MODELLING OF MICROTURBINE SYSTEMS

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Abstract

The paper describes the development of a dynamic model of a microturbine system. The paper was done in close cooperation with the gas turbine manufacturer Turbec AB and the model was tuned and verified against their microturbine system T100. The microturbine unit consists of a compressor and a turbine connected on a single shaft to a high-speed generator. The model includes all the main thermodynamic and mechanical components of the microturbine.

Possible applications for the model are development of control strategies, dynamic performance verification, operator training and control software/hardware verification. The emphasis has been on the functionality and accuracy of the system model and not of the component models. The model is written in the Modelica modelling language and uses, when available, components from the standard libraries or from previous work as [11] and [6]. Steady-state verification has been done with good results. When the microturbine runs at full load of 100 kW, there is an average error of 0.6 % for the 13 most important thermodynamic variables compared to reference values. Dynamic verification in three different experiments has been done and the model shows a good fit to the measured data from the real microturbine.

1 Introduction and problem formulation

For control system design, process knowledge is essential. One way to obtain better process understanding is through models. The microturbine T100 from Turbec AB is constantly being upgraded. In order to predict the effects of explicit changes in the control system or in the hardware configuration of the microturbine system, a dynamic model saves time and money. The safety issue is also a strong incitement, where mistakes and deficiencies can be revealed in a simulator instead of in the real microturbine.

The objective with this project was to develop a dynamic model, which should capture the main characteristics of the microturbine system, but should also be easy to use in the every day work for the control engineers of the company. The model should be component-oriented and all developed component models should be easy to reuse in new microturbine system models.

This paper is based on [8], which is a continuation of [6]. The work of Tummescheit in [11], has been the main inspiration. For confidentiality, the axes of some figures are removed.

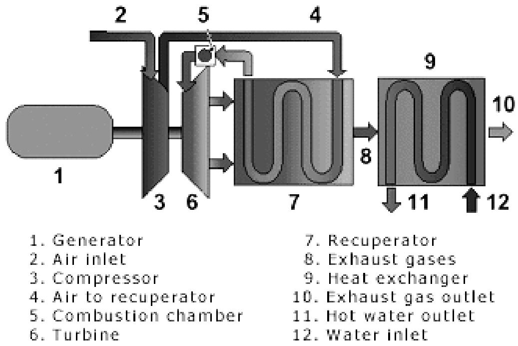

Figure 1: The thermodynamic components of the T100 microturbine

2 The T100 microturbine system

The microturbine T100 from Turbec AB is a combined heat and power (CHP) generation system, originally described in [9]. It is based on a small gas turbine engine directly connected via a rotating shaft to a high-speed generator without any intermediate gear box. There is only one moving part, the rotating shaft, which minimizes the friction.

The T100 is a microturbine for two reasons, the physical size and the power output. The compressor and the turbine are 0.15 m in diameter and the entire enclosure of the system is only 2.92 x 1.90 x 0.87 m. This makes installation very easy e.g. in can fit in a normal basement. The power output of 100 kW is chosen to fit a special market demand, which corresponds to hotels, green houses, sport facilities and wastewater treatment plants.

In Figure 1, a scheme of the thermodynamic stages of the T100 is shown. The microturbine's combustor normally runs on natural gas, but can be modified to accept various fuels such as diesel, ethanol and bio-gas. A recuperator regenerates heat from the exhaust gases to the air going into the combustor, increasing the electric efficiency to 30 %. After the recuperator, a gas/water heat exchanger uses the last heat of the exhaust gases to heat water, giving the T100 a total efficiency of 80 %. The electricity created by the high-speed generator is converted into AC voltage with a constant frequency using power electronics, which is not modelled in this paper.

2.1 The control system and its operation modes

The reference input to the controller is a power reference, i.e. the amount of electric power the generator should produce. The main control signal is the fuel rate. The measurement signals for the feedback controller are speed, power from the the generator and turbine outlet temperature (TOT). The control system has two different modes, parallel mode and stand alone mode. In the parallel mode, the T100 produces power in parallel with an existing external power grid, which serves as a power buffer. In the stand alone mode, the external grid is disconnected and the microturbine should provide all the power needed in a local power grid. The differences in the control system modes are elaborated in more detail in [8].

3 Simulation Tools

The model code is written in the Modelica language. If possible, complete components have been used from the standard Modelica libraries, especially the ThermoFluid library in [11] or from [6]. To simulate the models, the program Dymola has been used.

