Institutionen för systemteknik
Department of Electrical Engineering

Examensarbete

Effects of non ideal inlet and outlet pipes on measured compressor efficiency

Master's Thesis Performed in Vehicular Systems,
The Institute of Technology at Linköping University
by

Kristoffer Ekberg

LiTH-ISY-EX--15/4860--SE

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Linköpings universitet
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Sammanfattning
Abstract

The thesis is about investigating the inlet and outlet pipes effect on the compressors measured performance. From measurements made in a gas stand, a thermodynamic model is to be created and the compressor efficiency further investigated. The temperatures and pressures entering and leaving the compressor does not have to be the same as the temperatures measured in the gasstand, because of the thermodynamics of the pipes that connects the measurement equipment and the compressor. During a gasstand test the turbocharger is connected in a test bench, it is connected with pipes on both the compressor and turbine side, to simulate the hot exhaust gases from the car engine and the pressure increase over the compressor. The air entering and leaving the turbocharger through the different pipes is controlled and all the entering and leaving temperatures and pressures are measured. Gasstand data from different tests are available during the thesis, one specific turbocharger is used as references during the modeling. Models of the inlet and outlet pipes are created and connected to a compressor model. The model is controlled to give the same mass flow as the measured data, to ensure that the work cycle is followed. The effects of the non ideal inlet and outlet pipes on measured compressor efficiency is studied with help of this model and the main impacts on the measured compressor efficiency are discovered. The result shows that the measured values used to calculate the compressor efficiency could change, depending on the measurement positions on the inlet and outlet pipes.

Nyckelord
Keywords: Turbocharger, Compressor, Thermodynamic, Pipe, Efficiency.

Abstract

The thesis is about investigating the inlet and outlet pipes effect on the compressors measured performance. From measurements made in a gas stand, a thermodynamic model is to be created and the compressor efficiency further investigated. The temperatures and pressures entering and leaving the compressor does not have to be the same as the temperatures measured in the gasstand, because of the thermodynamics of the pipes that connects the measurement equipment and the compressor. During a gasstand test the turbocharger is connected in a test bench, it is connected with pipes on both the compressor and turbine side, to simulate the hot exhaust gases from the car engine and the pressure increase over the compressor. The air entering and leaving the turbocharger through the different pipes is controlled and all the entering and leaving temperatures and pressures are measured. Gasstand data from different tests are available during the thesis, one specific turbocharger is used as references during the modeling. Models of the inlet and outlet pipes are created and connected to a compressor model. The model is controlled to give the same mass flow as the measured data, to ensure that the work cycle is followed. The effects of the non ideal inlet and outlet pipes on measured compressor efficiency is studied with help of this model and the main impacts on the measured compressor efficiency are discovered. The result shows that the measured values used to calculate the compressor efficiency could change, depending on the measurement positions on the inlet and outlet pipes.

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Linköping, Maj 2015
Kristoffer Ekberg

Contents

Notation

Symbols

Symbol	Description
$\dot{m}$	Mass flow
$\dot{Q}$	Heat flux
c_p	Energy storage capacity
h	Heat transfer coefficient
m	Mass
L	Pipe length
D_c	Compressor scroll diameter
N_{tc}	Turbo rotational speed
A_s	Surface area
A_c	Cross section area

Sub-indexes

Sub-index	Description
c	Compressor
t	Turbine
01	Measured temperature inlet
$01'$	Temperature inlet after pipe
02	Measured temperature outlet
$02'$	Temperature outlet before pipe
03	Inlet turbine
04	Outlet turbine

1 Introduction

The thesis main area is to model the thermodynamics of a turbocharger in MATLAB & SIMULINK. A couple of data sets containing measurements from three different turbochargers tested in a gasstand are made available during the thesis. One of these turbocharger data sets is used to create a thermodynamic model of a turbocharger. When the thermodynamic model is created and tuned to one of the turbochargers, different experiments will be performed and analyzed to find the main impacts on the compressor efficiency.

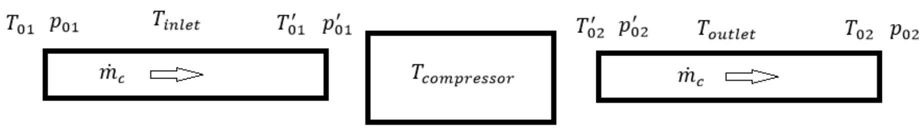

Figure 1.1: *Shows the notations of important states and measured values in the model. The pipe temperatures are T_{inlet} and T_{outlet}, the compressor temperature is $T_{compressor}$. The temperature and the pressure of the air entering and leaving the inlet pipe are T_{01}, T'_{01}, p_{01} and p'_{01}. Same notations are used for the outlet pipe, but the sub index is 02. $\dot{m}_c$ is the mass flow of air through the pipes.*

1.1 Background

Today the commercial turbocharger is used in a wide range of vehicles, from small passenger cars to big trucks and lorrys. To get a better understanding of the compressor efficiency, deeper investigations has to be done considering the air entering and leaving the compressor. The inlet and outlet pipes may affect the air flowing through the compressor. The actual temperatures and pressures may differ from the actual measurements, in for example a gasstand, and has to be accounted for when investigating the compressor efficiency. This thesis is about creating a model for the thermodynamic phenomena that affect the air when it flows through the compressor side of a turbocharger. The goal is to find out how the physical setup of the turbocharger affects the compressor map. In for example a gasstand, measurement instruments are not placed directly before and after the inlet and outlet. If the air is affected by the inlet and outlet pipe that it has to pass through, it could change the appearance of the compressor map. Also the manufacturing of the pipes could come into play, depending on the surface roughness in the pipes, the compressor map could be affected at higher shaft rotational speed.

1.2 Problem

The main task is to create a model of a turbocharger that takes the thermodynamic phenomena in the pipes into account. The model should be parametrized in a way so it can be changed and used for other turbochargers, this is achieved by letting the user adjust parameters such as the inlet and outlet pipes diameters and lengths. To be able to model the turbocharger, a large dataset of gasstand measurements, with series of experiments performed will be used as reference. The reference turbocharger is a TD04HL-15T from Mitsubishi, often used in passenger cars, such as the SAAB Aero. The gasstand measurements contains information about surface temperatures on the inlet and outlet pipes connected to both turbine and compressor, surface temperatures on compressor (at both inlet and outlet), surface temperatures on turbine, temperature measurements on cooling oil and water. It also contains inlet and outlet temperatures, pressures and mass flows on fluids entering and leaving both the compressor and turbine. With help of these measurements, a model of the compressor is to be created. The model will be built in a modular way to make it easy for users to customize it and make changes, such as adapting the model for another turbocharger. The model shall be created in MATLAB & SIMULINK. The thermodynamic model will be used to find the main impacts on the compressor efficiency, depending on the measurement setup. The area of interest in the compressor map is the lower RPM section, where the mass flows of air are relatively small, and the heat exhanges may be significant. When the thermodynamic model is created, different gasstand test setups will be examined in terms of compressor efficiency. The goal is to use the developed model to recalculate the compressor maps given by suppliers, to take the inlet and outlet pipes into account when calculating the compressor efficiency.

1.2.1 Schematic of the turbocharger

The drawing in figure 1.2 shows the placement of the different sensors on the turbocharger surface. There are pressure sensors, mass flow sensors and temperature sensors. The sensors denoted with sub-index 1-12 are thermocouples placed on the outer surface of the housing of the turbo. The sensor placements and index names are shown in figure (1.2). The mass flow, pressure and temperature sensors measuring the conditions of inlet and outlet air are placed at a certain distance from the actual compressor housing.

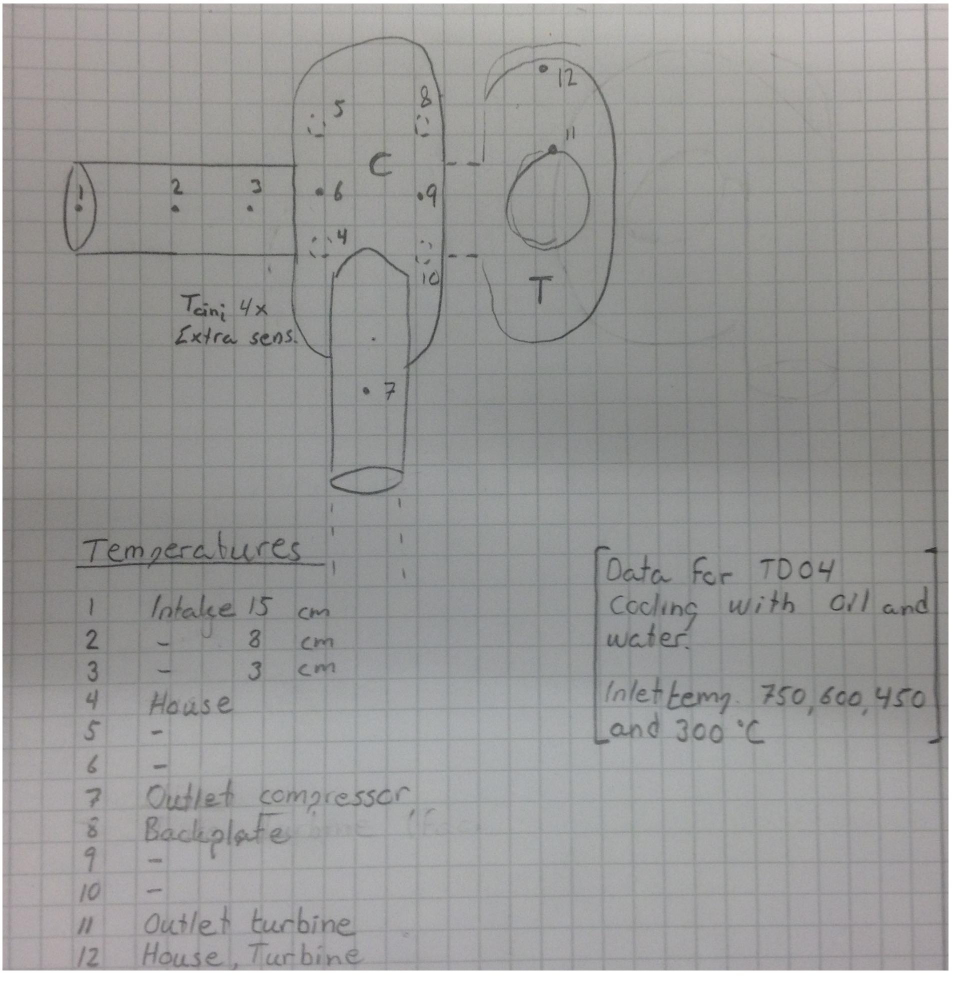

Figure 1.2: *Shows the location of the surface temperature sensors.*

1.3 Literature survey

A literature survey has been performed, mainly in the SAE database and two different books. The goal of the study was to achieve knowledge of how to model the turbocharger and which components that are the most important in terms of heat transfer and heat losses. No articles related to compressor efficiency and the inlet and outlet pipes are found. The following articles and books are relevant for the subject, a short description of contents is presented.

