turbine engine

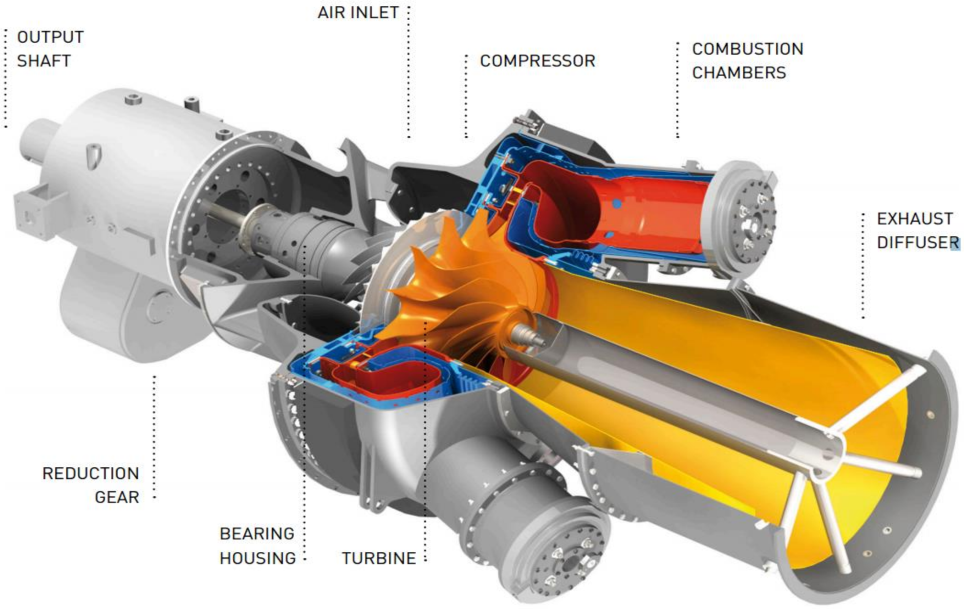

Figure 3.2 Cut out 3-D view of the OP16 engine

Table 3.1 OP16 gas turbine specifications

Number of shafts	1
Rotational speed [rpm]	26000
Pressure ratio [-]	6.7
TOT [K]	844
Engine inlet mass flow [kg/s]	8.8
Shaft power [kW]	1900
Thermal efficiency [%]	26

4 Steady-state performance modelling

As discussed earlier, GSP can simulate steady-state (and transient) performance of a gas turbine for any valid user specified arrangement of components. This calculation is based on the thermodynamic conservation laws and component characteristics (in the form of component maps) for a given operating condition (section 2.4). For steady-state simulations, the time derivatives in the conservation equations given by equations 2.1 through 2.6, become zero since all the cycle parameters are assumed to be stable and constant with time. The calculation is performed relative to a reference operating point, usually the design point. Selection of a suitable model design point is thus required before one can run simulations for operating conditions deviating from the engine design point. The off-design model can be adapted to match the measured performance data by adjusting the component maps. Performance data plays an important role in tuning the design and off-design parameters of the model so that it best replicates the actual engine behaviour.

This chapter starts with a description of the available performance data that is used to build the OP16 model in GSP. This is followed by an explanation of the model configuration, corresponding to OP16, in GSP. Next, the thermal network modelling of the gas turbine is described after which design point modelling and off-design modelling is discussed. Subsequently, the steady-state performance model of the OP16 is verified against measured data. Based on the model validation results, engine losses are investigated in the final section of this chapter.

4.1 Available performance data

Performance data used to create and validate the model of the OP16 in GSP comes from engine testing in the OEM's test-cell. The newly built engines are always tested before shipping them to the field. The engine is accelerated to the nominal speed until the engine is running at idle condition. Once steady-state is reached, load is applied in small steps until the engine is running on full load. At every load step, the engine is allowed to come to a steady-state before the next load step is applied. Upon reaching steady-state at full load, the engine load is brought down to zero by reducing the load in steps. During this entire manoeuvre, the control system regulates the fuel flow to maintain the rotor speed at the nominal value. The variation in load (with time) for one such engine loading manoeuvre is shown in figure 4.1 while figure 4.2 depicts the corresponding variation in fuel flow. The ambient conditions, the engine mass-flow, the compressor inlet and exit conditions, the fuel flow and the turbine exit conditions are measured in the test-cell. The shaft power is calculated back from generator load by applying generator efficiency curves.

The performance tests, thus, represent the part-load steady-state performance of a healthy OP16 engine. Data from such tests performed at varying ambient conditions and with fuels of varying calorific values is available from the OEM’s database.

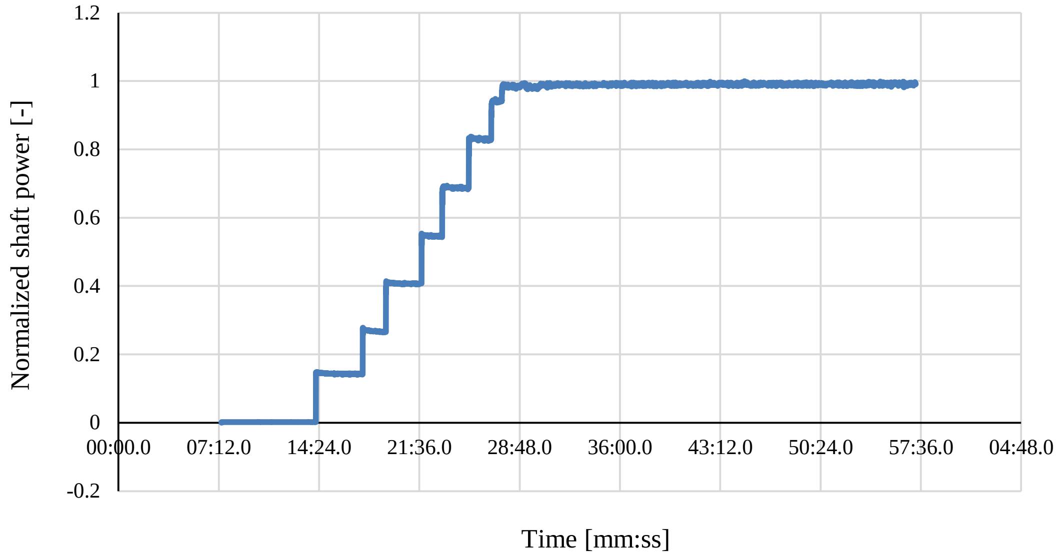

Figure 4.1 Variation in shaft power for OP16 engine loading

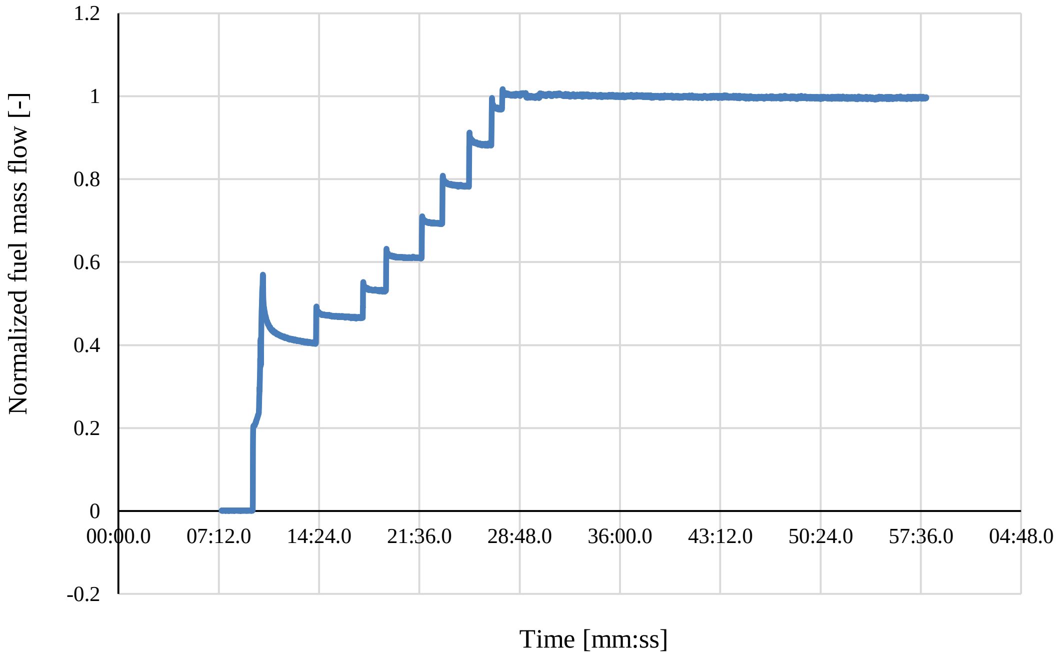

Figure 4.2 Variation in fuel mass flow for OP16 engine loading

4.2 Model configuration

Figure 4.3 represents OP16 in the GSP modelling environment.

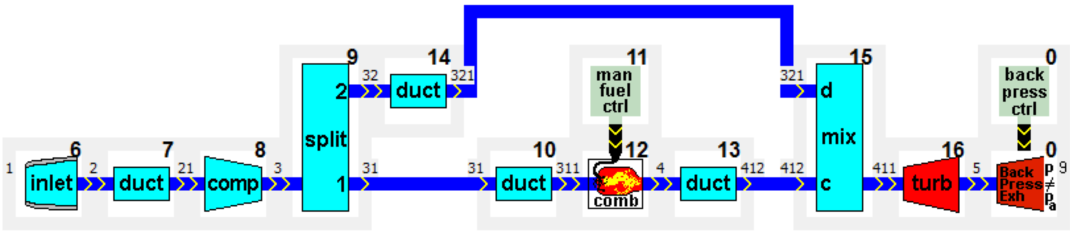

Figure 4.3 OP16 model configuration in GSP

An inlet, a compressor, a combustion chamber, a turbine and an exhaust comprise the primary gas path components of the OP16 model. The turbine component in the model represents the OP16 single-stage turbine as well as the exhaust diffuser. The ducts are used to implement pressure losses or heat transfers at relevant stations. The ducts do not allow for detailed heat transfer modelling but require the user to specify known or estimated heat flux. The splitters and mixers allow specification of secondary flows in the gas turbine.

GSP allows the user to specify a required power level in a form desired by the application. It could be in terms of fuel flow, turbine inlet temperature, turbine exit temperature, rotor speed, exhaust pressure ratio, and shaft power or torque or engine thrust. Since fuel flow directly affects the power output of a gas turbine, it is the simplest way to specify the power level in a gas turbine model. Unlike other power setting parameters, it does not require an additional equation to compute the unknown fuel flow corresponding to the specified power level. Moreover, the fuel flow is directly measured at the OEM’s engine test-cell. For these reasons, fuel flow is selected as the power level setting control for the OP16 model.

The compressor inlet does not always feel the ambient pressure and temperature owing to hardware installed in front of it, for example, inlet air filtration systems. Similarly, an exhaust system installed at the end of the turbine diffuser prevents the working fluid to expand to the ambient pressure. This is felt by the gas turbine as extra pressure losses depending on the application or the installation, also called installation effects. In the model, the total inlet pressure loss is specified in a duct before the compressor entry (see figure 4.3). The exhaust pressure losses are taken care of by specifying a back pressure at the exit of the turbine component, different from the ambient pressure. This back pressure is the exhaust pressure of the working fluid and is usually a measured quantity and thus can be conveniently specified through a back pressure control component. The difference between the total pressure at turbine exit and the ambient pressure is the exhaust pressure loss.

In OP16, a small part of the compressor exit flow is diverted from the core flow to cool the structure at the turbine entry. This is represented in the model by a splitter at the compressor exit (component 9 in figure 4.3). The heat picked up by this secondary flow before it mixes with the core flow at turbine entry is specified in a duct (component 14 in figure 4.3). The value of this heat flux is implemented as a function of the fuel flow setting by using a 1-D scheduler component within GSP. The mixer, component 15 in figure 4.3, represents mixing of the secondary cooling flow with the core flow. Thermodynamically, it is same as implementing the NGV cooling functionality available in GSP but it does not allow for specifying a heat transfer between the working fluid and the metal structure around it. Besides, using a splitter and a mixer creates an additional station at the turbine entry making it easier to analyse the flow conditions at that point.

4.3 Steady-state heat loss modelling

The casing around the gas path in a gas turbine is in contact with the hot gas on the inside and cooler air on the outside resulting in heat transfer from the hot gas to the environment, through the casing. Additionally, there will be some heat transfer from the hot section(s) of a gas turbine to any connected cooler section(s) through the casing material. The heat transfer depends on the temperature of the hot gas which in turn depends on the power setting and operating conditions. When running at a steady-state, there usually is a constant heat transfer from the hot gas to the ambient. The influence of these thermal effects on gas turbine performance increases for smaller gas turbines [9, 16]. To determine how significant the steady-state heat loss is in the case of the OP16 was enough motivation to model the thermal effects of this small (but not very small) gas turbine.

This section starts with a description of the thermal network modelling functionality recently added to GSP followed by modelling assumptions and simplifications introduced for modelling the thermal effects in OP16. The resulting thermal network model of OP16 is then discussed.