Dymola, Dynamic Modeling Laboratory, is a simulation program developed by Dynasim AB in Lund. It consists of a graphical user interface, compilers, numerical solvers and plot functions. For more information on Dymola, see [4].

Modelica is an object-oriented language designed to allow convenient, component-oriented modelling of complex heterogeneous physical systems. Important parts of Modelica are the object-oriented structure, the equation based modelling approach, multiple inheritance and the support for graphical representations for each component. Constructing a model with components from the standard libraries is very fast and easy, since it is just to drag and drop. For more detailed information see [10].

The ThermoFluid library has been developed at the department of Automatic Control, Lund Institute of Technology. The main purpose is to provide a general framework and basic building blocks for modelling thermo-hydraulic systems, written in the Modelica language. From this library, components as pipes, reservoirs, sensor and compressible medium models were used. More information about the library can be found in [11].

4 Theory and Modelling

The thermodynamic theory of turbomachinery and heat exchangers, based on the laws of physics, can be found in [1], [2] and [3]. The original models of the compressor, turbine and combustion chamber are taken from [6] and [7].

4.1 The compressor

From the conservation of energy, an equation can be derived that describes the specific work required to achieve a certain pressure ratio over the compressor at a given temperature. The

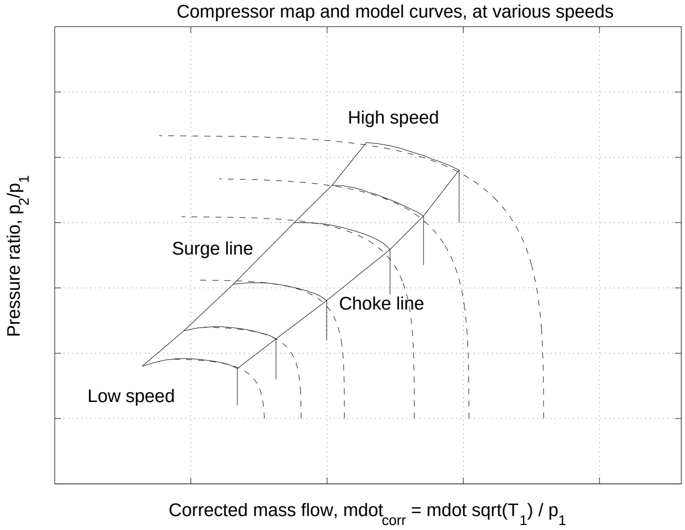

Figure 2: Compressor model (dashed) and the data (solid)

complete derivation can be found in [8]. The specific work w can be computed as:

$$w_{comp} = -\frac{1}{\eta_{is}}\left(\frac{\kappa}{\kappa-1}\right)RT_1\left(\left(\frac{p_2}{p_1}\right)^{\frac{\kappa}{\kappa-1}}-1\right) \quad (1)$$

where the isentropic efficiency η_{is} has been introduced to compensate for the non-adiabatic and non-reversible compression. The variable κ is the ratio of the specific heats, R is the universal gas constant, T is the temperature, p is the pressure and the subscript 1 and 2 refers to the inlet and outlet position. The work is negative since work is done on the gas by the compressor. The consumed power P can then be calculated as:

$$P_{comp} = \tau_{comp}\,\omega = \dot{m}\,w_{comp} \quad (2)$$

where ω is the angular velocity of the shaft, $\dot{m}$ is the mass flow rate and τ is the torque, the compressor needs.

There are some properties of a compressor that cannot be easily calculated analytically, e.g. the isentropic efficiency and the mass flow through the compressor. These must instead come from measured data, which is given in the form of a compressor map. In the map, the mass flow and efficiency are listed for different values of speed and pressure ratio. In order to reduce the number of variables needed to represent the map, the non-dimensional variables pressure ratio (pr), corrected speed (n_{corr}) and corrected mass flow ($\dot{m}_{corr}$) are used. The variables are normalized with the inlet temperature and inlet/outlet pressures during the experiments.

$$pr = \tfrac{p_2}{p_1} \quad \dot{m}_{corr} = \tfrac{\dot{m}\sqrt{T_1}}{p_1} \quad n_{corr} = \tfrac{n}{\sqrt{T_1}}$$

For more background on non-dimensional analysis, see [2] and [5].