1.3.1 Articles and papers

According to Aghaali and Ångström [2013] the heat transfer conditions of the turbocharger may be defined by the turbine inlet temperature, the ambient temperature, oil heat flux, and the air around the turbocharger. Some of their experiment results suggest that the temperatures of the turbocharger walls are predictable.

The main reason for my thesis is to find out how the inlet and outlet temperatures transfer energy to or from the turbocharger components. In Bannister [2014] the exhaust gas is the driving force, both the gas entering and leaving the turbine. They also say that the air entering and leaving the compressor is affected by the inlet and outlet pipe on the compressor side.

Eriksson [2002] have modeled the heat flux of an exhaust pipe in three different ways, two static models and one dynamic. The first model has no pipe wall conduction along the flow, also all the heat transfer coefficients where lumped together. The second model assumes equitemperature along the exhaust pipe wall. By this assumption the heat transfer from gas to wall is by convection only. The third (the dynamic model) describes the systems time dependent behavior. The conclusion is that even if these two quite different assumptions have been made about the wall temperature, the result isn't that different.

When looking at earlier turbocharger heat transfer modeling studies, one can see that some parts are more important than others. Paper Westin et al. [2004] claims that the heat losses from the turbine is important to model to get reliable results out of the turbocharger efficiency maps. If the turbine volute heat losses are taken into account, the turbine wheel inlet temperature will be different, and thereby the turbine map will change.

To model the turbocharger in MATLAB, a lot of work needs to be done by hand before the equations and parameters can be inserted into the program. Paper J. R. Wagner and Paradis [2001] have modelled the thermodynamics of an engine by creating a resistance network. If the material data and all the heat transfer parameters where known, a resistance network could be used to calculate the temperatures of different parts on the turbocharger. Their mathematical models would serve as model based ECU control algorithms for a commercial vehicle.

Cormerais et al. [2006] concludes that the heat transfer from the turbine has a major influence on the compressor performance. They claim that the compression process is non-adiabatic, and can't be assumed adiabatic anymore.

1.3.2 Books

Two books will mainly be used to make a thermodynamic model of the turbocharger. The first is Cengel et al. [2008], this book is helpful when calculating the energy of the fluids and estimating the heat transfer. The second is Eriksson and Nielsen [2014], this book is helpful when stating the turbine and compressor equations and also when plotting the compressor map.

1.4 Approach

The turbocharger is divided into thermodynamic subsystems. These systems have the same boundary temperatures, mass flows and pressures if connected to each other. The different subsystems are modeled one by one to verify the different parts. When all the subsystems are verified, they are connected, to form a functional model that describes the compressor. The inlet and outlet pipes are two subsystems, the compressor is one. When the model is created, experiments with different pipe lengths and different ambient temperatures will be carried out.

2

Heat transfer modeling

In this chapter the development of the thermodynamic model describing the compressor is presented. The compressor is divided into different subsystems. The different subsystems are the compressor housing and the inlet and outlet pipes. The compressor housing temperature is assumed to be known. The temperatures of the pipes are left as states to be solved by MATLAB & SIMULINK. A steady state data set is used during simulations, the dynamics are increased with a high gain in the SIMULINK model to reduce the settling time so that the model quickly moves to steady state. Thereby the dynamics will be very fast and the output considered to be steady state. The reason to use dynamic models is because the equations describing the pipe temperatures (see section 2.1.1) are hard to solve in a numerical way. The dynamic models are used to solve the steady state values, and thereby solve the equations. The heat transfers that acts on the systems are shown in figure 2.1. There is conduction between the pipes and the compressor housing, radiation and convection from the pipes to the surrounding air, and forced convection to the air flowing through the pipes. Some of the heat transfer coefficients are estimated and some calculated. The heat transfer coefficients from the compressor housing to the pipes are used as calibration parameters and the convectional heat transfer coefficients are calculated according to empirical formulas.

2.1 Modeling inlet and outlet pipes

During the start of the thesis, it was discovered that the inlet and outlet pipes changed temperature slightly depending on the mass flow and compression ratio, it is assumed that the pipes affect the inlet and outlet air when entering and leaving the compressor. A dynamic model is used to solve the energy balance to find the pipe wall temperatures, the pipes are connected to the compressor hous-

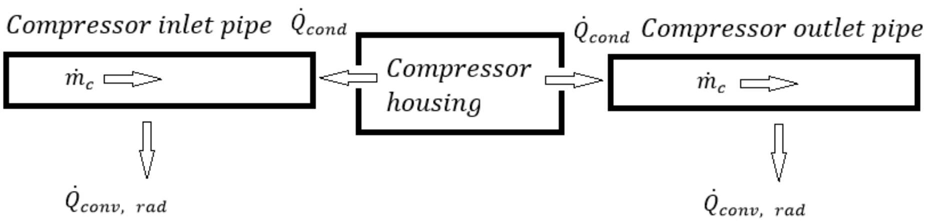

Figure 2.1: *Shows the heat transfers acting on the compressor and the pipes. The temperatures on the different subsystems will decide if the conductive heat flux goes to or from the compressor. The compressor housing and the inlet and outlet pipes, each have a temperature. Through the pipes there is a mass flow $\dot{m}$, that convects energy to or from the pipes depending on the temperature differences. The pipes convects and radiates heat to the environment.*

ing, which is connected to the bearing housing. The dynamic model stated to describe the change of the air temperature flowing through the pipe is found in Eriksson [2002]. The models that this paper describes are validated on exhaust pipes, and are used on the inlet and outlet pipes for the compressor. The wall temperature is left as a state, to make MATLAB & SIMULINK to solve the pipe temperature. Surface temperatures are available from the inlet pipe and the outlet pipe on the compressor. During the gasstand measurements, the pipes where wrapped in low conductive cord to minimize the heat exchange with the environment. When the heat transfer coefficients describing the conduction from the compressor to the pipes are calibrated, the heat transfer coefficient is adjusted until a good result in terms of reference following is achieved. The radiation and natural convection are assumed to be zero. The effect of the natural convection and radiation are later added to get a better view of how the surroundings could affect the measurements.

2.1.1 Wall temperature model

The wall temperatures of the pipes are used to calculate the heat transfered to or from the air in the pipe, and also to estimate heat flux from the pipe to the environment. The pipe surface temperature is close to ambient temperature on the inlet pipe, connected to the compressor. The result of this is that the energy transfer between the surrounding air and the inlet pipe is small. The inlet air temperature is also close to ambient temperature. If there is any heat transfered to raise the temperature on the inlet pipe, that energy should mainly come from the compressor housing. The outlet pipe has a slightly higher temperature than the inlet pipe, because the compressed air leaving the compressor is heating the pipe through convection and the compressor housing is conducting heat to the pipe. To be able to describe the pipe wall temperature, a model that takes both conduction, convection and radiation into account is used. The model used to describe the pipe wall temperature is from Eriksson [2002], and is given by:

$$\dot{Q}_e = A\left(h_{cv,e}(T_w - T_a) + F_v \epsilon\sigma(T_w^4 - T_a^4) + h_{cd,e}(T_w - T_c)\right) \tag{2.1a}$$

Equation (2.1a) is changed to equation (2.1b) to get the connection area involved separately.

$$\dot{Q}'_e = A_s\left(h_{cv,e}(T_w - T_a) + \epsilon\sigma(T_w^4 - T_a^4)\right) + A_c h_{cd,e}(T_w - T_c) \tag{2.1b}$$

Where $\dot{Q}'_e$ is the external heat flux, A_s is the surface area, A_c is the pipe cross section area, T_w is the pipe wall temperature, T_a is the ambient temperature, T_c is the temperature on the compressor housing, ε is the material emissivity, σ is Stefan Boltzmans constant, $h_{cv,e}$ and $h_{cd,e}$ are the heat transfer coefficients for convection and conduction.

$$\dot{Q}_i = h_{g,i} A_s (T_i - T_w) \tag{2.1c}$$

$$\frac{dT_w}{dt} m_w c_w = \dot{Q}_i(T_w, T_i) - \dot{Q}'_e(T_w, T_a, T_e) \tag{2.1d}$$

$$T_0 = T_w + (T_i - T_w)e^{-\frac{h_{g,i}A_s}{\dot{m}_c c_p}} \tag{2.1e}$$

$\dot{Q}_e$ are external heat fluxes and $\dot{Q}_i$ are internal. The model is implemented in SIMULINK and T_w is used as a state. T_i and T_o is the inlet and outlet temperature from the pipe, $h_{g,i}$ is the convection heat transfer coefficient inside the pipe, $h_{cv,e}$ is the natural convection heat transfer coefficient, c_w is the thermal storage capacity, m_w is the pipe mass, $\dot{m}_c$ is the mass flow, c_p is the heat storage capacity of the air flowing through the pipe. The values are selected according table 2.1. Model validation is shown in figure 2.2. Parameters used during calculations are selected according to table 2.1. The pipe temperatures is not following the measured data exactly, but are in the same temperature range. The model is assumed to be accurate enough to find trends in the pipes affect on the measured compressor efficiency. The reference measurements and the modeled temperatures will be different, since the model temperatures are the mean temperatures of the pipes, and the measured temperatures are at specific locations and is affected by the temperature gradient along the pipe.