4.3.1 Modelling approach in GSP

The thermal network modelling functionality available in GSP is a generic approach to heat transfer modelling in a gas turbine, developed by Visser and Dountchev (2015) [24]. The heat transfer effects can be modelled by adding non-gas path components, called *Heat Sinks*, to the model configuration. A heat sink can be thought of as a structure with a user specified mass and specific heat capacity, through which heat transfer can be simulated. The heat sinks can be connected to each other, to the gas path components, to the ambient or to any user-specified external conditions, forming a thermal network. Each connection in the network represents a heat transfer link. The associated heat transfer is calculated based on user-specified parameters and the heat sink temperature, using fundamental heat transfer theory. It is possible to simulate convection, conduction, and radiation heat transfers depending on the nature of the heat sink connections. In a 0-D thermal network, each heat sink is assumed to be a point mass with a mean temperature. If a heat sink *i* (with mass M_{hsi} , specific heat capacity c_{phsi} and temperature T_{hsi}) is connected to *n* other elements, we have, from conservation of heat energy, the following equation:

$$\sum_{1}^{n} Q_j - M_{hsi}.c_{phsi}.\frac{dT_{hsi}}{dt} = 0 \tag{4.1}$$

The first term of the equation above represents heat transfer rates between the heat sink *i* and the elements it is connected to while the second term represents the rate of change of heat content of the heat sink itself (which is proportional to the rate of change of the heat sink temperature). The latter becomes zero for steady-state simulations. Hence, addition of a heat sink component to a model configuration adds a state, the heat sink temperature, and an error equation corresponding to equation 4.1 to the gas turbine modelling system. It is up to the user to define the heat transfer models by providing emissivity for radiation, Reynolds number and Nusselt number expressions for convection and characteristic thickness and conductivity for conduction. The modelling details of each of the heat transfer types can be found in [25].

The 0-D thermal network modelling does not provide room for detailed modelling of heat transfers and temperatures but it is able to represent the thermal effects on the performance of a gas turbine.

The calculated heat transfer rates are added or subtracted from the enthalpy at the component exit in case of components like ducts, combustor etc. In case of turbomachinery (the compressor and the turbine components), heat transfer implies that the process is no longer adiabatic and the power consumed or produced cannot be calculated using (adiabatic) component characteristics. Since the effect of heat transfer on the turbomachinery power is not that pronounced and the heat transfer models are based on empirical constants, a fraction of the heat is assumed to be transferred before and the remaining after the process of expansion or compression. This allows for calculating expansion and compression processes using the adiabatic component maps [25].

4.3.2 Modelling assumptions

To model the thermal effects of OP16 using *Heat sinks* (see section 4.2.1), it is important to first identify the heat transfers that are likely to have a significant effect on engine performance.

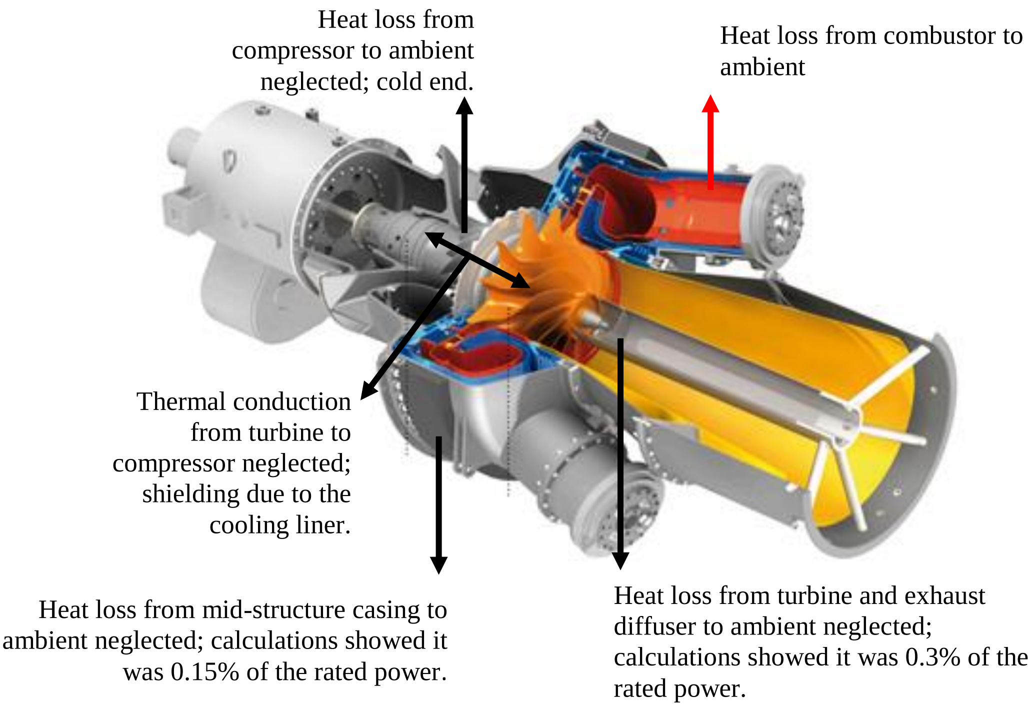

Figure 4.4 Cut-out 3-D view of OP16 describing the various possible heat transfers

The following points discuss the same for (almost) all possible heat transfers within the OP16 and to the ambient outside.

- The heat transfer from the compressor casing to the ambient is neglected as it is the cold end of the engine and hence, the heat loss is small, less than 0.15% of the design shaft power.
- Similar to above, heat transfer from the mid-structure casing to the ambient is neglected. The mid-structure casing lies between the cold and the hot side of the engine and contains the compressor discharge flow before it enters the combustion chamber.
- The conduction heat transfer from the combustor casing to the mid-structure casing is also assumed to be insignificant since both the casings are flushed on the inside with compressor discharge flow and the difference in the casing temperatures is very small.
- The heat loss from the turbine, together with the exhaust diffuser, to the ambient is found to be negligible too with a magnitude of around 0.3% of the design shaft power (based on analytical calculations using fundamental theory of heat transfer).
- The conduction heat transfer from the turbine casing to the compressor casing is shielded by a cooling liner located within the mid-structure of OP16. Hence, it is assumed to be insignificant as well.

- The heat transfer from the hot gas inside the combustor to the ambient via the combustor casing is expected to be considerable and is modelled using the heat sinks in GSP.

Figure 4.4 attempts to give a better understanding of the heat transfers discussed so far.

4.3.3 Model configuration and calibration of heat transfer parameters

The OP16 combustion chamber and the associated gas path flow are shown in figure 4.5. The flame tube forms the gas path wall enclosing the combusted gas within. The combustor casing is the outermost cover of the combustion chamber and is exposed to the ambient on the outside. In between the flame tube and the combustor casing, there is an annular space. This annular space forms the gas path for the compressor discharge flow which then enters the flame tube and undergoes combustion. Thus, the heat transferred from the hot combusted gas to the flame tube is conducted through the flame tube wall and transferred to the compressor discharge fluid flowing around it. A part of the heat goes into increasing the temperature of the flow and the rest is transferred to the combustor casing wall and ultimately transferred out to the ambient.

It is worth pointing out that the heat transfer from the OP16 combustion chamber described here is a simplification of the actual process which is much more complex. All the compressor discharge fluid does not enter the flame tube at its fuel injection point (far right in figure 4.5) but a part of it enters the flame tube before, to form film cooling layers and to cool the dilution zone. Since it is impossible (and not necessarily required for engine performance analysis) to model these effects in a 0-D framework, all these effects have been captured by configuring the model in a way that it matches the performance measurements. Figure 4.6 shows the corresponding thermal network model in GSP.

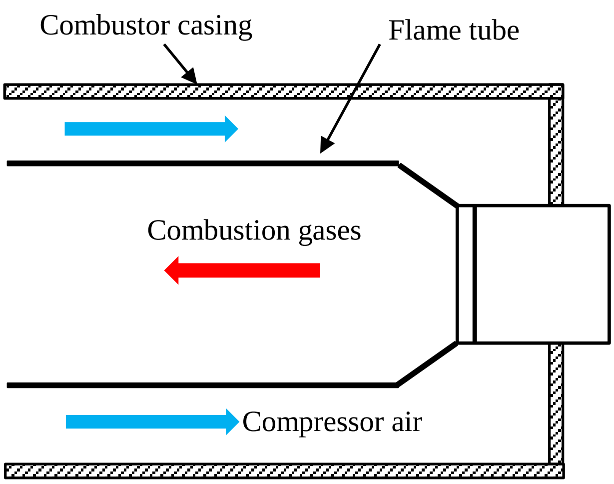

Figure 4.5 Compressor discharge flow path in OP16 (simplified depiction)

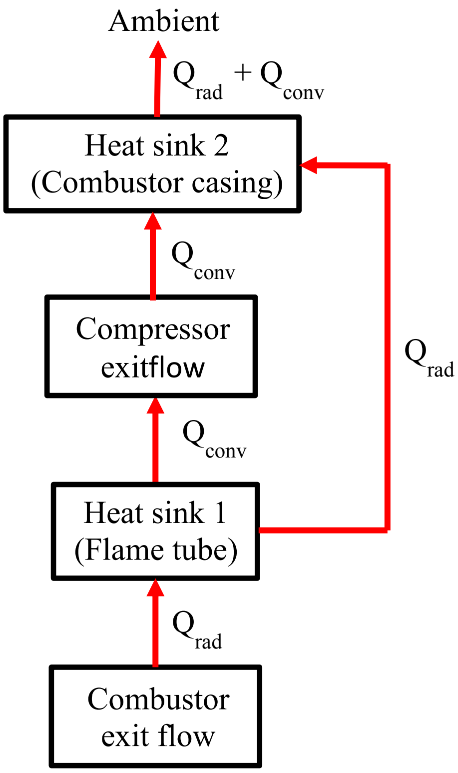

Figure 4.6 Thermal network of the OP16 combustor in GSP

To lump all these effects into one simple model, heat transfer from the gas to Heat sink 1 is assumed to be only by radiation. The associated emissivity value is tuned such that the radiation heat transfer matches the net heat transfer due to radiation and convection at full load conditions. The net heat transfer rates at full load were obtained from OPRA's in-house tool. The tool uses 1-D heat transfer model and includes film cooling effects. The chosen emissivity value, thus, compensates for lower gas temperature and absence of film cooling in the model.

Heat sink 1 is linked to the duct at the compressor exit (see figure 4.3) to simulate convection and radiation heat transfer between the flame tube and the compressor discharge flow. It is also linked to another heat sink, Heat sink 2, which represents the combustor casing. The heat transfer link between Heat sink 1 and Heat sink 2 models the radiation heat transfer from the hot flame tube wall to the casing wall.

The duct at compressor exit is linked to Heat sink 2 to simulate convection heat transfer between the compressor discharge flow and the combustor casing wall. Since, the temperature of the compressor exit flow is relatively lower, radiation heat transfer is assumed to be negligible.

Heat sink 2 is linked to the external ambient conditions to simulate free convection and radiation heat transfer from the combustor casing wall to the environment.

There are four combustion cans in the actual OP16 engine. In the thermal network model, all the four cans are lumped into one large combustion chamber and the associated model parameters (like combustor geometry) are scaled accordingly.

4.4 Design point modelling

As discussed previously in section 2.1, design point simulation is required to be performed before the off-design behaviour of an engine can be simulated. The design point simulation serves as the cycle reference point for off-design simulations.

The first step in design point modelling is selection of an operating condition that would serve as the model design point (i.e. the cycle reference point). It is ideally the engine design point but an actual engine (during its development) might have multiple design points corresponding to specific performance requirements. It is a common practice to choose any high power operating point as the model design point. Once the model design point is defined, the model is matched to the known data at this operating point by tuning the model design parameters. The model design parameters include mass flows, component efficiencies, pressure-ratios, turbine inlet temperature etc. Thus, design point modelling amounts to selection of suitable model parameters that simulate the actual engine performance at a chosen reference point.

Table 4.1OP16 operating conditions used to define the cycle reference point

Operating condition	Value
Ambient temperature	288.15 K
Ambient pressure	1.01325 bar
Ambient humidity	60%
Load	Full load; max. allowable turbine inlet temperature

Table 4.2OP16 given model design model parameters

Given design point parameters	Value
Fuel LHV	46 MJ/kg (Natural gas)
Rotor speed	26016 rpm
Inlet loss	0
Exhaust loss	0

The OP16 model design point is chosen to be the full load operating point with diesel as fuel, nominal rotor speed, no installation effects and standard day ambient conditions.

The operating condition corresponding to the chosen OP16 model design point is listed in table 4.1. The design parameters that are fixed (or given) are listed in table 4.2. It could be difficult to obtain measured performance data for the exact conditions defined in these tables. For example, it is practically impossible to have zero inlet and exhaust losses and the ambient conditions in the available data may not be exactly the same as those in table 4.1. These deviations may not be too significant but it is good to realize that they exist. From the available measured performance data (section 4.1), the data set closest to the chosen model design point is used to find the suitable model design parameters. This data comes from the performance test of a newly built engine tested in ambient conditions close to standard day at the OEM's test-cell. Hereafter this particular measured (off-design) performance data set as referred to as '*design point calibration data*'. An overview of the testing condition of the *design point calibration data* is given in table 4.3. Data from one engine is assumed to be representative of the OP16 engine behaviour.

Tuning of the model design point parameters is done as follows: Design point performance is simulated using a realistic first guess for the model design parameter set. After this, preliminary engine model parameters are generated according to this design point calculation, off-design simulation is run corresponding to the operating condition of the measured performance data set (*the design point calibration data*). The resulting off-design performance simulation is then compared to the measured performance. If the match is not good enough, the model is switched back to the design mode and the design parameters changed and the whole exercise is repeated until a satisfactory match is obtained.

The off-design simulation requires component maps and the location of the design point on them (section 2.2.1).The component maps (for both the turbine and the compressor) are obtained from the OEM's database. The maps are assumed to be accurate for this exercise. The location of the design point in the component maps is determined by 'map design beta' and 'map design rotor speed' in GSP. These parameters can also be tuned along with the model design parameters to obtain a good match. It is interesting to note that when the simulations are close to the measurements, the full load off-design operating point nearly overlaps the design point in the compressor and the turbine maps. A list of the model design point parameters that are tuned and the simulated parameters that are matched is given in table 4.4.