Each of the solid curves in Figure 2 corresponds to one constant speed of rotation. The range is from 20 000 rpm up to

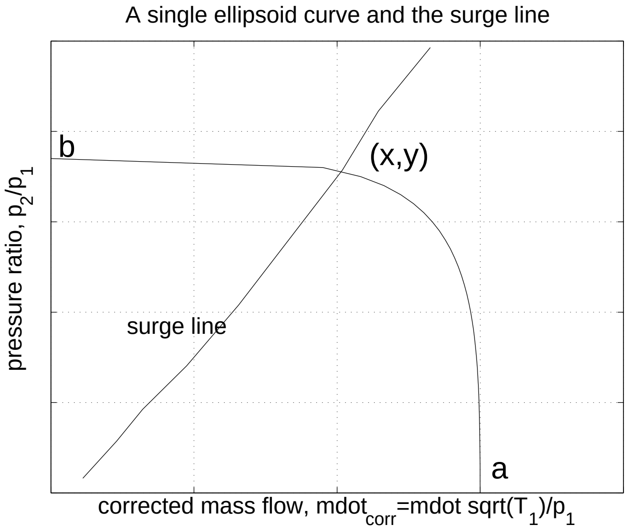

Figure 3: Ellipsoid curve for one particular speed of rotation

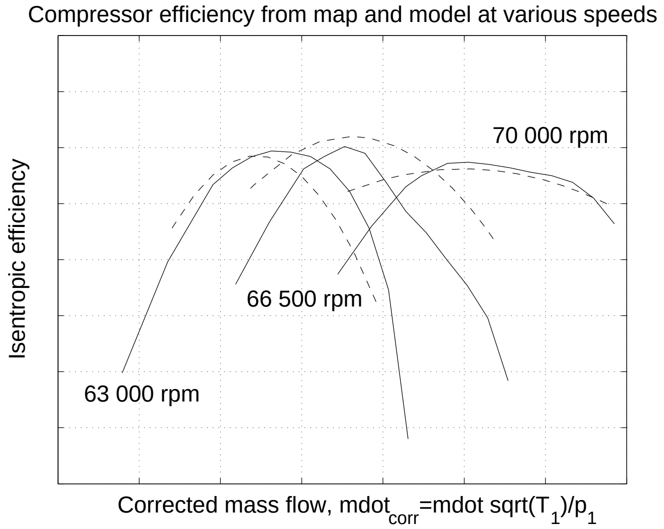

Figure 4: Compressor efficiency, map data (solid) and model (dashed)

74 000 rpm. The area enclosed between the surge line and the choke line is the normal operating range for the compressor. For a given pressure ratio and inlet temperature, the compressor model should give a unique mass flow using data from the map.

The method used in this paper was to fit continuous ellipsoid curves to the different speed curves and then parameterize them so that the whole map can be continuously represented, see [7]. The curves in Figure 2 can be approximated as ellipsoid curves and can be represented with an ellipsoid equation:

$$\left(\frac{x}{a}\right)^z + \left(\frac{y}{b}\right)^z = c \tag{3}$$

When the parameters a, b, c and z are varied, the form of the ellipsoid curve can be adjusted to fit any speed curve from Figure 2. The parameter a corresponds to the corrected mass flow at pressure ratio one (where the curve would cross the x-axis). Similarly the parameter b represents the pressure ratio at zero mass flow (where the curve would cross the y-axis). The parameter c is a constant usually taken to one. The parameter z represents the curvature of the curve. For each speed curve, one set of these parameters are calculated, see Figure 3. To make the model continuous in respect to speed based on these discrete sets of parameters, the parameters are fitted as polynomial functions of speed.

The compressor model is calculated in the following way: For each speed, the value of the parameter a is taken directly from the map, i.e. the mass flow value at choking conditions. The parameter z is at first set to an arbitrarily value, e.g. 5. One data point (x, y) is taken from the map, at which the ellipsoid curve and data will be identical matched. Using this point, the ellipsoid equation is solved for b. The curve is plotted and compared to data from the map. By visual inspection, the parameters a and z are modified to ensure a better fit for this specific speed inside the operating range. The procedure is repeated for each speed of rotation, which results in a set of parameters for each speed. The values of the each parameter are fitted in Matlab into polynomial functions as a function of speed to achieve a continuous model. The mass flow is now continuously given by speed and pressure ratio.

The compressor efficiency was modelled as a parabolic degradation function depending on mass flow rate. Both the compressor map model and the efficiency model originate from and are in detail described in [7].