Parameter	Value
$h_{cd,e}$	$700\ W/m^2K$
ρ_{steel}	$7900\ kg/m^3$
$c_{p,steel}$	$477\ W/kgK$
$c_{p,air}$	$1007\ W/kgK$
Pr	0.729
$d_{i\ inlet}$	$60\ mm$
$d_{y\ inlet}$	$62\ mm$
$d_{i\ outlet}$	$49.8\ mm$
$d_{y\ outlet}$	$52\ mm$
L_{inlet}	$0.15\ m$
L_{outlet}	$0.15\ m$

Table 2.1: *Parameters selected for modeling, the fluid and material data is found in Cengel et al. [2008]. The diameters of the inlet and outlet pipe with index y are estimated diameter to the outer surface. $h_{cd,e}$ is calibrated to make the temperatures of the pipes reasonable (see figure 2.2). The lengths of the inlet and outlet pipes are made 15 cm, since there is a surface measurement made on the inlet pipe 15 cm from the compressor.*

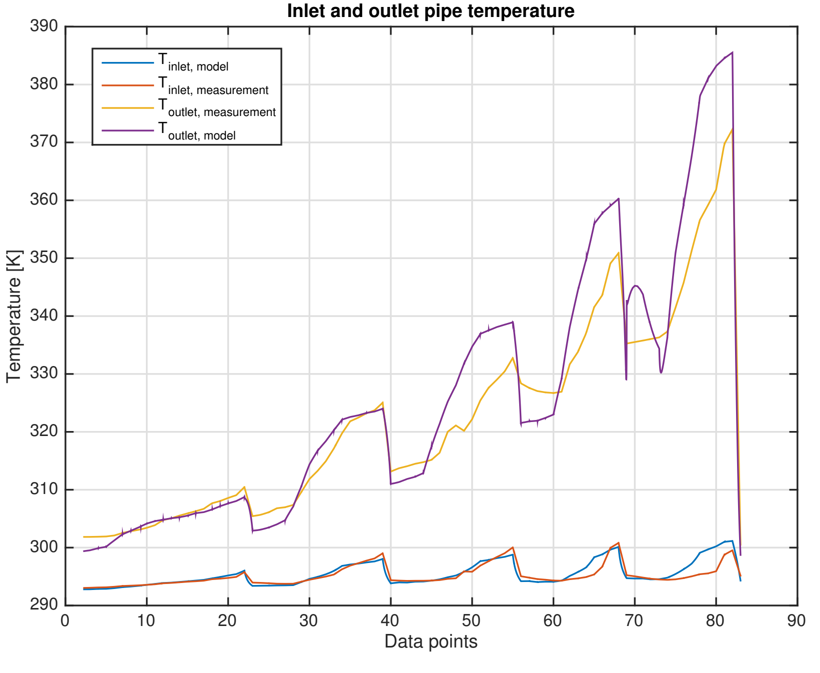

Figure 2.2: *Wall temperature model, equation (2.8), validation. The measured pipe temperature on the inlet is made 15 cm from the compressor, the outlet temperature is made on the compressor outlet (see schematics of the turbocharger in 1.2).*

2.1.2 Calculating heat transfer coefficient $h_{g,i}$

The temperature increase model is used to calculate the change of inlet and outlet air temperature to and from the sensor positions to the compressor. The outlet temperature of the air flowing through the pipe is calculated using equation (2.6), the heat transfer coefficient $h_{g,i}$ is calculated using Nusselts number and the air conduction. The coefficient of convectional heat transfer is calculated using the following equations:

Parameter	Value
ν	$1.562 \times 10^{-5}\ \frac{m^2}{s}$
ρ_{air}	$1.184\ \frac{kg}{m^3}$
k_{xx}	$0.02551 \frac{W}{m} K$

Table 2.2: *Selected properties of air flowing through the pipes, values are found in Cengel et al. [2008]. The properties are selected at 1 atm pressure, 25 ^oC*

$$v_{fluid,avg} = \frac{4\dot{m}_c}{\rho_{air}\pi D^2} \tag{2.3a}$$

$$Re = \frac{v_{fluid,avg}D}{\nu} \tag{2.3b}$$

$$Nu = Pr^{\frac{1}{3}} 0.023 Re^{0.8} \tag{2.3c}$$

$$h_{g,i} = \frac{k_{xx}Nu}{D} \tag{2.3d}$$

Where the average speed for the fluid inside the pipe is calculated in equation (2.3a), the flow coefficient Reynolds number in equation (2.3b), Nusselts number for a circular pipe in equaiton (2.3c) and finally the heat transfer coefficient in equation (2.3d). The parameters in table 2.2 are used in equations (2.3a)-(2.3d). The heat transfer coefficients between the pipes and the flowing air is shown in figure 2.3. The heat transfer coefficient values are different because the pipes have different dimensions. The outlet pipe (the blue stars in figure 2.3) has a smaller diameter, which makes $h_{g,i}$ in equation (2.3d) larger then for the inlet pipe. The heat transfer coefficient (in figure 2.3) tells that the outlet pipe has easier to transfer energy to or from fluid then the inlet pipe, for a certain temperature difference between the pipe wall and the fluid inside the pipe.

2.1.3 Calculating natural heat transfer coefficient $h_{cv,e}$

Natural convection is acting on both the inlet and outlet pipes. The natural convection is calculated according to the following equations:

$$T_{film} = \frac{T_{pipe} + T_{amb}}{2} \tag{2.4a}$$

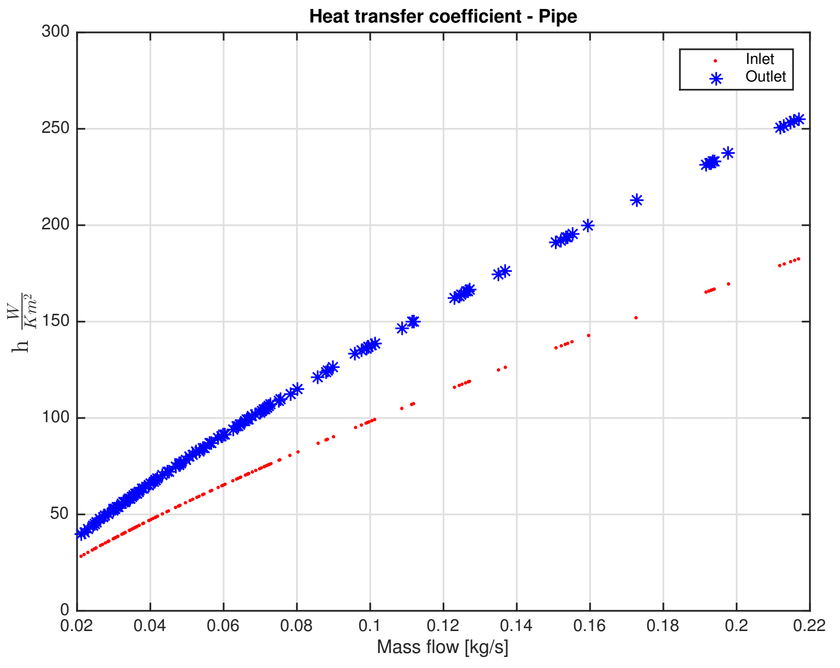

Figure 2.3: *Heat transfer coefficient plotted against mass flow. The outlet pipe has a higher heat transfer coefficient for a given mass flow, than the inlet pipe.*

$$\beta = \frac{1}{T_{film}} \tag{2.4b}$$

$$Ra_D = \frac{g\beta D_{pipe}^3(T_{surface} - T_{amb})Pr}{\nu^2} \tag{2.4c}$$

$$Nu = \left(0.6 + \frac{0.387Ra_D^{1/6}}{(1 + (0.559/Pr)^{9/16})^{8/27}}\right)^2 \tag{2.4d}$$

$$h_{cv,e} = \frac{k_{xx}Nu}{D_{pipe}} \tag{2.4e}$$

Data used in equations (2.4a)-(2.4e) are described in table 2.1 and table 2.2. The equations (2.4a)-(2.4e) are found in Cengel et al. [2008]. T_{film} is the film temperature on the pipe, T_{pipe} is the pipe temperature, Ra_D is the Rayleigh number, Nu is the Nusselt number and D_{pipe} is the outer diameter of the pipe. The resulting natural heat transfer coefficient is shown in figure 2.4.