The thermal network model parameters (like component geometry, fluid properties, heat transfer coefficients etc.) are taken from the OEM's database and fundamental heat transfer theory. They are further tuned to match the combustor casing wall temperature measurements and the heat transfer calculations from OPRA's in-house tool at the design point operating conditions. The process is called 'synthesis by analysis'. The resulting set of model parameters is given in table 4.6. Since parameters specific to component geometry are classified as OEM proprietary information, they have not been listed here. Modification of the thermal model network parameters changes the simulated engine performance and might require adjustments to the model design parameters and vice versa.

The selected model design parameters (table 4.4) and thermal network model parameters (table 4.6) together result in a well matched design point model of OP16.

Table 4.3 Testing condition in measured performance data set used for model design point calibration

Testing condition	Value
Ambient temperature	289.8 K
Ambient pressure	1.012 bar
Ambient humidity	58%
Load	Full load

Table 4.4OP16 model design point parameters

Tuned parameters	Matched parameters
Inlet mass flow	Engine shaft power
Compressor	
Efficiency	Exit temperature
Pressure ratio	Exit pressure
Compressor exit bleed for turbine inlet cooling	
Heat picked up by cooling flow before mixing	
Turbine	
Efficiency	Exit temperature
	Exit pressure

Table 4.5Location of the model design point relative to the component maps

	Compressor map	Turbine map
Map design rotor speed	1	1
Map design beta value	0.6	0.5

Table 4.6 OP16 thermal network model parameters

Thermal network model parameter	Value
Duct at combustor exit - Heat sink 1	
Hydraulic diameter [m]	0.324
Conductivity of gas [W/m K]	0.158
Nusselt number [-]	0
Material conductivity [W/m K]	27
Gas emissivity [-]	0.55
Heat sink 1- Heat sink 2	
Heat transfer coefficient [W/ K]	25
Duct at compressor exit – Heat sink 1	
Hydraulic diameter [m]	0.2
Conductivity of gas [W/m K]	0.043
Nusselt expression [-]	$0.023 \cdot Re^{0.8} \cdot Pr^{0.4}$
Material conductivity [W/m K]	27
Material emissivity [-]	0.7
Duct at compressor exit – Heat sink 2	
Hydraulic diameter [m]	0.2
Conductivity of gas [W/m K]	0.043
Nusselt expression [-]	$0.023 \cdot Re^{0.8} \cdot Pr^{0.3}$
Material conductivity [W/m K]	24
Gas emissivity [-]	0
Heat sink 2 - Ambient	
Hydraulic diameter [m]	0.48
Conductivity of air [W/m K]	0.025
Nusselt number [-]	2760
Material conductivity [W/m K]	26
Gas emissivity [-]	0.5

4.5 Off-design modelling

Off-design performance simulations require knowledge of component characteristics to estimate the component efficiencies at operating conditions deviating from the design point. In GSP, the component characteristics are expressed in the form of component maps. Once the simulations match the measured data at the model design point, the simulations at off-design operating points may deviate from the measurements depending on the accuracy of the component maps. Obtaining the component maps that best represent the turbo-machinery being modelled is the key to off-design modelling.

The operating range of the turbine is usually small. In the case of OP16, the turbine is choked across (almost) the entire operating envelope. Also, the part-load performance behaviour (in terms of specific fuel flow) is directly dependent on the compressor efficiencies read from the map. As such, the compressor maps are usually considered more critical to off-design performance simulations.

Component maps available in the OEM’s database provide a good starting point for off-design modelling of OP16 on GSP. The part-load performance data (section 4.1) of an engine tested on diesel in ambient conditions close to the standard day is used for estimating the maps. The data set is from the same engine test as used for the design point modelling.

Component map estimation begins with the compressor map. The compressor map basically provides the correlation between the corrected mass flow and efficiency and corrected mass flow and corrected speed, along an operating line. As discussed in [12], corrected mass flow and efficiency correlations should be dealt with first as they are more relevant thermodynamically. The measured compressor efficiency-corrected mass flow relation is compared with the one in the compressor map. Based on the comparison, the map is adjusted (a little) to match the measured data. Since the ambient conditions in the measured data correspond quite closely to the design ambient conditions and because OP16 is a constant speed engine, the measured part-load operating line (from zero load to full load) follows the corrected speed line, $N_c = 1$. This means that the corrected mass flow and efficiency correlation in the map is modified for this particular speed line. Based on this modification, the mass-flow efficiency relationship for other speed lines is scaled accordingly.

Once the corrected flow-efficiency correlation is satisfactory, one could try to adjust the corrected flow-corrected speed correlation. However, this was not needed here since the adjusted corrected flow-efficiency correlation alone resulted in accurate compressor performance predictions. The scaled efficiency-corrected flow characteristics are compared with the original characteristics in figure 4.7.

In case of turbines, since the turbine inlet conditions are hardly ever measured, it is more difficult to accurately estimate the turbine maps. Nevertheless, the turbine entry temperature and, thus, the turbine efficiency can be estimated using energy balance.

The estimated turbine efficiencies are used to adjust the efficiency-pressure ratio correlation in the map so that the simulated turbine exit conditions match the measured data.

Since the maps available in the OEM database are good estimations of the OP16 turbo-machinery characteristics, only small adjustments to the component maps led to quite accurate results as can be seen in the next section.

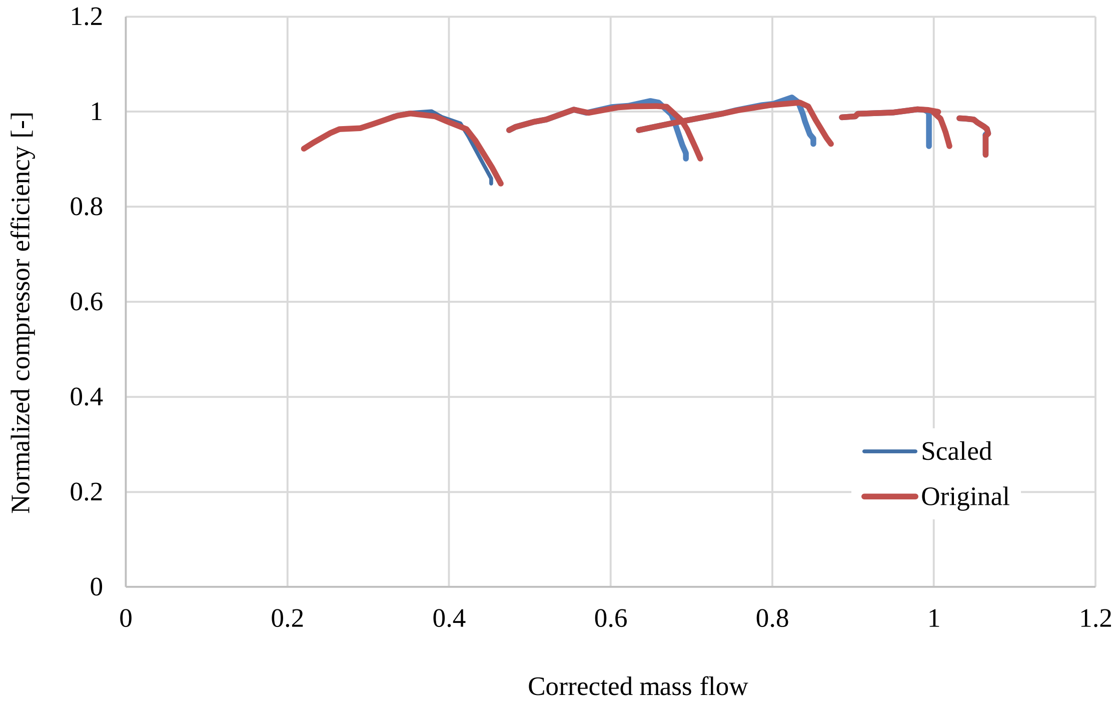

Figure 4.7 Scaled efficiency-corrected flow characteristics of the OP16 compressor for various speed lines

4.6 Validation of the steady-state model

To assess the accuracy of the created OP16 model, the measured performance data of the OP16 engine is compared to model simulations across the entire load range for varying ambient conditions. The model is given as an input and the ambient conditions, inlet loss, exhaust loss and the power level in the form of fuel flow are specified. This input data is taken from the measured performance data set under comparison. The performance data sets used here come from engine testing on diesel fuel. The simulated compressor and turbine behaviour and the shaft power are compared to the measured values.

It is clear from validation results that the simulated component performance matches well with the measured data. The relative errors are mostly within 1.5% of the measured value. The accuracy is highest near the model design point, i.e. full load and standard day ambient conditions. The errors slightly increase as the operating point moves to off-design conditions, i.e. lower loads and ambient conditions deviating from standard day. Nonetheless, the error in component performance simulation for standard day ambient conditions essentially falls within measurement accuracy limits, across the entire OP16 load range. For ambient conditions deviating from standard day, the error may increase but still remains fairly close to the measurement accuracy limits.

Surprisingly, the simulated and the measured shaft power do not seem to agree with each other as the load decreases. The error at the full load operating point is almost zero because the design point was modelled using the full load operating point data. At part-load, the model over-predicts the shaft power and this error increases with decreasing engine load as can be distinctly seen in figure 4.13. It seems that the engine is losing power somewhere in the cycle which is not accounted in the model. It could also be due to a fault in one or more measurements. The next section discusses the potential reasons of the apparent loss in engine power.

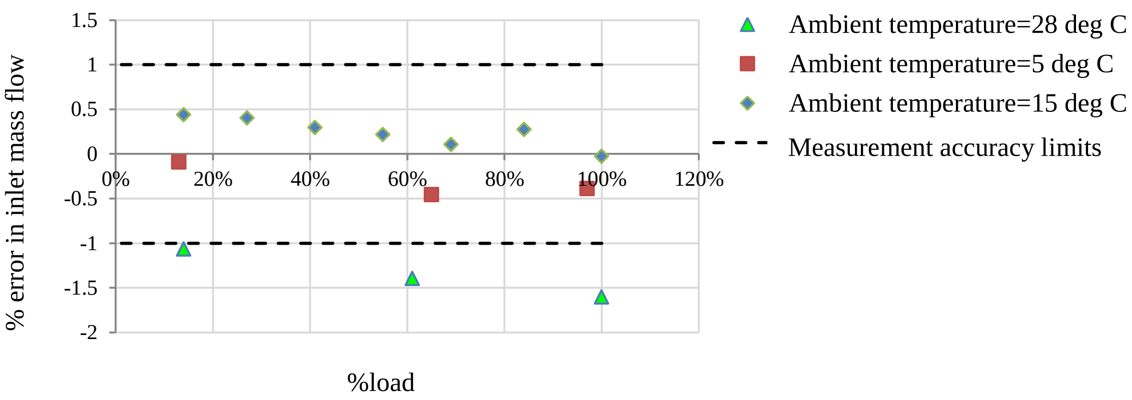

Figure 4.8 Relative error in inlet mass flow between the model simulation and the measured data across the OP16 load range.

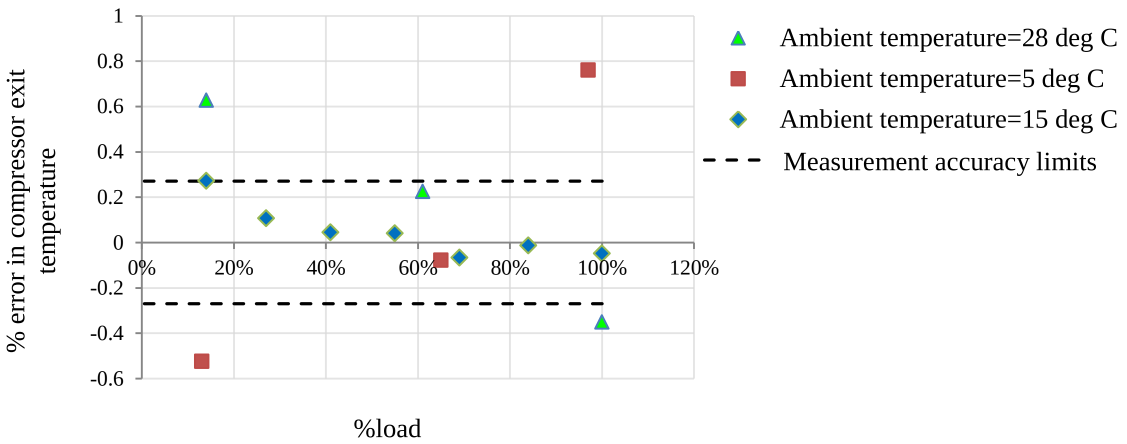

Figure 4.9 Relative error in compressor discharge temperature between the model simulation and the measured data across the OP16 load range

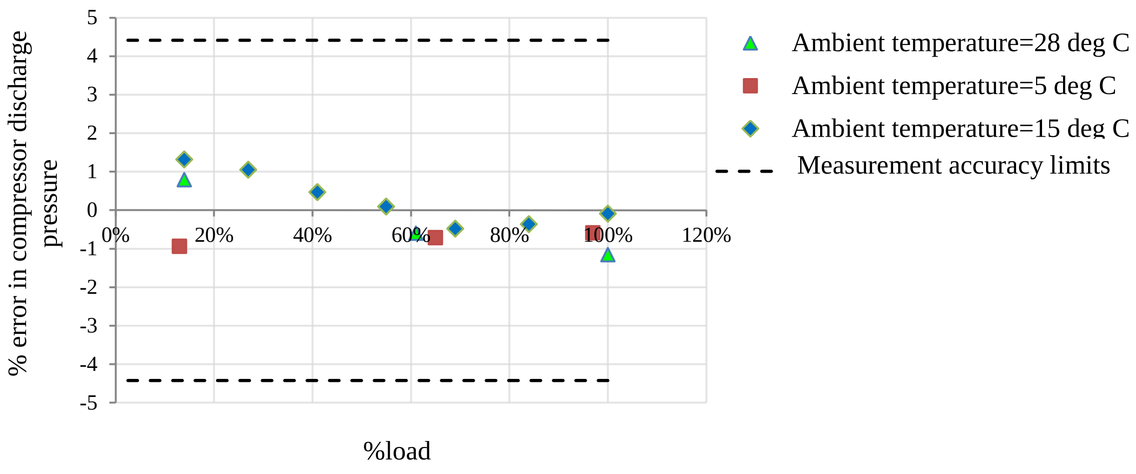

Figure 4.10 Relative error in compressor discharge pressure between the model simulation and the measured data across the OP16 load range