For a given rotational speed, the maximum efficiency is well-known, i.e. the top value of the curves seen in Figure 4. Then the efficiency map can be modelled as parabolic degradation curves. The amount of curvature/degradation is denoted d and is fitted based on numerous data to get parameterization for all speeds. The following equation is taken from [7] and valid for one speed.

$$\eta_{comp} = \eta_{comp,max} - d(\dot{m} - \dot{m}_{maxeff})^2 \tag{4}$$

where $\eta_{comp,max}$ is the maximum efficiency at that speed and $\dot{m}_{maxeff}$ is the corresponding mass flow for this maximum efficiency. To make to model continuous, these variables are fitted to polynomial functions of speed.

As can be seen in Figure 4, the solid curves are not exactly symmetrical. Near choking conditions (to the right), the efficiency is decreased rapidly. Another difficult part in this case is that the curvature changes a factor of 20 from the lowest to the highest speed. Due to the control system, the microturbine often operates near optimum conditions and there the efficiency model is accurate.

4.2 The turbine

The equation for the specific work done by the gas on the turbine is almost the same as for the compressor, see Equation 1, except for a sign change and that the isentropic efficiency is

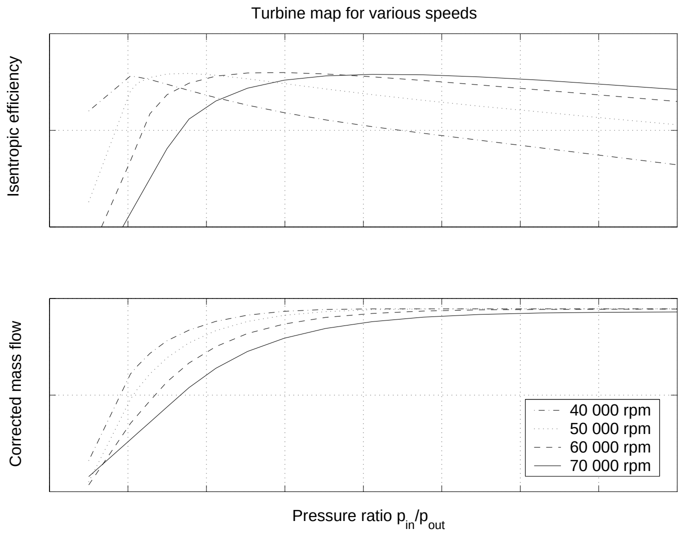

Figure 5: Turbine map for different speeds

included in the numerator:

$$w_{turb} = \eta_{is}\left(\frac{\kappa}{\kappa-1}\right)RT_1\left(1-\left(\frac{p_2}{p_1}\right)^{\frac{\kappa}{\kappa-1}}\right) \tag{5}$$

The turbine is mounted on the same shaft as the compressor, producing enough torque to power the compressor and the generator. The relation between the turbine, the compressor and the generator will then be:

$$P_{turb} = \tau_{turb}\,\omega = \dot{m}\,w_{turb} = \dot{m}\,w_{comp} + \tau_{gen}\,\omega = P_{comp} + P_{gen} \tag{6}$$

In the turbine map, Figure 5, the mass flow and efficiency are shown for different values of speed and pressure ratio. The corrected mass flow is approximated as near or at its maximum, i.e. a choked flow. The maximum value of the corrected mass flow can then be calculated by the choked flow equation from [1].

$$\dot{m}_{corr} = \frac{\dot{m}\sqrt{T_1}}{p_1} = \frac{A_{thr}M\sqrt{\kappa}}{\sqrt{R\left(1+\left(\frac{\kappa-1}{2}\right)M^2\right)^{\frac{\kappa+1}{\kappa-1}}}} \tag{7}$$

The variable A_{thr} is the smallest nozzle throat area (the cross-section area) at the inlet to the turbine. The variable M is the Mach number, defined as the ratio of flow speed inside the turbine and the speed of sound.

The efficiency data from the turbine map in Figure 5, is used in a bilinear (2-dimensional) interpolation method to model the efficiency of the turbine, see [8].

5 Other components of the microturbine model

The recuperator and the gas/water heat exchanger are both of mainly counter flow type and are therefore modelled in the same way, except for the medium and the geometry. From [3], standard 1-dimensional convection and conduction heat transfer laws are used.