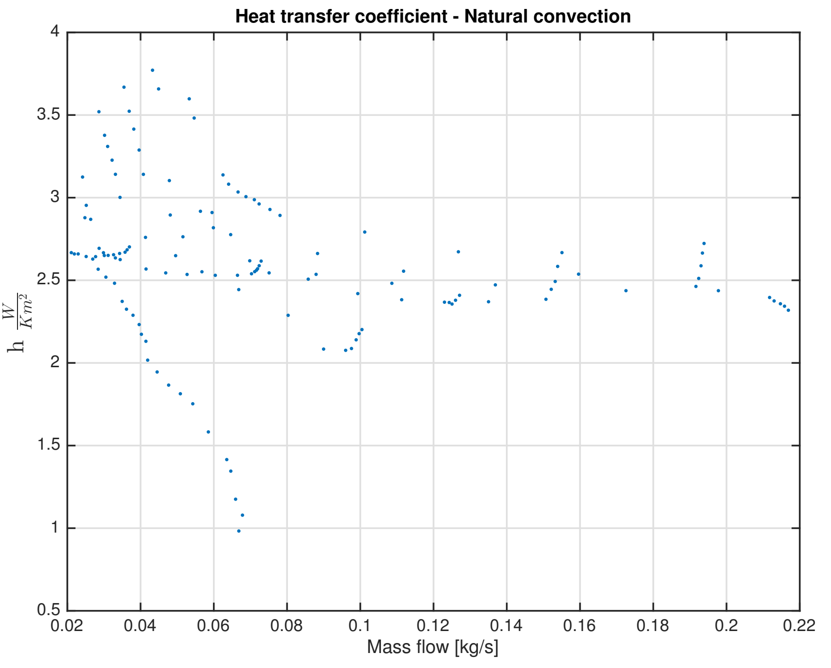

Figure 2.4: *The natural heat transfer coefficient acting between the air and the pipes. The heat transfer coefficient is plotted against the mass flow. The heat transfer coefficient is assumed to be the same for the inlet and outlet pipes.*

2.1.4 Pressure drop model

The pressure drop model is used to calculate the pressure drop in the inlet and outlet pipe, to and from the sensor position all the way to the compressor housing. The main equation (2.5a) that is used to calculate the pressure loss, depends on the friction factor f and the fluid mean speed $v_{fluid,avg}$. The surface roughness is unknown, the pipes are assumed to be stainless steel, the surface roughness ε is estimated to be $\varepsilon = 0,002$ mm, according to Cengel et al. [2008], p 546, TABLE 14-1, Material: Stainless steel. The pressure loss is calculated using:

$$\Delta p_L = f \frac{L \rho v_{fluid,avg}^2}{2D} \tag{2.5a}$$

Where L is the selected pipe length, ρ is the density of air, $v_{fluid,avg}$ is the mean velocity of the air flowing through the pipe and D is the selected pipe diameter. The friction factor f is calculated in different ways depending on if the flow is laminar or turbulent. The flow is considered laminar if $Re < 2300$, the region between $2300 < Re < 10000$ is a region where the flow is called transitional,

the flow is changing from laminar flow to turbulent. If $Re > 2300$, the flow is considered to be turbulent in this thesis.
At low fluid velocities (low Re), the flow is laminar. When the flow is laminar, the friction factor is considered to have a linear relation to the fluid velocity. The friction factor at laminar flow is:

$$f = \frac{64}{Re} \tag{2.5b}$$

At higher fluid velocities (higher Re), the flow is considered to be turbulent. When the flow is turbulent, Colebrook equation (2.5c) is used to calculate the friction factor. But to get a reliable result, the equation should be iterated until the stable factor f is found. This iteration is not implemented, instead the friction factor f is calculated using equation (2.5d). The Colebrook equation (Cengel et al. [2008], p.545) :

$$\frac{1}{\sqrt{f}} = -2.0log\left(\frac{\varepsilon/D}{3.7} + \frac{2.51}{Re\sqrt{f}}\right) \tag{2.5c}$$

The surface roughness is estimated to be $\varepsilon = 0,002$ mm:

$$\frac{1}{\sqrt{f}} = -1,8log\left(\frac{6,9}{Re} + (\frac{\varepsilon}{3.71D})^{1,11}\right) \Longleftrightarrow \tag{2.5d}$$

$$f = \left(\frac{1}{-1,8log\left(\frac{6,9}{Re} + (\frac{\varepsilon}{3,7D})^{1,11}\right)}\right)^2 \tag{2.5e}$$

Equation (2.5b) is found in Cengel et al. [2008], p.540. Equation (2.5d) is an approximate explicit relation for the friction factor according to Cengel et al. [2008], p.546. The result should be within 2 % compared to equation (2.5c). The Reynolds number is calculated using:

$$Re = \frac{v_{fluid,avg}D}{\nu} \tag{2.5f}$$

The average fluid velocity is calculated using equation (2.3a). The resulting power loss due to friction in pipes is given by:

$$\dot{W}_{f,pipe} = \dot{V}\Delta p_L = \frac{\dot{m}_c \Delta p_L}{\rho_{air}} \tag{2.5g}$$

Where $\dot{m}_c$ is the mass flow of air through the pipe, Δp_L is the pressure drop in the pipe and ρ_{air} is the air density. The power loss from the air due to friction is transfered into the pipe walls, which results in a heat increase. This effect is already accounted for when calculating the heat transfer coefficient (see 2.1.2). The pressure drop in the inlet and outlet pipes are very small compared to the pressure difference over the compressor, but still included in the model. The resulting pressure drop in both the inlet and outlet pipes is plotted against the mass flow and shown in figure 2.5. In the inlet pipe the pressure decreases if moving in the same direction as the flow, towards the compressor. In the outlet pipe the pressure also decreases if moving along the pipe, in the same direction as the flow. Both the inlet and outlet pipe seems to be dependent of the mass flow in square, since it is a fluid flowing in the pipe it seems reasonable if looking at equation (2.5a), where the average fluid velocity $v_{fluid,avg}$ is in square. It also seems reasonable due to the friction involved being viscous friction more then mechanical friction.

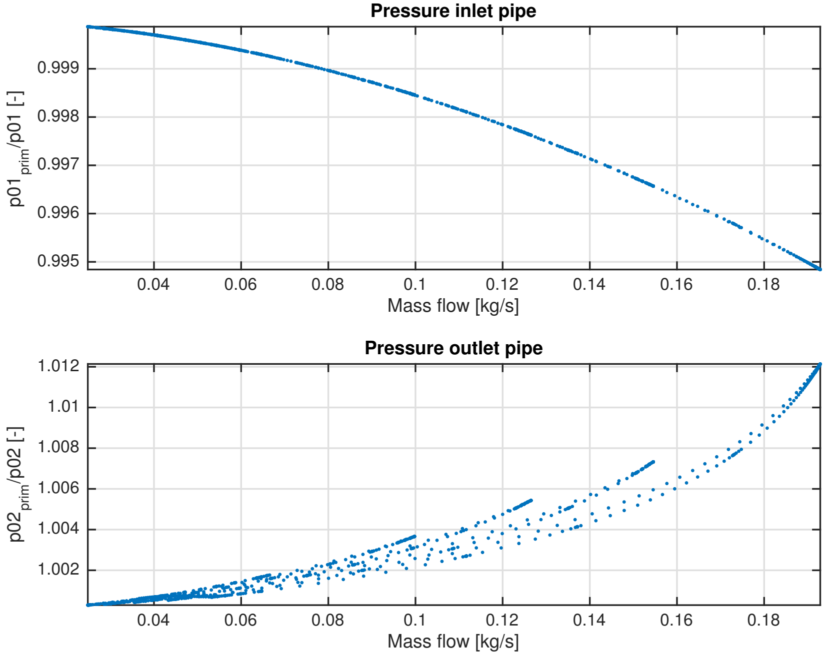

Figure 2.5: *Pressure drop in both the inlet and outlet pipes, plotted against mass flow. Both plots shows the pressure close to the compressor divided with the measured pressure at the inlet and outlet. The pressure close to the compressor inlet gets lower with increasing mass flow, the pressure close to the compressor outlet gets higher with increasing mass flow. The inlet and outlet pipe lengths are 1 m.*

2.1.5 Temperature change model

When the heat transfer coefficients for the two pipes are calculated, the outlet temperatures of the air traveling through the pipes can be calculated according to equation (2.6). The air temperature leaving the inlet pipe and entering the compressor is denoted T'_{01}, the air leaving the compressor and entering the outlet pipe is denoted T'_{02}. The power loss from the air due to the friction in the pipe is transfered to the pipe wall, this effect is included in the heat transfer coefficient $h_{g,i}$.

$$T_0 = T_w + (T_i - T_w)e^{-\frac{h_{g,i}A}{\dot{m}_c c_p}} \tag{2.6}$$

Parameter	Description
T_0	The temperature of the fluid at pipe outlet.
T_i	The temperature of the fluid at pipe inlet.
T_w	The pipe wall temperature.
A	Surface area inside pipe (area in contact with fluid).
$h_{g,i}$	Heat transfer coefficient (different for inlet and outlet pipe).
$\dot{m}_c$	Mass flow of air through the pipe.
c_p	Heat storage capacity for pipe.

Table 2.3: *Parameters in equation (2.6).*

The resulting outlet temperatures when using equation (2.6) are shown in figure 2.6 and 2.7. The inlet air temperature to the inlet pipe is denoted T_{01} and the outlet temperature is denoted T'_{01}. The inlet air temperature to the outlet pipe is denoted T'_{02} and the outlet temperature is denoted T_{02}. Both the inlet and outlet air is changing temperature through the inlet and outlet pipe, this can be seen if comparing T'_{01} and T'_{02} with the measurement value T_{01} and T_{02} as seen in figure 2.6 and 2.7. The temperature change in the inlet and the outlet pipes are different, the different temperature changes depends on the different heat transfer coefficients $h_{g,i}$ (see section 2.1.2) , different pipe temperatures (see figure 2.2) and the different pipe lengths (see table 2.1).

2.2 Modeling the compressor heat transfer

The compressor is affected by many different heat fluxes, it is connected to two pipes and the bearing house which conducts heat, it radiates heat to the surroundings and loses energy due to convection to the air around it. During the compression of air, energy is transfered to and from the air from the backplate of the compressor, where the bearing house is connected. When the compression is complete, the air is hotter then the air entering the compressor, and this may rise the temperature of the compressor housing. To calculate the compressor housing temperature, two different models has been developed, first a static model, dependent of mass flow $\dot{m}_c$ and the compressor outlet temperature T'_{02}. The second model is a dynamic model, dependent of the internal and external heat fluxes.