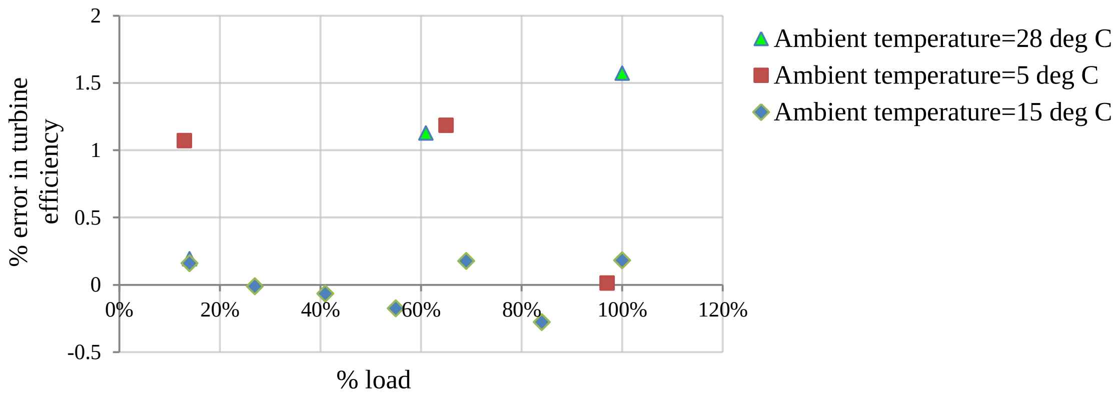

Figure 4.11 Relative error in turbine efficiency between the model simulation and the measured data across the OP16 load range

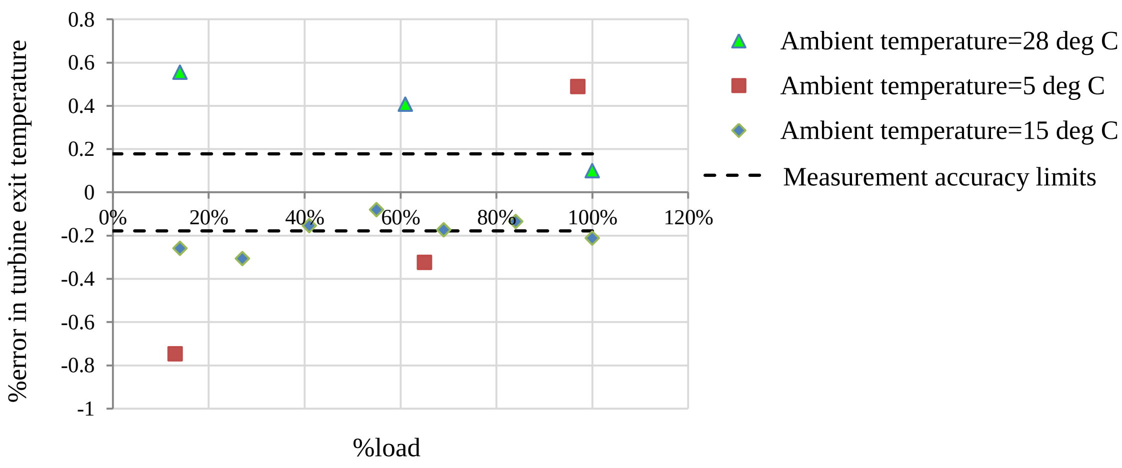

Figure 4.12 Relative error in exhaust gas temperature between the model simulation and the measured data across the OP16 load range

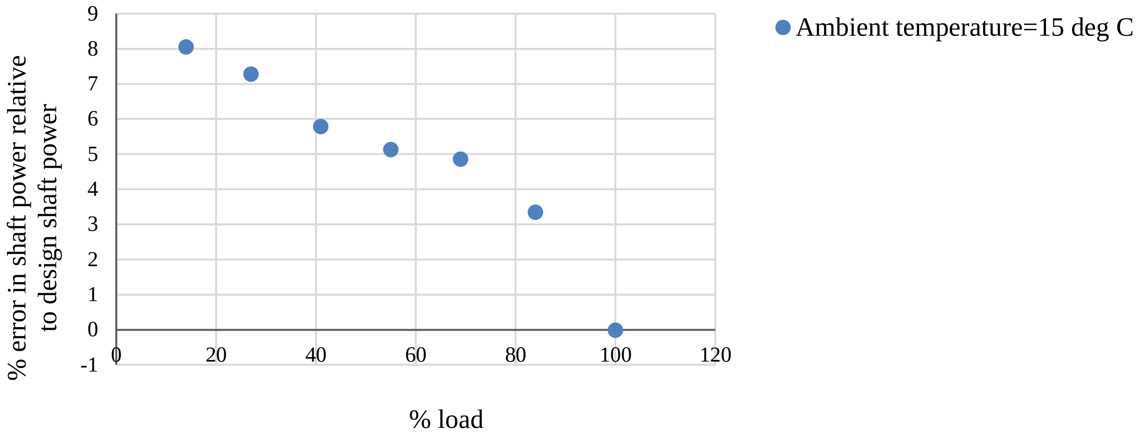

Figure 4.13 Error in shaft power relative to design shaft power between the model simulation and the measured data across the OP16 load range

The model shaft power predictions are within an accuracy of 2.5% near the full load condition. In order to correct for the power loss at lower loads, a linear correcting polynomial is applied to the simulated power within GSP. The polynomial is a function of the power level setting (figure 4.14).

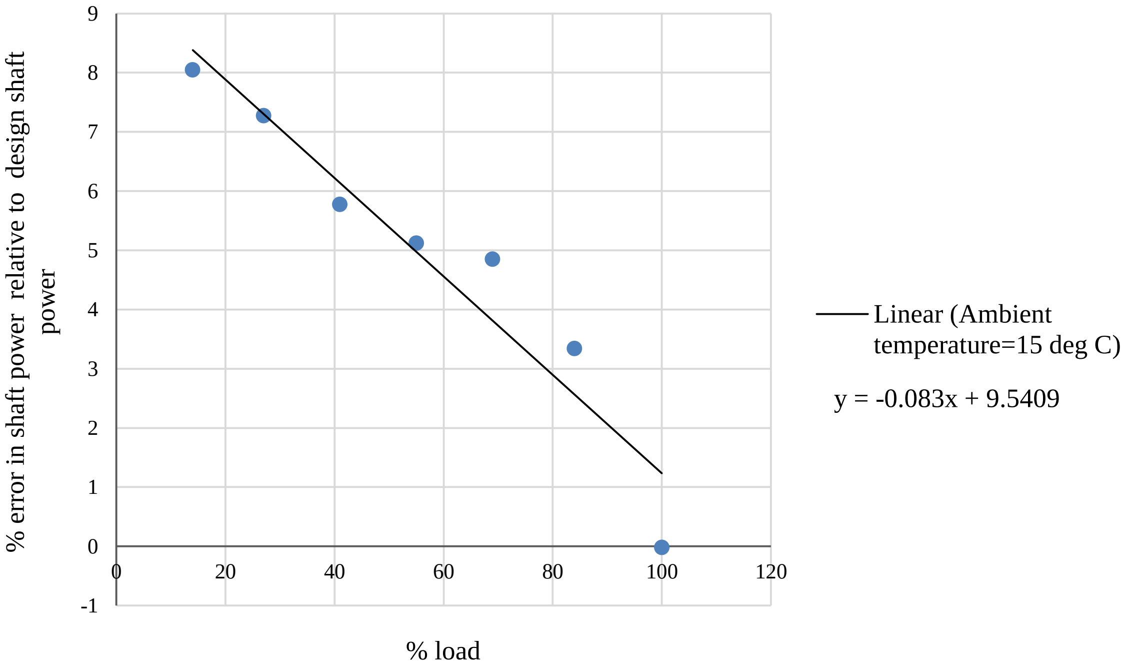

Figure 4.14 Correction of simulated shaft power in GSP using a linear correcting polynomial as a function of power level setting

4.7 Investigation of loss in shaft power

As discussed in the previous section, the OP16 model over-predicts the engine shaft power. Since the simulated component behaviour matches well with the measured data it is unlikely that any inaccuracy in the component models can result in error (in shaft power) as large as shown in figure 4.14. The steady-state heat loss is a small fraction of the error observed and it is unlikely to amount to as much as 8% of design engine shaft power. Moreover, thermal losses are bound to decrease with decreasing engine load since the temperature of the hot gas inside the combustion chamber decreases. This makes it difficult and at the same time intriguing to get to the source of the observed error between the measured and the predicted shaft power.

To put things under a broad perspective, the following could be the probable causes of engine power loss:

- Losses in the generator

  The shaft power is calculated back from the measured generator load by using generator efficiency curves. It is possible that the generator is a source of power loss. The generator efficiency curves might be over-predicting the actual generator efficiency.

- Faulty measurements

  One or more measurements could be incorrect because of a faulty sensor or incorrect sensor calibration or inappropriate sensor location etc. resulting in an incorrect cycle computation and hence, an apparent loss of power.

- Additional losses in the engine

  There could be losses in the engine that have not been accounted for in the model. For example, thermal losses, gearbox losses, leakages etc.

In order to investigate the effect of the generator on delivered power, it is essential to isolate the engine from the generator. The best solution is to use a torque meter and measure the shaft power right after the engine, thus excluding the gearbox and the generator from the measurements. Due to unavailability of a torque-meter, measured data from an engine test where the engine is run decoupled from the generator, i.e., run at idle, is compared to the model simulations. The per-centage error in shaft power relative to the design power is found to be around 11% which is consistent with the errors found in the earlier simulation results (figure 4.13). Basically, the model still predicts non-zero shaft power where the engine is actually running at no load. This indicates that the generator is not the source of power loss. This also implies that the shaft power measurement is not faulty.

To verify if the test setup is the cause of lower value of the measured power, performance data of OP16 from a different test cell is compared to the model simulations. The other test-cell is referred to here as '*Test-cell 2*'. It is observed that the simulated shaft power deviates from the measured data by magnitudes similar to what was found before in the previous section. Again, the error increases at part-load conditions. But unlike the previous observations, the simulated exhaust gas temperature doesn't match very well with the measured data as can be seen in figure 4.15. Further, the error increases at part-load conditions. The simulated inlet mass-flows and the compressor exit conditions are observed to be consistent with the measured data.

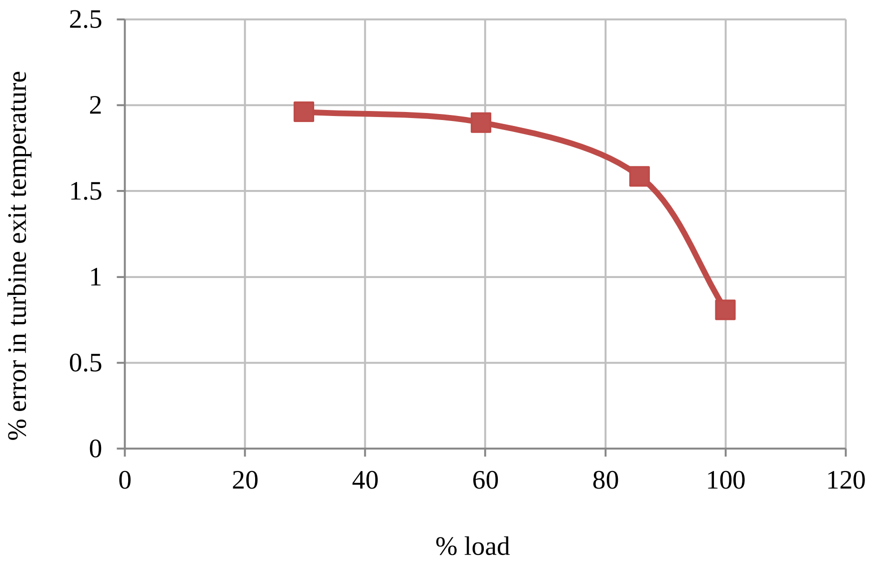

Figure 4.15 Relative error in turbine exit temperature between the model simulation and the measured data from Test-cell 2

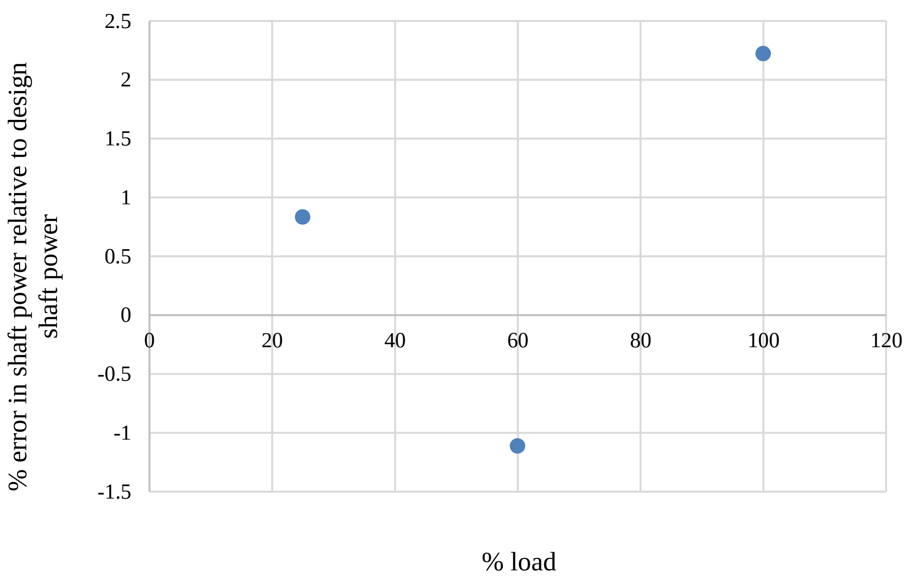

Figure 4.16 Error in shaft power relative to design shaft power between the model simulation and the measured data across the OP16 load range (Stavanger test-cell)

Adjusting the turbine efficiency to better match the simulated turbine exit temperature to measured data, results in significant reduction in shaft power errors as can be seen in figure 4.16. The error between the measured and the predicted shaft power is considerably lower. In fact, the data in figure 4.16 looks like a scatter and there is no trend of increasing power loss with decreasing engine load, unlike that seen in figure 4.13.