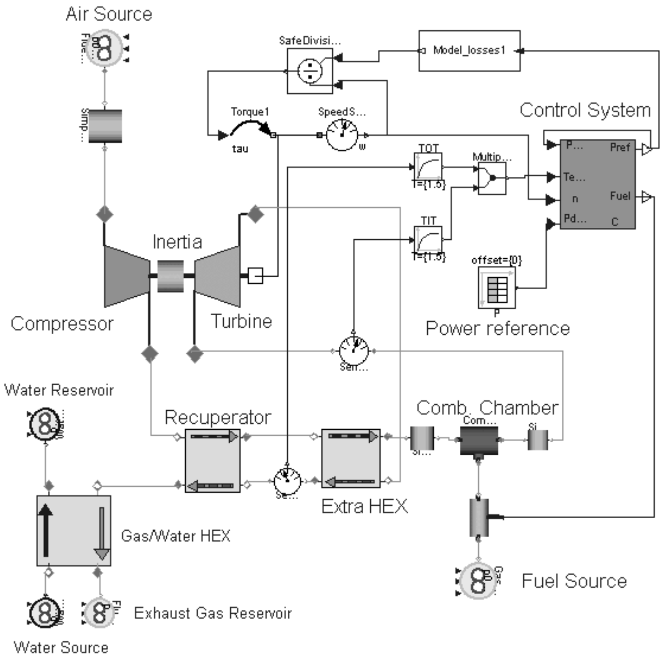

Figure 6: The complete model with graphical representation of each component

The combustion chamber is modelled with chemical and energy conservation equations. For a given flow rate of the fuel and the compressed air, the theoretically released energy can be computed and the temperature of the exhaust gases is increased accordingly. For the complete equations and the chemical reactions, see [6].

The electric power to be generated is set by the user of the model. Knowing the rotational speed, the power is converted to a torque, which is subtracted from the common shaft of the compressor and the turbine.

6 Simulations and Verification

With the overall model, it is open for the user to simulate the microturbine under any circumstances the user chooses. For each simulation case, there are numerous parameters that are to be set, in total there are about 6000 equations and 130 states and the simulation time is about a minute. These numbers can vary depending on the number of discretizations used in the heat exchangers and the accuracy of the medium models. The model can be linearized in Dymola for theoretical analysis and with appropriate model reduction, used in controller design. It is also possible to include the Modelica model in Simulink as a separate block.

In Figure 6, the system model is presented. Each component model has a graphical representation, to allow drag and drop actions to quickly build new system models. Each component (e.g. a heat exchanger) can itself be constructed, either graphically or by code, using other sub components (e.g. pipes and wall models).

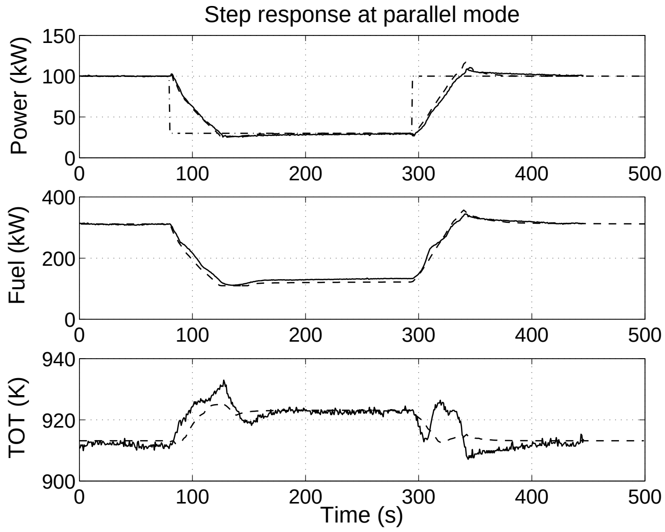

Figure 7: Step response in the electric power reference in parallel mode, measured data (solid) and model data (dashed)