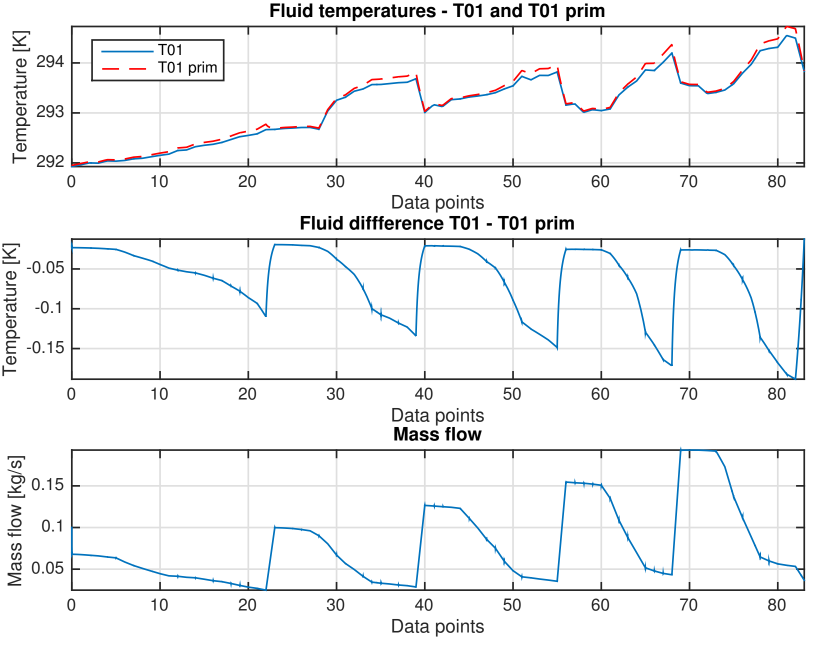

Figure 2.6: *Air temperatures in and out from inlet pipe. T_{01} is the temperature entering the pipe, T'_{01} is the air leaving the pipe. The lower plot shows the temperature difference $T_{01} - T'_{01}$. The air flowing through the pipe is heated after the actual value of the air temperature is measured.*

2.2.1 Static model

This model is static and only dependent of the mass flow and the inlet temperature from the inlet pipe. It doesn't take the conduction between the different subsystems into account (the inlet and outlet pipes and the bearing house). But it still gives a quite good fit compared to the measurement data, see figure 2.8. The model is fitted to the data with the constants k_1 and k_2. The constants are tuned until a satisfying result is achieved. The model equation (2.7) gives the output in figure 2.8. This model is very dependent of the outlet temperature T_{02}, that is measured at the outlet of the outlet pipe. It might not be a good idea to use this model since the pipe temperature also may depend on the compressor housing temperature, which might be affected by the conduction from the bearing house.

$$T_{compressor} = \frac{k_1}{\dot{m}_c c_{p,c} T_{02}} + k_2 T_{02} \tag{2.7}$$

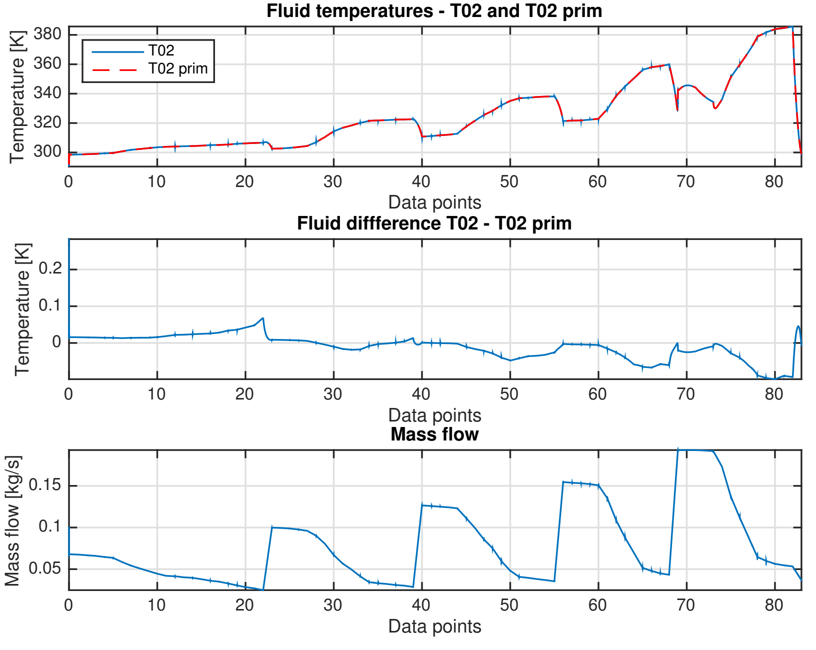

Figure 2.7: Air temperatures in and out from inlet pipe. $T02\ prim$ is the temperature of the air entering the pipe, $T02$ is the temperature of the air leaving the pipe. The lower plot shows the temperature difference $T_{02} - T'_{02}$. The air flowing through the pipe is heated and cooled before the actual air temperature is measured.

2.2.2 Dynamic model

The second compressor temperature model is a dynamic lumped mass model, using the thermal mass and the heat storage coefficient to calculate heat storage capacity. The change in energy is calculated by the heat flux balance. The equations to calculate the compressor temperature are described here:

$$\frac{dT_{compressor}}{dt} = \frac{1}{m_c c_{p,c}} \left(\dot{Q}_{external} - \dot{Q}_{internal} \right) \tag{2.8}$$

External heat transfer

The external heat transfer is taking place between the compressor and surrounding environment, it includes convection, conduction and radiation from connected parts. Both the radiation and convection are very small since the compressor housing temperature is close to ambient temperature. These two factors are included even though the measurements are made with the turbocharger wrapped

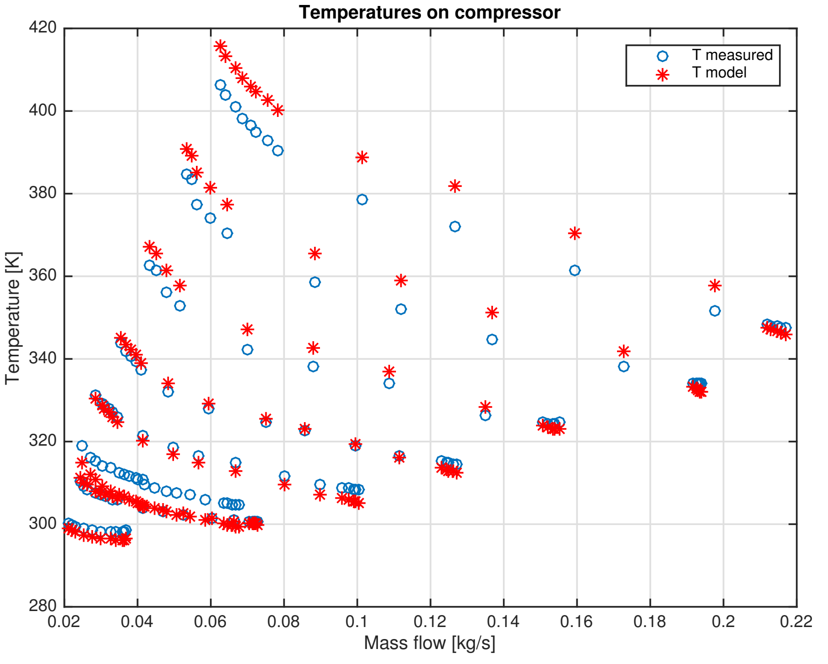

Figure 2.8: *Static model of compressor temperature,* $k_1 = 4.8$ *and* $k_2 = 1$ *gives the displayed result. The model fit is better for lower air temperatures and mass flows but still not sufficient.*

in low conductive cord, to give the capability to experiment with the model and have the ability to simulate the heat losses from the compressor housing to the environment.

$$\dot{Q}_{external} = \dot{Q}_{radiation} + \dot{Q}_{convection} + \dot{Q}_{conduction} \tag{2.9}$$

Internal heat transfer

The internal heat transfer is the heat transfered from the air flowing through the compressor. The air is, dependent of its temperature, taking or leaving energy to the compressor housing. The energy from the bearing housing is entering the compressor as an internal heat flux because it is entering directly into the backside of the compressor housing.

$$\dot{Q}_{internal} = h_{compressor} A(T_{mean,air} - T_{comp}) \tag{2.10}$$

The heat coefficient $h_{compressor}$ is calculated in the same way as 2.1.2, the flow path through the compressor is assumed to be a pipe with diameter $D = 30mm$.

The surface convection is plotted against mass flow and shown in figure (2.9) and calculated according to section 2.1.2.