The findings suggest that there are some differences between the test-cell in Hengelo and test-cell 2. It seems that there is a fault somewhere in the test setup in Hengelo which results in inconsistency between the model simulations and the measured data (from Hengelo test-cell) at lower loads. This also indicates that the source of error between the simulated and the measured shaft power is not due to additional unaccounted losses in the engine.

Assuming that the ambient condition measurements are accurate, the suspicious measurements include; the inlet mass flow, the compressor exit conditions, the fuel flow and the turbine exit temperature. Since modification of model turbine efficiency results in well-matched (to the data from Test-cell 2) simulations, including the shaft power, one (or more) of the parameters influencing the energy balance of the turbine component are likely to be faulty. Assuming that the heat transfer from (or to) the turbine is negligible and that the compressor exit condition measurements are accurate, this makes the fuel-flow, the inlet mass flow, and the turbine exit temperature measurements, the prime suspects.

To account for an error in simulated shaft power as high as 8% (of design shaft power); the model design mass-flow requires to be changed by roughly 32% of its original value. This change is too large to be justified and hence, it seems unlikely that the inlet mass-flow is the cause of the error in simulated shaft power. On the other hand, a change of around 7% in model fuel flow or 2% in turbine exit temperature (by changing the model off-design turbine efficiency by 4%) alone is sufficient to account for an error of 8% in engine shaft power. The off-design turbine isentropic efficiency in the turbine performance maps is derived from (among others) the measured fuel flow and turbine exit temperature (by using energy balance) and any fault in these measurements results in inaccurate off-design turbine efficiency. Thus, it is possible that either one or both of the fuel flow and the turbine exit temperature measurements have larger uncertainties and/or errors.

Getting to the root cause of the error in simulated shaft power is quite an elaborate task and is not carried out as a part of this project. The study undertaken in this work simply points out the probable source(s) and it is up to the OEM to look into it further. For the purpose of this project, the OP16 model is deemed accurate near the full-load operating condition and at part-load, the power correcting polynomial discussed in section 4.6 can be used for satisfactory gas turbine performance simulation and analysis.

5 Transient performance modelling

A gas turbine engine operates in a transient regime whenever one or more operational conditions vary with time. GSP simulates the transient behaviour of a gas turbine engine based on steady-state component characteristics, using the laws of conservation (section 2.4). Unlike the off-design steady-state simulations, the time-derivative terms in the conservation equations are non-zero for transient calculations, increasing their complexity. Owing to the quasi-steady-state nature of transient simulations, the transient performance model of an engine in GSP requires a good steady-state model to begin with. The steady-state model can be extended to a transient model by selection of suitable model parameters to best represent the engine-specific details like rotor moment of inertia, and component geometrical and material properties. Usually, rotor inertia effects alone are sufficient to capture the transient behaviour of a gas turbine engine but heat soakage effects and gas dynamics could be significant depending on the engine size and design. The physical phenomena associated with transient operation of a gas turbine engine are discussed in section 2.3.1. One of the objectives of transient performance modelling of OP16 is to establish which of these transient effects play a significant role in its performance.

This chapter begins with a description of measured data used to create and validate the transient model of OP16. Next the model configuration is explained followed by model validation against measured performance. The chapter ends with a few concluding remarks on the OP16 transient performance model.

5.1 Available performance data

The measured data sets used for building up and validating the transient performance model of OP16 comes from the OEM's test-cell in Hengelo, The Netherlands. The performance data from the OP16 engine tests discussed in section 4.1 represents slow engine acceleration from idle to full load. At each load step, the shaft speed drops before the control system reacts to bring it back to the nominal state by regulating the engine fuel flow. These data sets depict system response to control variables (like fuel flow) and are deemed useful to design basic control philosophy and fuel flow schedules. Data corresponding to load steps near the full load operating region are employed to set up the transient performance model of OP16. The full load operating point is the most critical in case of occurrence of a *load shed* (see chapter 6). Moreover, the steady-state OP16 model is more accurate near the full load operating region (see section 4.6). Thus, model validation at full load is expected to result in a more accurate calibration of transient model parameters. The operational conditions corresponding to this transient manoeuver are given in table 5.1. The load and fuel flow variations with time are shown in figure 5.1 and 5.2, respectively. The time step for data acquisition is 1 millisecond.

Table 5.1 Testing conditions corresponding to measured transient data-set used for model set-up and validation

Testing condition parameter	Value
Ambient temperature	289.8 K
Ambient pressure	1.012 bar
Ambient humidity	58%
Load range	90%-100% of full load

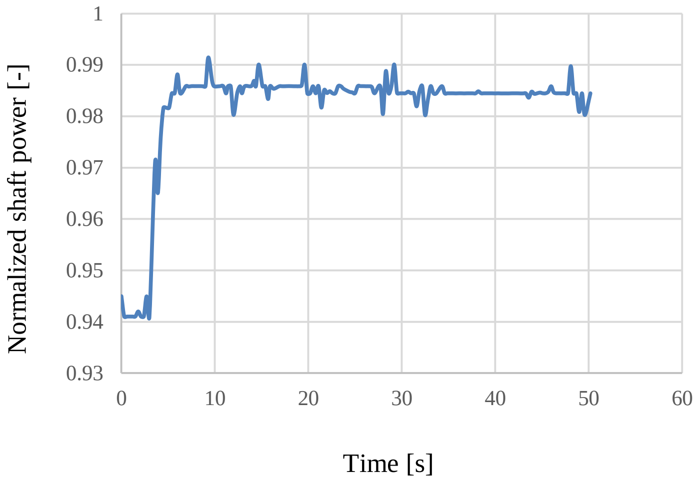

Figure 5.1 Shaft power measurement for a load step from 95% load to full load

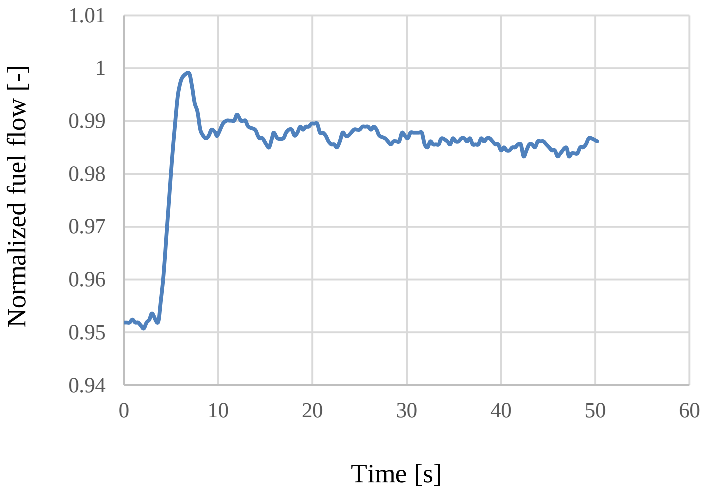

Figure 5.2 Flue flow measurement for a load step from 95% load to full load

5.2 Model configuration

Besides rotor inertia effects, GSP allows modelling of heat soakage effects and volume dynamics for each gas path component and control system dynamics in case control system models are added. To build the transient model of OP16 the following effects are considered:

5.2.1 Rotor inertia effects

To calculate rotor speed acceleration (or deceleration) rates, the total moment of inertia of all elements connected to the shaft, namely, the compressor, the turbine, the gearbox, the generator and the low speed coupling is specified in the turbine component. GSP employs equation 2.4 to compute the rotor speed acceleration using the specified spool inertial moment, *I*. The equation has been reproduced here for sake of completeness.

$$I.\frac{d\omega}{dt}.\omega = \Delta PW$$

5.2.2 Volume dynamics

Volume dynamics has been included for two components, the combustion chamber and the turbine together with the exhaust diffuser. The cold side of the engine, i.e. the inlet and the compressor, doesn't experience considerable changes in mass flow with changes in engine load. This can also be seen in the part-load performance of OP16 (section 4.6). Moreover, the volume of the compressor is comparatively smaller. The associated volume dynamics is assumed to be negligible and hence has not been considered here. The volume of the OP16 combustion chamber and the turbine (along with the exhaust diffuser) is relatively larger. The four combustion cans of OP16 have been lumped into one big combustion chamber of equivalent volume. In the model, the combustor volume is specified in a duct component at the combustor exit (see figure 4.1) to avoid numerical convergence problems. GSP uses equation 2.2, reproduced below, to simulate the volume dynamic effects for a gas path component based on the user-specified volume. The equation is based on the assumption that the flow through the component is adiabatic.

$$\frac{dM_v}{dt} = \frac{V_{comp}}{\gamma RT}.\frac{dp}{dt}$$

5.2.3 Heat soakage effects

The heat soakage effects have been implemented for the combustion chamber and the turbine. Owing to limited variation in gas temperature at the cold side of the engine, no heat soakage is modelled for the inlet and the compressor. To model heat soakage effects for the combustion chamber, the *heat sink* elements used to model the steady-state combustor heat transfer (Section 4.3.3) are extended to simulate dynamic heat transfer by specifying mass and specific heat capacity representative of the structure they depict. The corresponding equations used by GSP are given as equation 2.5 and 2.6, reproduced below:

$$Q_{hs} = U.A.(T_{hs} - T_g)$$

$$M_{hs}.c_{phs}.\frac{dT_{hs}}{dt} = \sum Q_{hs}$$

In order to model heat soakage effects due to the structure around the turbine, the heat soak functionality of GSP is used. Heat soak allows simulation of heat transfer from hot gas to (or from) the surrounding material during transients. Unlike *heat sinks*, the *heat soak* functionality implements a simplistic heat transfer model based on conduction and convection. It doesn't allow simulation of heat transfer between the gas path components and to the ambient. The heat transfer equations are similar to equation 2.5 (reproduced above) except that there is just one *heat sink*, namely, the gas path component material and computation of the overall heat transfer coefficient is much simpler. Since it was assumed that heat flow from the turbine to the ambient or to the compressor is negligible (section 4.3.2), using *heat soak* is deemed sufficient to compute the transient heat flow between the hot gas and the component material in contact with it. The heat soakage effects in the turbine are modelled for the turbine blades (the impeller and the exducer) and the turbine hub.

The user-specified model parameters that collectively determine the transient behaviour of the OP16 are derived from engine-specific details and further tuned to match the simulations with measured data. The transient model parameters are listed in table 5.2. The corresponding values are considered OEM proprietary information and are not included here.

Table 5.2 Transient model parameters of the OP16

Turbine	Combustion chamber
Spool inertial moment [kg m^2]	Volume [m^3]
Volume [m^3]	Effective mass of flame tube [kg]
Effective contact surface area [m^2]	Effective mass of casing [kg]
Film coefficient at design point	
Effective mass [kg]	
Specific heat component material [J/kg K]	
Effective length heat flow transport [m]	
Material thermal conductivity [W/m K]	

5.3 Validation of transient performance model

As discussed in section 2.3, gas turbine transient performance and control system dynamics are inseparable. The objective of this work is to model the physical transient behaviour of the OP16 gas generator (excluding the control system dynamics). But, in order to simulate meaningful results which can be verified against measured data, it is essential to take into account the dynamic effects of the control system. In the performance data used for the OP16 transient model verification, the fuel flow is the sole controlled input to the gas turbine, governed by the control system to maintain the nominal engine speed. The measured variation of fuel flow with time (figure 5.2), thus, includes the control system dynamics. Using this measured fuel flow as the model input eliminates the need of modelling the control system dynamics while obtaining meaningful results. The OP16 transient model parameters can then be adjusted to match the measured data to result in an accurate representation of the physical transient effects of the OP16 gas generator.

To verify the transient performance model of OP16, the model simulations are compared to the measured data for a load step from 95% load to full load. To run a model in transient mode, one or more model inputs must be specified as a function of time. The measured fuel flow and shaft power are provided as time-varying inputs to the model and ambient conditions and installation effects (the inlet and the exhaust loss) are given as time-independent inputs. The fuel flow and shaft power measurements are approximated, as shown in figure 5.3 and figure 5.4, to provide a smoother input to the GSP model. The load input to the model is offset by 2% of the measured full load in accordance with model power correction discussed in section 4.6. The offset lies within the measurement accuracy of engine shaft power. The model rotor speed is left at '*free state',* meaning, it is one of the model outputs. The simulated rotor speed is compared with the measured data along with other engine performance parameters.

From figure 5.6, it is clear that the model predicts the transient behaviour of rotor speed quite accurately. The same is true for engine inlet mass flow as can be seen in figure 5.5.

The simulated compressor discharge temperature and turbine exit temperature show a peaky transient behaviour which cannot be traced back to the measured data (figure 5.7 and 5.9). Since the temperature measurements come from thermocouples, it is due to a lag in thermocouple response time. As is widely known, thermocouples *always* lag behind the actual gas temperature during transient. To better understand this effect, a first order lag function with a time constant of 3 seconds and a gain of 1 is applied to the simulated exhaust gas temperature. The resulting variation in temperature is shown in figure 5.10. It is clear that the application of a lag function to the simulated exhaust gas temperature makes it look quite similar to the measurements. The reasoning can be extended to compressor discharge temperature.