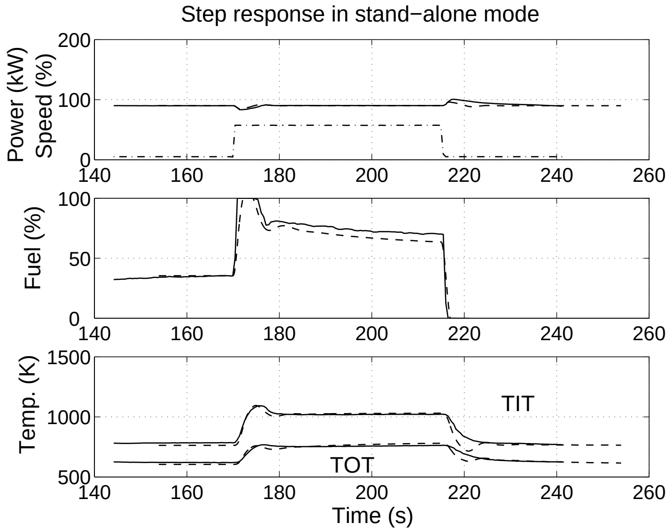

Figure 8: Step response in the stand-alone mode, measured data (solid) and model data (dashed)

6.1 Steady-state verification

The steady-state results of the model were verified with data provided by Turbec AB. The simulations were done with the control system in parallel mode. The model was verified at three different loads, 100 kW, 70 kW and 50 kW of electric power. The 13 most important thermodynamic variables were used in the verification process and a percentage error of the model output compared to the provided data was calculated. The thermodynamic variables used in the verification were e.g. speed, fuel rate, temperatures, efficiencies, pressure drops and pressure ratios for the compressor. The average error was 0.61%, 1.2% and 1.1% for the 100 kW, 70 kW and 50 kW case respectively. The complete result can be found in [8].

The largest errors were in the compressor's pressure ratio (3.7%) and the pressure drop between the compressor and the turbine (3.5%). The first error is caused by errors in the ellipsoid curves that model the compressor mass flow, which is tightly coupled to the pressure ratio. The second error was due to too simple pressure drop models.

6.2 Dynamic verification

Three dynamic experiments were investigated and the data from the real microturbine was plotted and compared with the model output from simulations of the same experiments.

The first case is a simple step response experiment shown in Figure 7. The microturbine runs at full load, 100 kW of electric power and is in steady state. The control system is in parallel mode. The power reference (dashed-dot) is, at a given time t, set to 30 kW. After approximately 200 seconds, the reference is set again to 100 kW. In Figure 7, the trajectories of the power output, consumed fuel and TOT (turbine outlet temperature) are shown from the measured data (solid) and model output (dashed).

The model output show large similarity to the measured values. At t = 320 s, there is a difference in the TOT signal, probably due to unmodelled dynamics and differences in the model of the control system.

The second case is also a step response experiment, but now the control system is in stand-alone mode using a speed controller. The power reference is set to 5 kW load, i.e. some kind of idle load. The speed is kept constant at 63 000 rpm. At a given time t the power reference (dashed-dot) is set to 59.2 kW and is then after another 45 seconds set back to 5 kW. In Figure 8 the trajectories of the power output, speed, consumed fuel, TOT and TIT (turbine inlet temperature) are shown, from the measured data (solid) and model output (dashed).

When the load is suddenly increased, the speed drops quickly, but the controller reacts and increases the fuel flow to the maximum value, 100 %. When the load disappears, the control system cuts down the fuel rate to a minimum and the fuel rate should then rise to the original level. Due to problems in the log file, neither measured data or model output is plotted after t=220 s. The difference between the model and the measured data in TOT and TIT, right after the step at 220 seconds, is probably caused by changes in the time constants of the temperature sensors due to varying flow rate.

The third case is a brake test verification. The microturbine is run at part load. At a given time t, the electric power is disconnected to simulate a power circuit failure or a sudden decrease in load. Then the fuel valves are closed immediately down to a minimum. To prevent the microturbine from over speed, i.e. speeds over 70 000 rpm, a brake chopper is switched on. The brake chopper consists of resistors, which brake and dissipate the kinetic energy of the machine via the generator.

The data from the model in Figure 9 demonstrates great similarity with the measured data down to speeds around 35 000 rpm, the lowest part of the valid range of the model. The er-

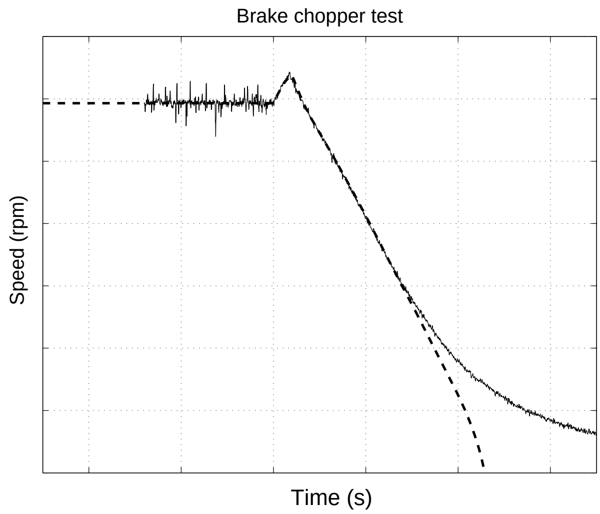

Figure 9: Brakechopper test

ror comes from a difference in the simple brake chopper model (the power it dissipates) and the control system model. The difference increases for lower speeds.