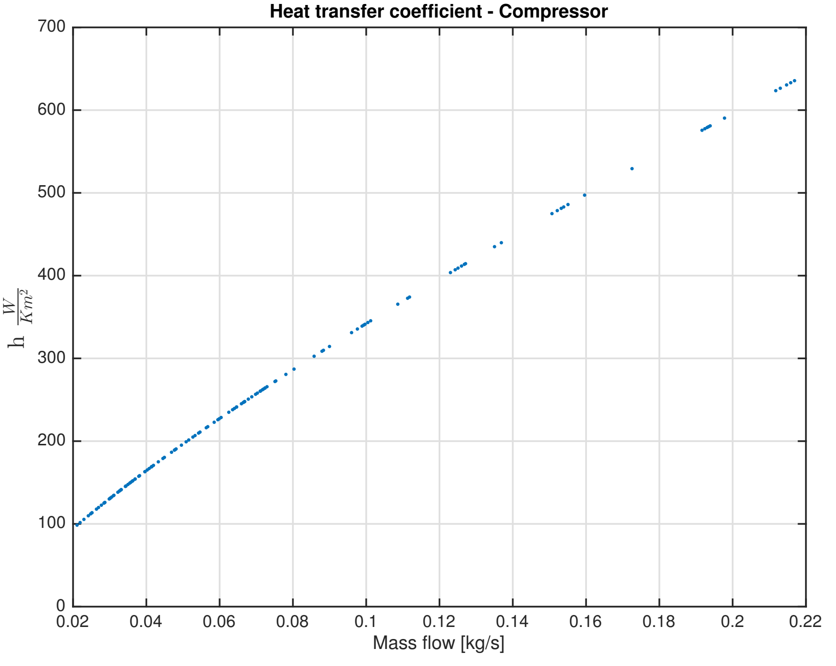

Figure 2.9: *Calculated heat transfer coefficient h that acts between the wall surface and the air inside the compressor, plotted against mass flow.*

Compressor temperature

It turned out to be very hard to tune the heat transfer coefficients, so the compressor temperature is assumed to be known. When the dynamic compressor temperature model and the dynamic bearing housing model (see appendix A, for a description of the bearing housing model) is connected, the system gets very hard to tune. The measured compressor temperature is used to calculate the heat conducted to or from the inlet and outlet pipes.

2.3 Modeling the compressor

The compressor model that is used is from Leufvén and Eriksson [2013]. The model is used as a tool in which the compressor data is loaded, and the modeled compressor behavior is optimized to fit the loaded data. The compressor model is loaded with data from a gasstand measurement made with the TD04HL-15T turbocharger, where the inlet temperature at the turbine is 750 ^{o}C. The model fit to the selected data is shown in figure 2.10. The model gives a good fit to the input data. The compressor data is loaded into the SIMULINK environment to describe the compressor behavior in the simulation model. The main reason to have the compressor model is to get the outlet temperature T'_{02}, outlet pressure p'_{02} and the mass flow $\dot{m}_c$ for given input temperature T'_{01}, input pressure p'_{01} and shaft speed N_{tc}.

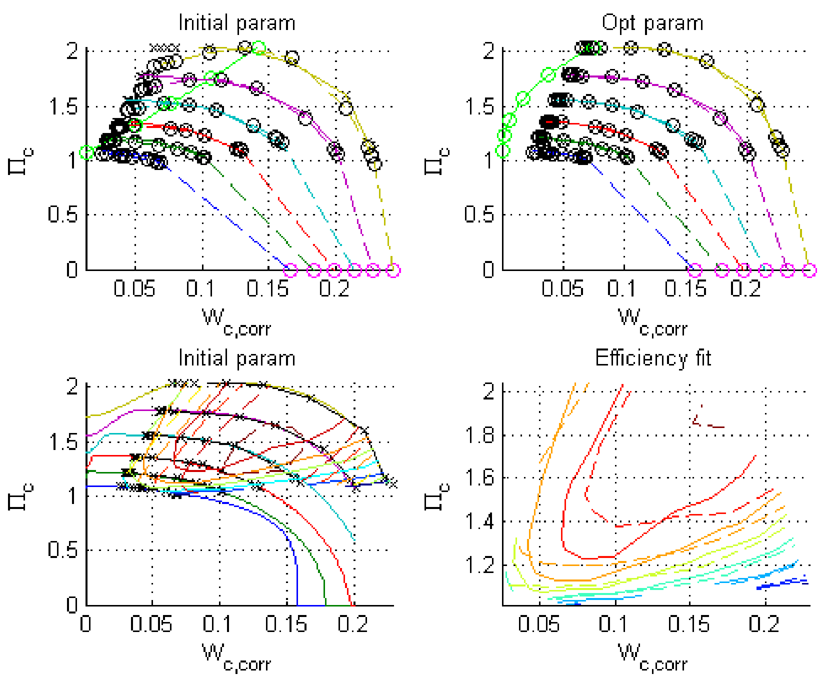

Figure 2.10: *Curve fit of the compressor model. The circles are the measured data used for model parametrization and validation, the colored lines are the curve-fitted compressor model at different shaft speeds. Top left and right shows the guess and the optimized model. The bottom left shows the optimized flow parameters, and the efficiency estimate plot. The bottom right shows the efficiency reference and the optimized efficiency model, reference data in solid lines.*

2.3.1 Controlling the compressor model

Mass flow controlled model

To control the compressor model and make it follow a given reference, a PD controller is used to make the model give the demanded mass flow. The measurement input to the regulator is the mass flow in the compressor, the reference is the measured mass flow $\dot{m}_c$ from the gasstand data, the control signal is the area of a restriction. Since the model data uses measured compressor temperature, the compressor needs to be running in those work points where measured temperatures exists.

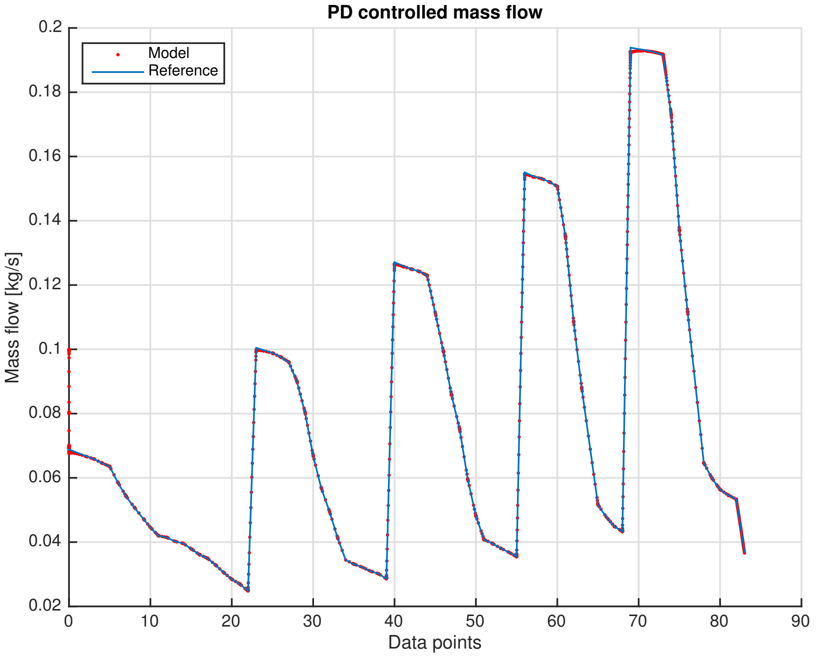

Figure 2.11: *Reference following for the PD-controller. The model mass flow follows the reference very good, the PD-controller works good.*

2.3.2 Implementation of the model

The compressor model is integrated in SIMULINK, and receives the temperature after the inlet pipe T'_{01}, the pressure after the inlet pipe p'_{01} and the desired rotational speed of the compressor shaft N_{tc}. The output from the model is the temperature out from the compressor, T'_{02}, the pressure after the compressor p'_{02} and the mass flow.

2.3.3 Modeling control volume

In order to make the simulation running, there is need of a control volume to connect the outlet pipe to the compressor. The control volume is needed by the compressor model, since it's controlled by a restriction at the end of the outlet pipe. The control volume is found in Eriksson and Nielsen [2014] modeled according to:

$$\frac{dT}{dt} = \frac{RT}{pVc_v}\left(\dot{m}_{in}c_v(T_{in} - T) + R(T_{in}\dot{m}_{in} - T\dot{m}_{out}) - \dot{Q}\right) \tag{2.11a}$$

$$\frac{dp}{dt} = \frac{RT}{V}(\dot{m}_{in} - \dot{m}_{out}) + \frac{p}{T}\frac{dT}{dt} \tag{2.11b}$$

Where R is the gas constant, T is the air temperature leaving the volume, V is the specific volume, p is the pressure leaving the volume, $\dot{m}_{in}$ and $\dot{m}_{out}$ are the mass flows in and out from the control volume, T_{in} is the temperature entering the volume, $\dot{Q}$ is a external energy input that isn't used and c_v is the specific heat for constant volume of air.

2.4 Connecting the subsystems

The connections of the subsystems are explained in the following chapter. The inlet and outlet pipes are connected to the compressor model and the compressor temperature is sent into the pipe models. The inlet pipe is connected directly to the compressor model, the outlet pipe is connected at the compressor model outlet. Between the compressor outlet and the outlet pipe, it is a control volume (see section 2.3.3) to make the simulation runnable. After the outlet pipe it is a restriction, that creates counter pressure at the end of the pipe. This restriction is controlled to make the compressor model give the demanded mass flow.

2.4.1 System descriptions

Mass flow controlled model

This model have the input and output signals according to table 2.4. The connection setup in the SIMULINK model is shown and described in figures 2.12 and 2.13.