The simulated compressor discharge pressure seems to have a slight offset from the measured pressure but the error is only about 0.6% of the measured value which is well within the accuracy limits of the measurements. Moreover, the offset seems to be coming from (small) inaccuracies in steady-state prediction before the load step is made. The simulated transient behaviour of compressor exit pressure during the load step clearly follows the measured trend as can be seen in figure 5.8.

From the validation results, it is safe to conclude that the model well predicts the actual OP16 transient behaviour due to changes in power level setting at a synchronous speed. The model helps in assessing the significance of physical transient phenomena that influence the transient behaviour of OP16. It is found that the rotor inertia effects dominate the transient response of the engine. The heat soakage effects and volume dynamics remain negligible in comparison. The heat soakage in the combustion chamber is completely neglected as can be seen from table 5.3. It is good to know that though OP16 is a small gas turbine engine but it is still not small enough to have significant thermal effects on the transient behaviour.

The model can also be used to simulate the response of engine performance parameters (that need to be regulated), like rotor speed, to step changes in controlled inputs like fuel flow. The resulting data is useful to evaluate the associated time constants and process gains which make a good starting point for designing a controller.

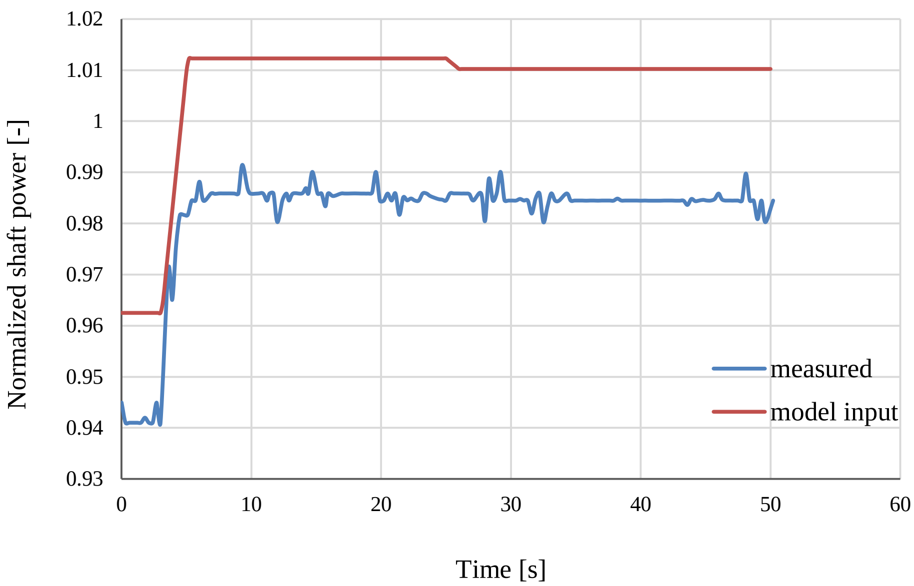

Figure 5.3 Shaft power model input compared to measured shaft power

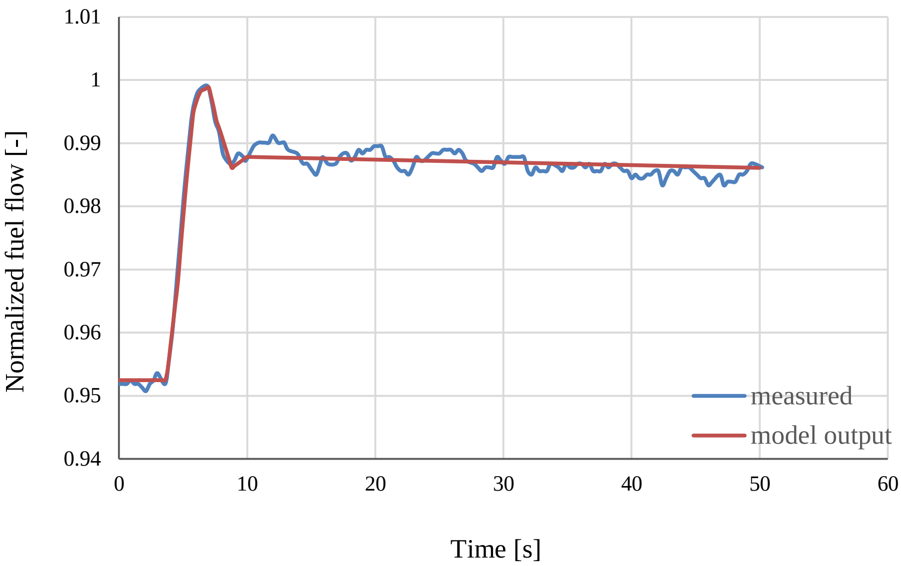

Figure 5.4 Fuel flow model input compared to measured fuel flow

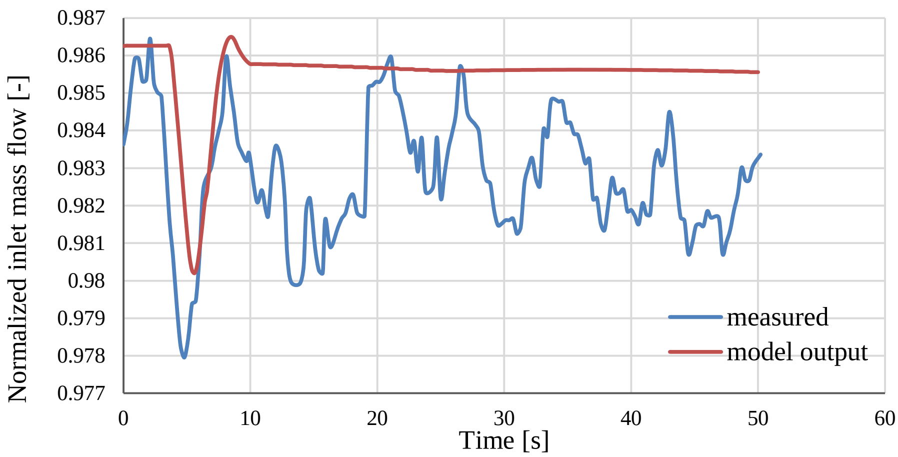

Figure 5.5 Variation of simulated inlet mass flow against the measured data

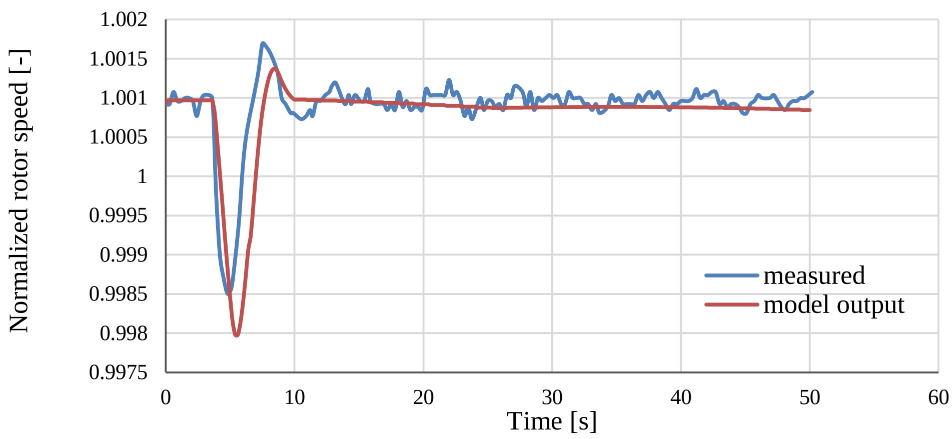

Figure 5.6 Variation of simulated engine rotor speed against measured data

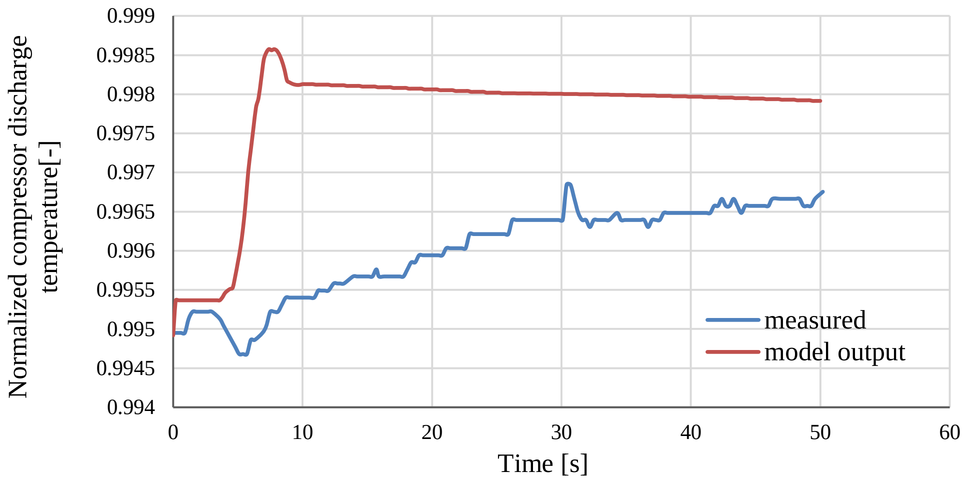

Figure 5.7 Variation in compressor discharge temperature against measured data

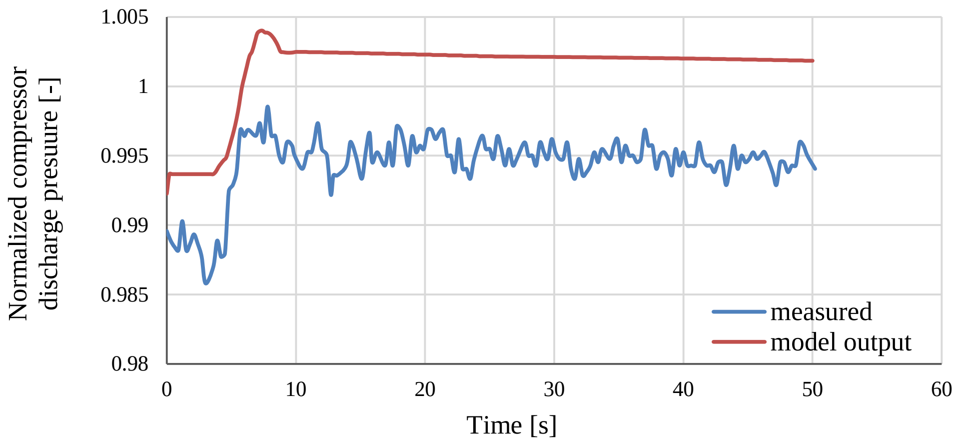

Figure 5.8 Variation in simulated compressor discharge pressure against measured data

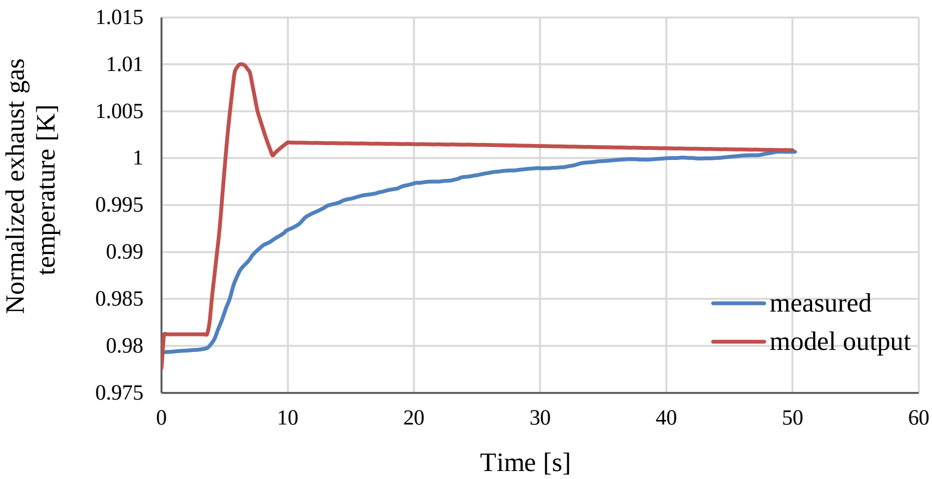

Figure 5.9 Variation in simulated exhaust gas temperature against measured data

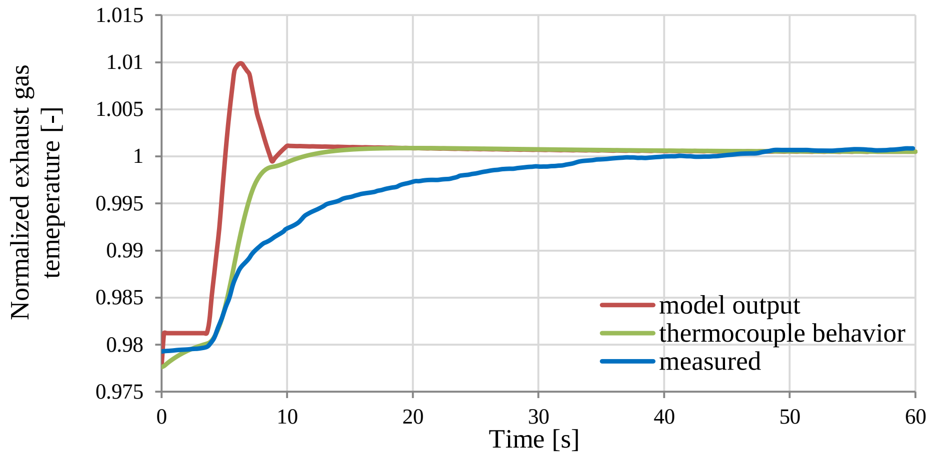

Figure 5.10 Variation in simulated exhaust gas temperature against measured data

6 Rotor over-speed analysis during load shedding

Operation in transient regime presents additional operability concerns that do not arise when a gas turbine is running at steady-state. Transient working lines of a compressor deviate considerably from their steady-state location. For example, in case of a single spool engine running at synchronous speed, the operating point moves closer to the surge line during engine loading. This is accompanied by a decrease in rotor speed until the control system reacts by increasing the fuel flow. During a transient manoeuvre it is important to make sure that the working line does not cross the surge line in order to prevent the compressor from stalling or surging. The transient phenomena such as heat soakage or volume dynamics might also have an impact on the transient working line excursions. Combustor stability is another operability concern during transient operation of a gas turbine. Weak extinction during engine decelerations and rich extinction during engine starting must be avoided. Such undesirable events are prevented from occurring by suitable engine and control system design.