7 Applications

The model has at first been used at Turbec AB, to study braking strategies and the choice of brake choppers. The objective is to prevent over speed and depending on size, duration of the brake action and also the ambient temperature, the response from the microturbine will be different.

The model can also be used to simulate the microturbine behaviour at internal and external variable changes, e.g. variable load, ambient temperature and control system settings. Other fields of use can be dynamic performance verification, operator training and control software/hardware verification.

8 Conclusions

A dynamic model of a microturbine has been developed and thoroughly verified against a real microturbine from the industry. The steady-state results are very good with an average error of 0.6% for the 13 most important variables compared to reference values. The dynamic simulation results have demonstrated that the model captures the characteristics of a real microturbine very well in three different situations.

The model was mainly constructed by complete components from [11] and [6] and when not available, new component models were developed. A large effort has been made to accurately model the compressor and turbine map. Even though this model describes a microturbine, the model can be adjusted to a gas turbine of any size, by changing the geometrical properties and the special characteristics of the compressor and the turbine. This is achieved by the multiple inheritance, the object-oriented structure of the Modelica language and its support for graphical representation of each component, see Figure 6.

9 Future work

To model the compressor and the turbine accurately, a large manual effort was made in order to tune the model parameters as much as possible. It is desirable that the models are modified, such that more of the work is done automatically.

To make the system model more general, a model of the power electronics can be developed. It might also be interesting to have a model of the auxiliary system, to make the system model more complete.

10 Acknowledgments

The project has been made largely at Turbec AB under the supervision of Anders Åberg, which has been a great help and mentor. At the department, Hubertus Tummescheit has been the expert on modelling in general and on ThermoFluid in particular.

References

[1] Y. Cengel and M. Boles. *Thermodynamics, An Engineering Approach.* WCB/McGraw-Hill, New York, 3 edition, 1998.

[2] G. Cohen, H. Rogers and H. Saravanamuttoo. *Gas Turbine Theory.* Longman Group Ltd, London, 4 edition, 1996.

[3] D. DeWitt and F. Incropera. *Fundamentals of Heat and Mass Transfer.* Wiley, New York, 4 edition, 1996.

[4] Dynasim AB, Lund, www.dynasim.se. *Dymola, Dynamic Modeling Laboratory User's Manual*, 2001.

[5] R. Fox and A. McDonald. *Introduction to Fluid Mechanics.* Wiley, New York, 5 edition, 1998.

[6] A. A. Gómez Pérez. "Modelling of a gas turbine with modelica." Technical Report Masters thesis ISRN LUTFD2/TFRT--5668--SE, Department of Automatic Control, Lund Institute of Technology, Sweden, May 2001.

[7] J. Gustafsson. "Static and dynamic modelling of gas turbine in advanced cycles." Technical Report Licentiate Thesis ISRN LUTMDN/TMVK--98/7030--SE, Lund Institute of Technology, Lund, Sweden, 1998.

[8] S. Haugwitz. "Modelling of microturbine systems." Technical Report Masters thesis ISRN LUTFD2/TFRT--5687--SE, Department of Automatic Control, Lund Institute of Technology, Lund, Sweden, May 2002.

[9] A. Malmquist. *Analysis of a Gas Turbine Driven Hybrid Drive System for Heavy Vehicles.* PhD thesis ISSN --1102-- 0172, The Royal Institute of Technology, Sweden, June 1999.

[10] Modelica Association, www.modelica.org. *Modelica - A Unified Object-Oriented Language*, 2000.

[11] H. Tummescheit. *Design and Implementation of Object-Oriented Model Libraries using Modelica.* PhD thesis ISRN LUTFD2/TFRT--1063--SE, Department of Automatic Control, Lund Institute of Technology, Sweden, August 2002.