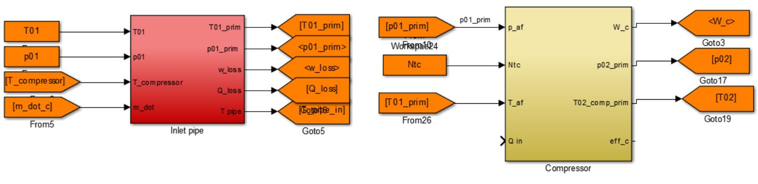

Figure 2.12: *The inlet pipe is red and the compressor model is yellow. The outlet pipe is implemented inside the compressor block, see figure 2.13.*

Input	Output
T_{01}	T_{01}'
p_{01}	p_{01}'
N_{tc}	T_{02}'
$\dot{m}_{c,desiered}$	p_{02}'
$T_{bearing}$	T_{02}
T_{amb}	p_{02}
$T_{compressor}$	$T_{inlet\ pipe}$
	$T_{outlet\ pipe}$

Table 2.4: *Input and output data for the compressor model with connected pipes.*

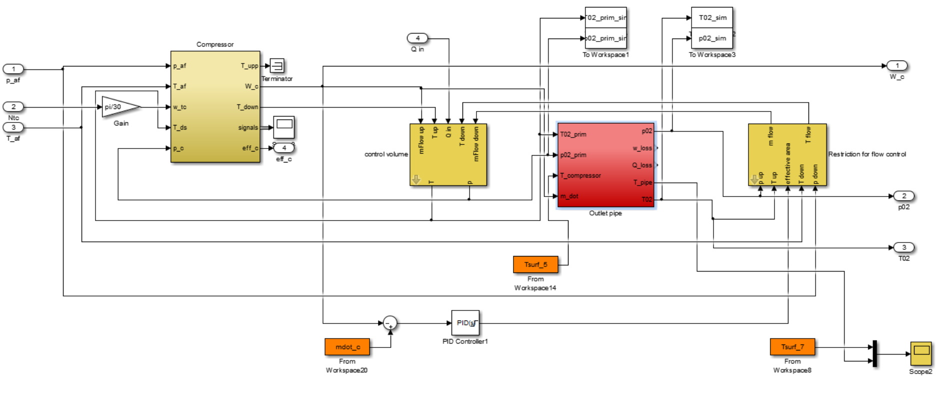

Figure 2.13: *Inside of the top center compressor model in figure 2.12. The red block is the thermodynamics of the outlet pipe, the yellow block to the left is the compressor model, the center yellow block is a control volume and the right yellow block is a restriction used to control the model.*

3

Result

The results from the compressor model with connected pipes are presented in this chapter, and further discussed in chapter 3.5. The result shows that the effects of including the inlet and outlet pipes when calculating the compressor efficiency are noticeable. The effects of the pressure drop in the pipe and the effects of temperature change will be separately presented. The pipes will be investigated separately from the model to find the main impacts on the compressor efficiency.

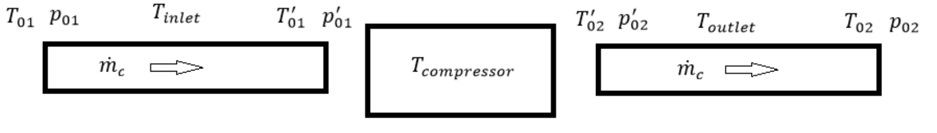

Figure 3.1: *The notations of the air entering and leaving the inlet and outlet pipes. T_{inlet}, T_{outlet} and $T_{compressor}$ are the temperatures of the subsystems. The notations with a prim (') are the values of interest, because those are the temperatures and pressures actually entering and leaving the compressor.*

3.1 Model fit to measured data

The model for the compressor with the pipes is validated with measured values for the entering and leaving air. The entering temperature, pressure, demanded shaft speed and the the demanded mass flow affects the model output temperature and pressure. The total validation of the model compared to the measurement data is described in the following subsections. The inlet and outlet measurements from the gasstand are compared to the model inlet and outlet data.

3.1.1 Temperature validation

Figure 3.2 shows the outlet temperatures from the model and from the gasstand measurements. The inlet temperature, inlet pressure, mass flow and shaft speed are the same for the model and the measurement. The reference temperature is a bit higher than the model temperature at lower mass flows, this is due to the calibrated heat transfer coefficient from the compressor to the pipes and also the selected lengths of the inlet and outlet pipes. Even thou the peaks in outlet temperature is not fulfilled by the model, the validations seems sufficient enough to investigate the affects of the inlet and outlet pipes.

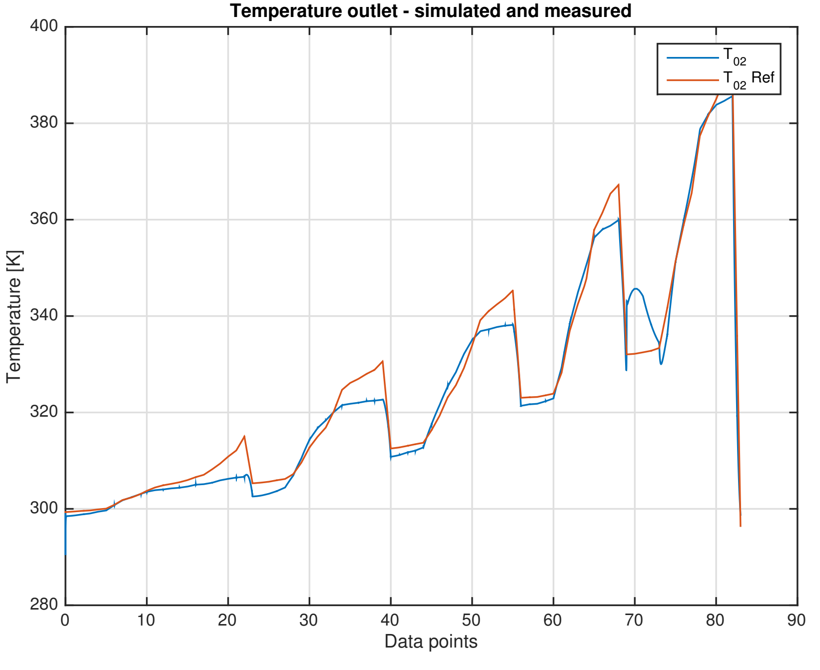

Figure 3.2: *Outlet temperature T_{02}, the model output is compared to measured data. T_{02} Ref is the measured outlet temperature and T_{02} is the calculated temperature leaving the outlet pipe.*

3.1.2 Pressure validation

Figure 3.3 shows the outlet temperatures from the model and from the gasstand measurements. The inlet temperature, inlet pressure, mass flow and shaft speed are the same for the model and the measurement. The pressure has a good fit to the measured outlet pressure. The model fit is dependent of the surface roughness ε, the pipe internal diameters and the length of the pipes.

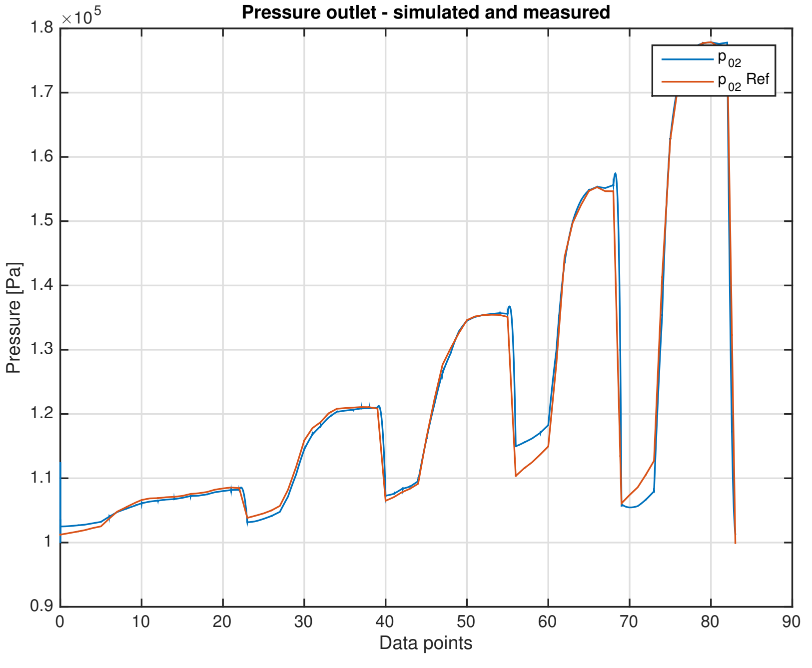

Figure 3.3: *Outlet pressure p_{02}, the model output is compared to measured data. p_{02} Ref is the measured outlet pressure and p_{02} is the calculated pressure leaving the outlet pipe*

3.2 Compressor efficiency

The compressor efficiency is calculated for the measured temperatures and pressures T_{01}, p_{01}, T_{02} and p_{02} and compared to the efficiency calculated with T'_{01}, p'_{01}, T'_{02} and p'_{02}. Equation used for calculating compressor efficiency (from Eriksson and Nielsen [2014]):

$$\eta_c = \frac{(\frac{p_{02}}{p_{01}})^{\frac{\gamma-1}{\gamma}} - 1}{\frac{T_{02}}{T_{01}} - 1} \tag{3.1}$$

The efficiency for the calculated temperatures and pressures at the inlet and outlet of the compressor is made according to:

$$\eta'_c = \frac{(\frac{p'_{02}}{p'_{01}})^{\frac{\gamma-1}{\gamma}} - 1}{\frac{T'_{02}}{T'_{01}} - 1} \tag{3.2}$$

When looking at figure 3.4, the efficiency is higher when using the calculated (prim) values than the standard efficiency calculated with the measured values