For power generation gas turbines, occurrence of *load shed, or load shedding,* is one of the transient manoeuvres of concern. Load shed refers to sudden loss of generator load, caused by an electrical fault or by exceeding a critical system parameter initiating an emergency shut-down. In case it happens, the fuel flow is required to be reduced quickly to avoid rotor over-speed. Load shed is particularly critical when the engine is running at full load since it results in higher rotor over speeds due to a larger power unbalance. Besides, a load shed might move the compressor working line closer to the surge line if volume dynamic effects are large enough (See section 6.4). This is especially dangerous for the full-load operating point since it is already close to the surge-line.

This chapter analyses rotor over-speed during a load shed in case of OP16 gas turbine engine. The influence of fuel heating value, fuel valve closing time and combustor volume is looked into. The OP16 transient model developed in the last chapter is applied to simulate this manoeuvre.

6.1 Model input

To simulate the load shed, generator load and fuel flow are provided as transient input. Rotor speed is made a *free state*. The load decreases from 100% to 0 in a time-step of 1 millisecond. Fuel flow decreases from the full load value to zero, linearly, depending on the valve closing time which ranges from 0.1 second to 1 second. The ambient conditions correspond to standard day conditions.

6.2 Effects of varying fuel LHV

The rotor over-speed is analysed for fuels with varying calorific values. The details of the simulated fuels are given in table 6.2. Load shed manoeuvre is simulated for different fuels by providing model inputs as per section 6.1. An overview of the operating condition parameters is given in table 6.1.

The power level setting in all cases has been determined by turbine entry temperature where full load operating point corresponds to maximum allowable turbine entry temperature. In case of low calorific value fuels, the fuel mass flow required to obtain a certain turbine entry temperature is higher. There are several ways to increase the surge margin. For all-radial gas turbines it has been found that opening of the turbine nozzle throat area yields high performance [5]. In addition, the throat area can be easily increased by re-staggering the nozzle guide vanes during the assembly process. In current work, the increased throat area is simulated by implementing a map-modifier to represent the increase in off-design flow capacity of the turbine component. The increase in throat area is such that the compressor surge margin is maintained at the same level as for the design point.

Figure 6.1 shows the variation in rotor speed during load shed for different fuel LHVs. The fuel valve closing time is 0.5 seconds. Figure 6.2 depicts variation in peak rotor speed achieved during the manoeuvre and how it varies with different fuels. The fuel valve closing time in this case is 1 second. It is clear from figures 6.1 and 6.2 that for a particular fuel valve closing time, the rotor over-speed in case of lower LHV fuels is higher. It is probably due to higher engine shaft power at the initial steady-state full load operating point. Fuels with lower calorific values can produce higher shaft power, if run at the same turbine entry temperature, because of increased engine mass flow. This results in an increased power unbalance in case of occurrence of load shed or load shedding. A large power unbalance, in turn, leads to higher accelerations and consequently, higher rotor speeds before the fuel flow is brought down. The figures 6.1 and 6.2 also show that for longer fuel valve closing times, the peak rotor speed is higher.

It should be noted that the rotor over-speed trip setting in case of the OP16 engine is 1.075 of the design value. However, an over-speed of less than 5% is generally sought for. It is easy to realize that running an engine at maximum allowable turbine entry temperature on a lower LHV fuel (relative to design fuel) requires fuel valves that close-off faster in order to maintain the rotor speed within acceptable limits in case of a load shed or an engine trip.

Table 6.1 Operating condition parameters provided as OP16 model input to simulate load shed for various fuels

Operating condition parameter	Value
Ambient temperature [K]	288.15
Ambient pressure [bar]	1.01325 bar
Ambient humidity [-]	60%
Initial steady-state load [-]	Maximum allowable turbine inlet temperature
Surge margin [-]	Same as the design point

Table 6.2 Composition and LHV of different fuels used for rotor over-speed analysis of OP16

Fuel composition [%volume]						Fuel LHV [MJ/kg]
CH_4	C_2H_4	H_2	CO	CO_2	N_2	
4	1	16	22	11	46	5.6
33	0	0	0	15	52	10
69	0	0	0	15	16	25
100	0	0	0	0	0	50

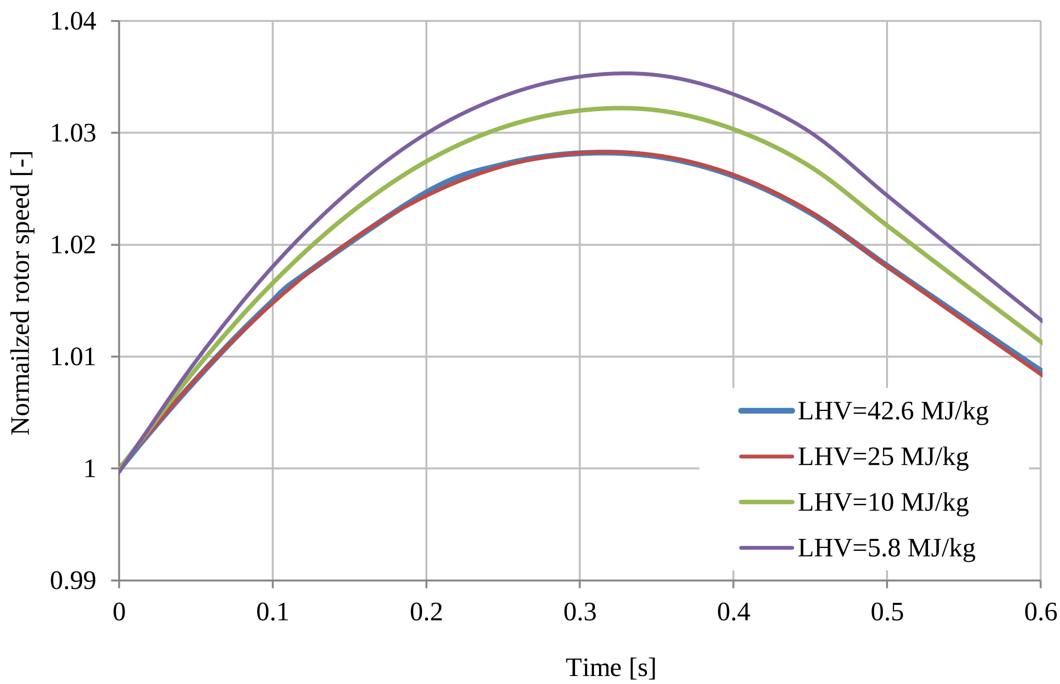

Figure 6.1 Variation in simulated rotor speed during load shed for different fuel LHVs for a fuel valve closing time of 0.5 seconds.

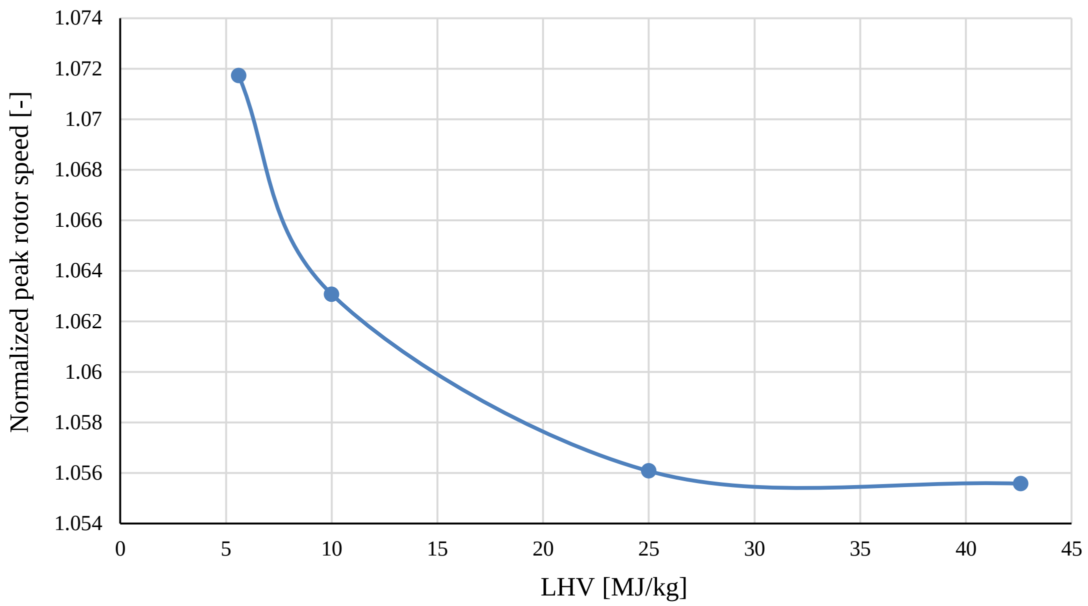

Figure 6.2 Variation in peak rotor speed with fuel LHV during a load shed for a fuel valve closing time of 1 second.

6.3 Effect of fuel valve closing time

As discussed at the start of this chapter, occurrence of load shed in power generation engines requires the fuel flow to decrease rapidly to avoid rotor over-speeds beyond a certain limit. A fast reduction in fuel flow requires fuel valves which are able to close quickly. This could be more critical when running on low LHV fuels. As observed in the previous section, low LHV fuels lead to higher rotor over-speeds when operating at the same power level. Besides, the low calorific fuels, such as syngas, have typically a high temperature due to the process in which they are being produced. Fast acting fuel valves capable of handling high temperatures are scarcely available. In order to avoid customized and expensive valves it is important to know the maximum time required to safely close the valve in case of an engine trip. This section discusses the effect of fuel valve closing time on rotor speed during a load shed to determine the safe closing time limits.

The load shed manoeuver is simulated at full load operating condition for different fuel valve closing times. The resulting rotor over-speeds are analysed for the OP16 gas turbine engine. The model inputs are as per section 6.1. The model is run on natural gas fuel (LHV = 50 MJ/kg).

As can be seen from figures 6.3 and 6.4; the maximum rotor speed during load shed increases with increasing valve closing time. Also, it takes longer time for the rotor speed to reach within allowable limits in case of slower fuel valves. It is interesting to note from figure 6.4 that the valve closing time of 1.0 second is not acceptable in order to maintain the rotor over-speed limit of 5% (of nominal rotor speed).

Figure 6.5 shows the maximum allowable fuel valve closing time in order to maintain the rotor over-speed within the acceptable limit for a range of fuel LHVs. As the fuel LHV decreases, the allowable fuel valve closing time increases exponentially. For a fuel having a heating value as low as 5.8 MJ/kg, the fuel valve closing time during a load shed should not exceed 0.7 seconds.

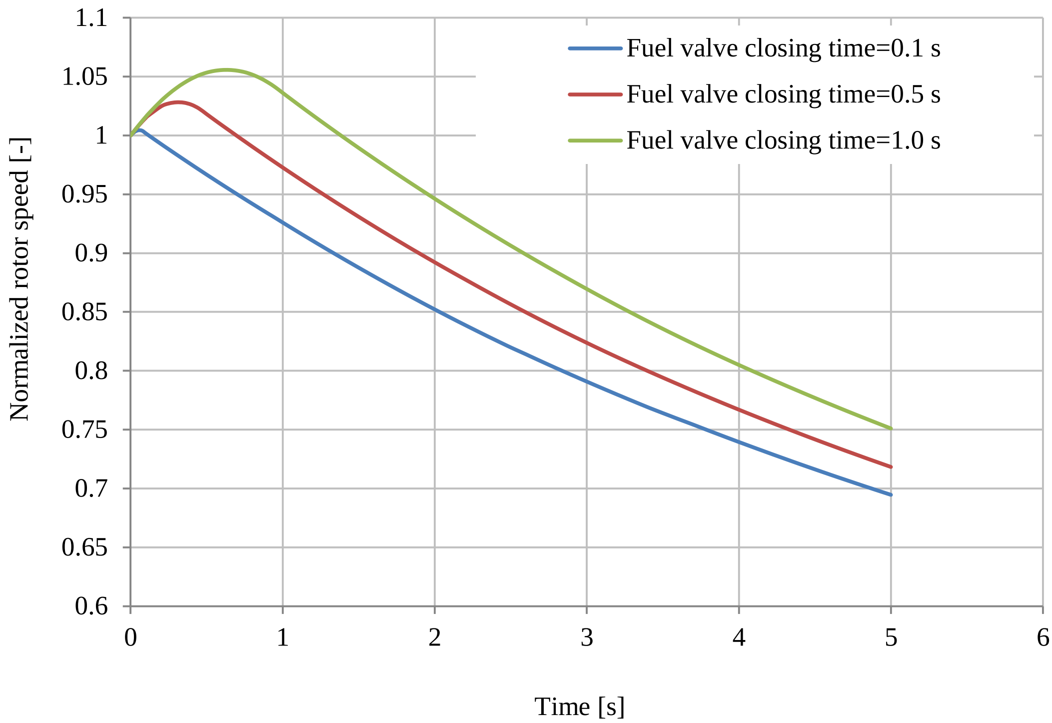

Figure 6.3 Variation in rotor over-speed during load shed for different fuel valve closing times

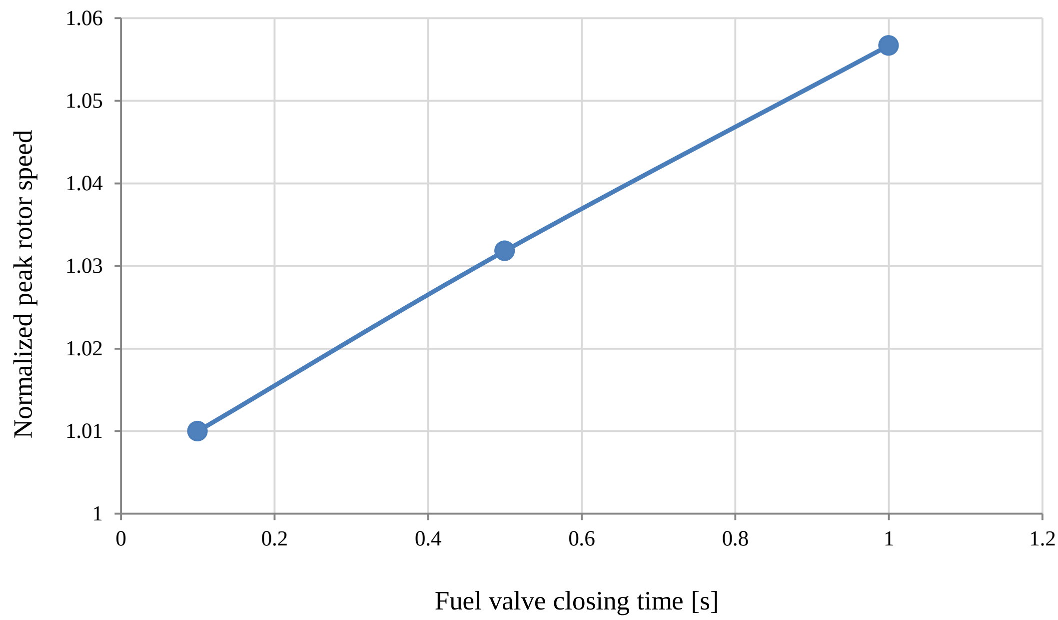

Figure 6.4 Variation in peak rotor speed during load shed with fuel valve closing time

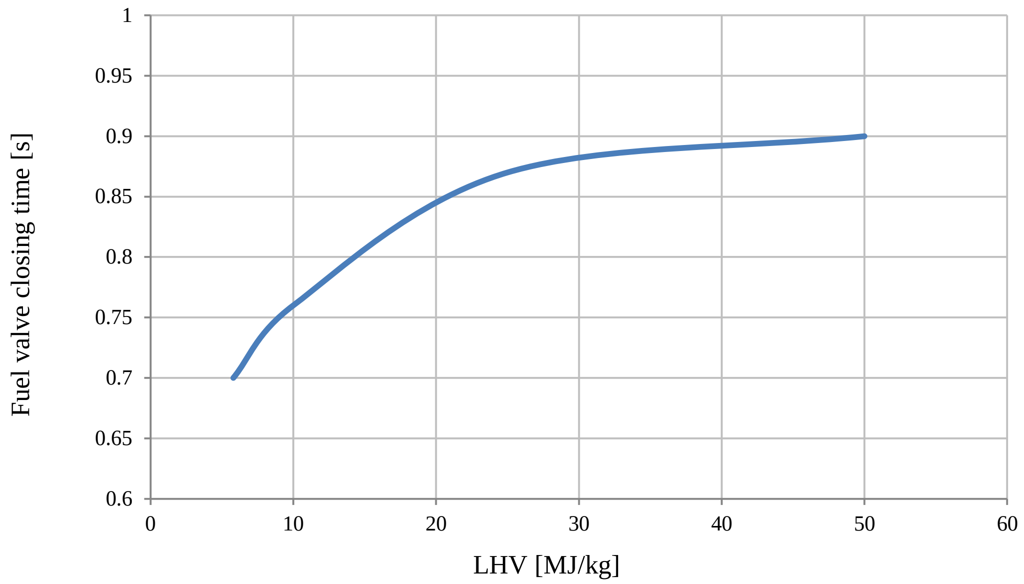

Figure 6.5 Maximum allowable fuel valve closing time to maintain maximum 5% rotor over-speed for different LHV's

6.4 Combustor volume effects

In case of engines with large volume components like some silo combustor combustors or recuperators, compressor transient operating line during a fast deceleration tends to rise initially before it moves downward. This happens because as pressure decreases during deceleration, the exit flow leaving the volume is higher than that entering it. This leads to an increased mass flow into the turbine momentarily. The decrease in turbine inlet temperature is not enough to accommodate the higher mass flow. This forces the compressor to run at higher pressure ratio which means that the operating line is shifted closer to the surge line [26].

The following analysis is aimed at investigating the influence of the OP16 low-calorific fuel combustor, which has a bigger volume, on rotor over-speed during load shed. The combustor volume is not expected to be large enough to have a substantial impact on the compressor working line but it is still interesting to see if there is a notable effect.

The OP16 model is run to simulate load shed as per model inputs discussed in section 6.1. To represent the larger low-calorific fuel combustor, the component volume of the combustor is increased to 5 times the volume of the original combustor modelled in section 5.2 to represent the larger low-calorific fuel combustor.

Figure 6.6 compares the peak rotor speed for the two combustor volumes. It is clear from the figure that the larger combustor causes the rotor over-speed to slightly increase. Obviously, volume effects depend on specific combustor volume and design. For the OP16 engines equipped with the larger volume low calorific fuel combustor, the volume effect on rotor-speed during a load shed is visible but remains minor with an increase of about 0.3% in the peak rotor speed.

Figure 6.7 shows the compressor working line during load shed for the two combustor volumes at different fuel valve closing times. It is clear that the volume dynamic effects aren't strong enough to force the working line to move higher, towards surge. It can again be seen from the figure that a slower fuel valve tends to move the transient operating points towards higher speed lines during a load shed.

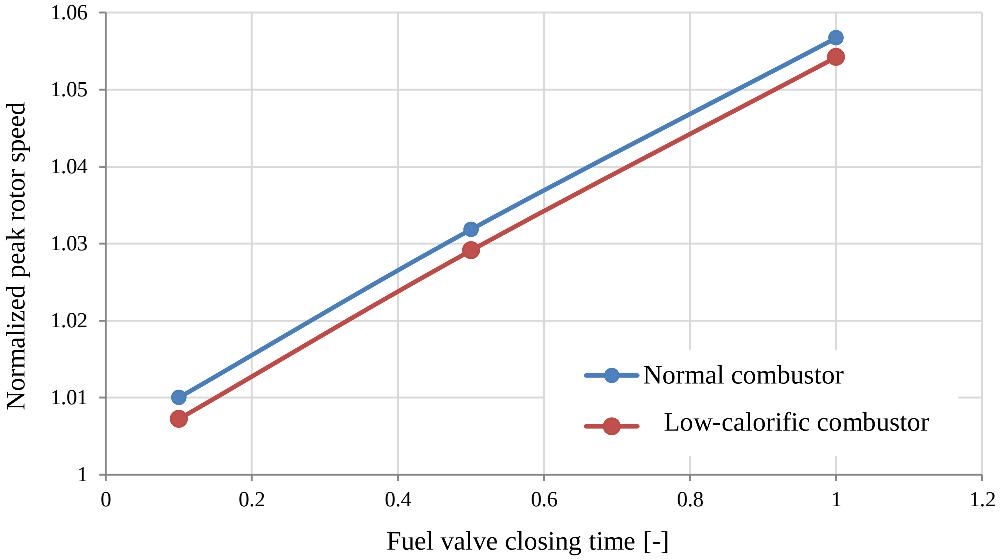

Figure 6.6 Variation in peak rotor speed with fuel valve closing time for two combustor volumes

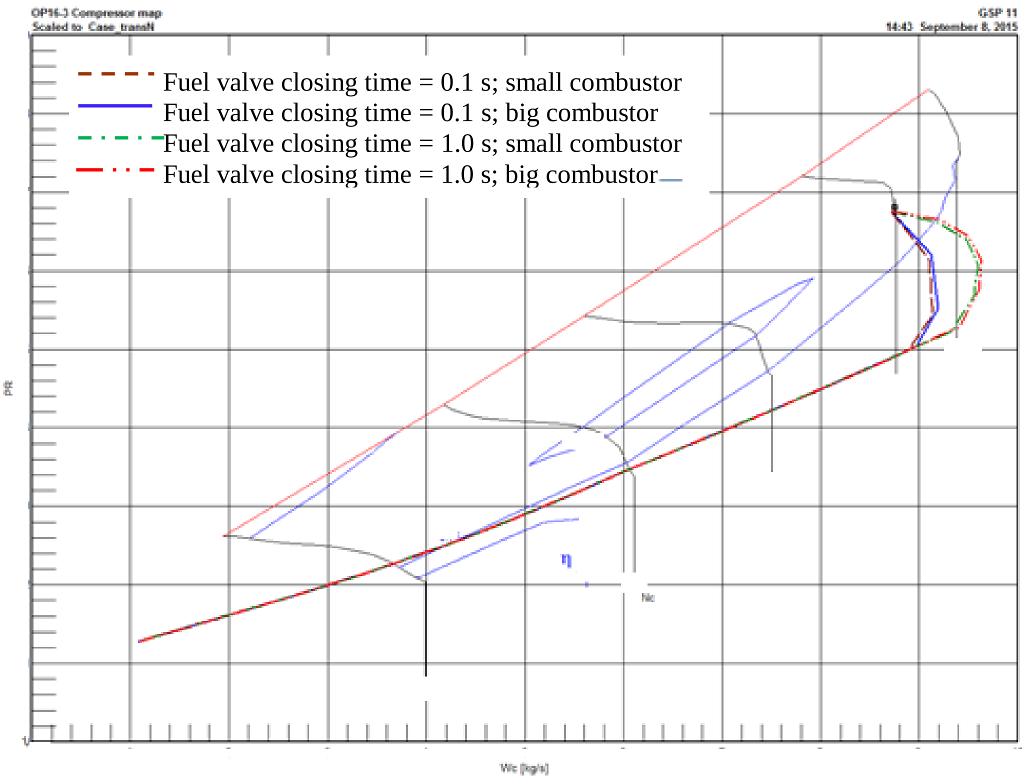

Figure 6.7 Compressor operating point response for two combustor volumes for different fuel valve closing times

7 Conclusions and recommendations

7.1 Conclusions

The conclusions derived from this work are as follows:

i. The steady-state performance model of OP16 developed using GSP accurately simulates the engine behaviour near full load operating point and can well be used for performance prediction or analysis in this region.

ii. At part-load, the simulated shaft power exceeds the measured shaft power and the difference increases with decreasing load. Investigation indicates that there may be a fault in the current test-setup, resulting in one or more flawed measurements. The prime suspects include the fuel flow and turbine exit temperature. In the current model, a correcting polynomial is implemented on simulated power to account for the apparent loss in engine shaft power. The polynomial is a function of model power level setting.

iii. Steady-state heat loss of the OP16 engine is found to have a minor effect on engine performance. At full load, the heat loss to the ambient is only about 1.5% of the design point shaft power.

iv. The dynamic model of OP16 model developed in GSP accurately simulates the engine transient performance near full load operating point. At part-load, the simulations may vary from the actual engine response, depending on the accuracy of the chosen steady-state power correction polynomial (ii). Without power correction, the simulated transient behaviour still captures the actual trends. The dynamic model of the OP16 gas turbine can be used to simulate and analyse transient manoeuvres of interest. The model can also be used to design initial control strategies for regulation of delivered power and exhaust gas temperature.

v. Rotor inertia dominates the transient behaviour of OP16 and is sufficient to accurately predict transient performance of the engine. Heat soakage effects and volume dynamics remain negligible in comparison.

vi. The fuel LHV is found to have a visible impact on transient performance of OP16. Lower LHV fuels tend to result in higher rotor over-speeds in event of engine trip or load shedding. This indicates that in order to ensure safe operation, the fuel valves need to close off quicker when operating on fuels with low heating value. For a LHV as low as 5.8 MJ/kg, the fuel valve closing time during a load shed should not exceed 0.7 seconds in order to maintain the rotor over-speed within acceptable limits, whereas for natural gas type of fuels the fuel valve closing can be up to 0.9 seconds.

vii. Combustor volume effects are observed to only slightly increase the rotor over-speed during load shedding and are not strong enough to move the compressor operating line towards surge, which is consistent with (v) above. This also remains true for larger combustors used in certain OP16 engine configurations to burn low LHV fuels. Obviously, volume effects depend on specific combustor volume and design. For OP16 engines equipped with the larger volume low calorific fuel combustor, the over-speed effect is visible but remains minor with an increase of about 0.3% in the peak rotor speed.

7.2 Recommendations

This section discusses the aspects of the OP16 model that can be improved and suggests further work that can be undertaken to generate information useful to OPRA Turbines and the gas turbine industry at large.

i. The apparent loss in measured engine shaft power at part load should be looked into in more detail in order to identify the source of error. The investigation carried out in this work suggests the possibility of fault in certain measurements, providing a good starting point for further investigation.

ii. Thermal network modelling of the OP16 combustor can be improved by extending GSP to calculate the gas temperature in the primary combustion zone, based on user-specified fuel-air equivalence ratio. This temperature can then be used to simulate radiation heat transfer from the hot gas to the combustor casing.

iii. The transient performance model of OP16 can be used to simulate the response of shaft power and exhaust gas temperature to changes in fuel flow. The resulting data can be linearized to obtain first order models which can be subsequently used for initial fuel control design purposes.

iv. To facilitate detailed design and optimization of fuel control, control system dynamics can be incorporated into the OP16 model configuration. This can be done by using a generic PID fuel control component available in GSP which allows modelling of a simple rotor speed governor. In order to model complex or engine specific control logic, custom control components can be implemented.

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