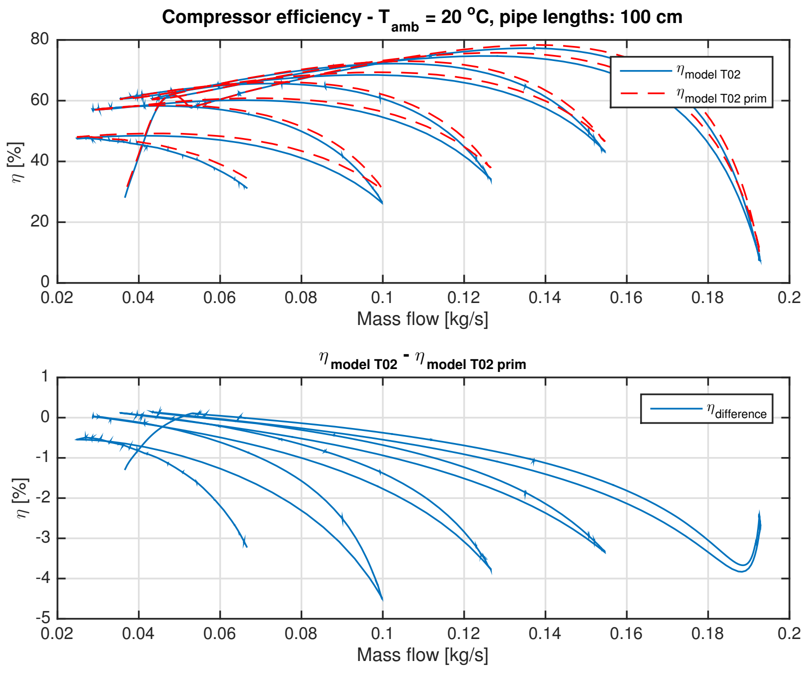

Figure 3.4: *Compressor efficiency when comparing the data from the inputs and outputs T_{01}, T'_{01}, T_{02}, T'_{02}, p_{01}, p'_{01}, p_{02} and p'_{02}. All these parameters are used to calculate the efficiency according to eq. (3.1) and eq. (3.2). The lower plot window in the figure shows the sum of the prim-values subtracted from the measured values. The efficiency is plotted against mass flow, each line in the figure corresponds to a speed line in the compressor map.*

before and after the pipes. Both at lower and higher speeds the efficiency is affected. In figure 3.5, the efficiency is shown when the pipe lengths are 50 cm. During lower mass flows, the main impact is the temperature change of the air flowing through the inlet and outlet pipes. In the presented case in figure 3.4, both the inlet and outlet pipes are made very long to clearly show the temperature change effect. When the mass flow increases, the temperature change of the air in the pipes decreases, but still the efficiency seems to be higher. This is due to the pressure losses, as the mass flow increases, the friction factor in the pipes increases and thus the pressure drop in the pipes.

3.3 Main impacts on compressor efficiency

The change in efficiency due to the pipes are divided into two main dependencies, the pressure dependence and the temperature dependence. Deeper investigation in these dependencies are made and the effects on the compressor efficiency is

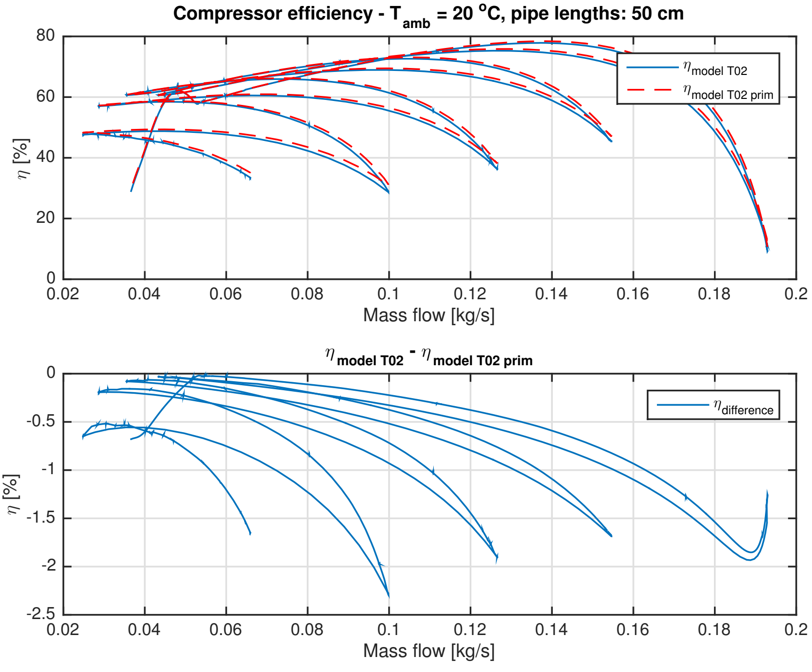

Figure 3.5: *Compressor efficiency when comparing the data from the inputs and outputs* T_{01}, T'_{01}, T_{02}, T'_{02}, p_{01}, p'_{01}, p_{02} *and* p'_{02}. *All these parameters are used to calculate the efficiency according to eq. (3.1) and eq. (3.2). The lower plot window in the figure shows the sum of the prim-values subtracted from the measured values. The efficiency is plotted against mass flow, each line in the figure corresponds to a speed line in the compressor map.*

connected to these two dependencies in different ways.

3.3.1 Pressure dependent effect

The pressure drop in the pipe affects the efficiency at higher mass flows. The pressure drop in the pipe is strongly dependent on the volume flow, the length of the pipe and the diameter of the pipe. Experiments with different pipe lengths has been performed. The results show that at low mass flows, the efficiency isn't affected as much as at high mass flows. In figures 3.6 to 3.9 the temperature change of the air in the pipes are neglected, the change in efficiency is affected by the pressure drop in the pipes only. At low mass flows, the pressure drop is not affecting the efficiency that much. With increasing mass flow, the efficiency change increases. If the length of the pipes are changed, the efficiency gets worse by increasing pipe length, this is shown in figures 3.6 to 3.9. This is because the pressure loss is dependent of the pipe length. The relation between the pipe length and pressure drop is linear (see equation 2.5a), so if the pipe length is made twice as long as a reference pipe, the pressure drop in the pipes are twice as big. A larger pressure drop in the pipes affects the calculated values p'_{01} and p'_{02} more. Figures 3.6 to 3.9 shows the efficiency, with increasing pipe lengths, the lengths are 10 cm, 20 cm , 40 cm and 100 cm.

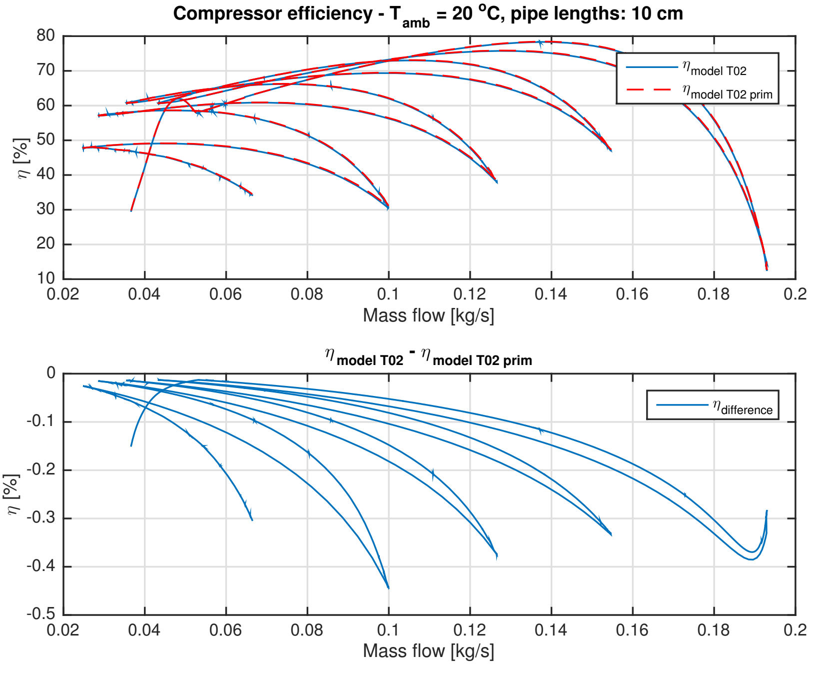

Figure 3.6: *Compressor efficiency with and without the pipes. The efficiency is calculated with measured values before and after the inlet and outlet pipes ($\eta_{model\ T_{02}}$) and the values directly before and after the compressor ($\eta_{model\ T_{02}\ prim}$). The pipe lengths are 10 cm.*

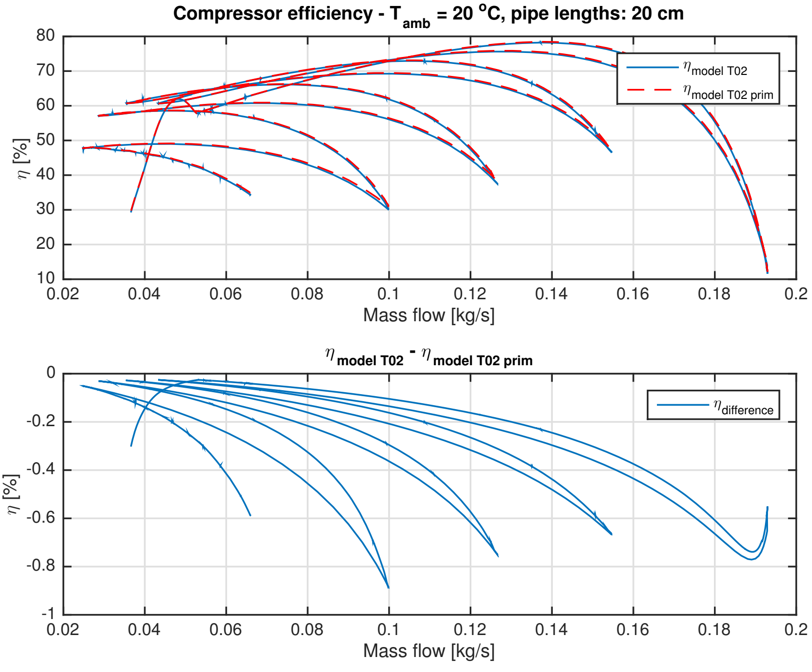

Figure 3.7: *Compressor efficiency with and without the pipes. The efficiency is calculated with measured values before and after the inlet and outlet pipes ($\eta_{model\ T_{02}}$) and the values directly before and after the compressor ($\eta_{model\ T_{02}\ prim}$). The pipe lengths are 20 cm.*

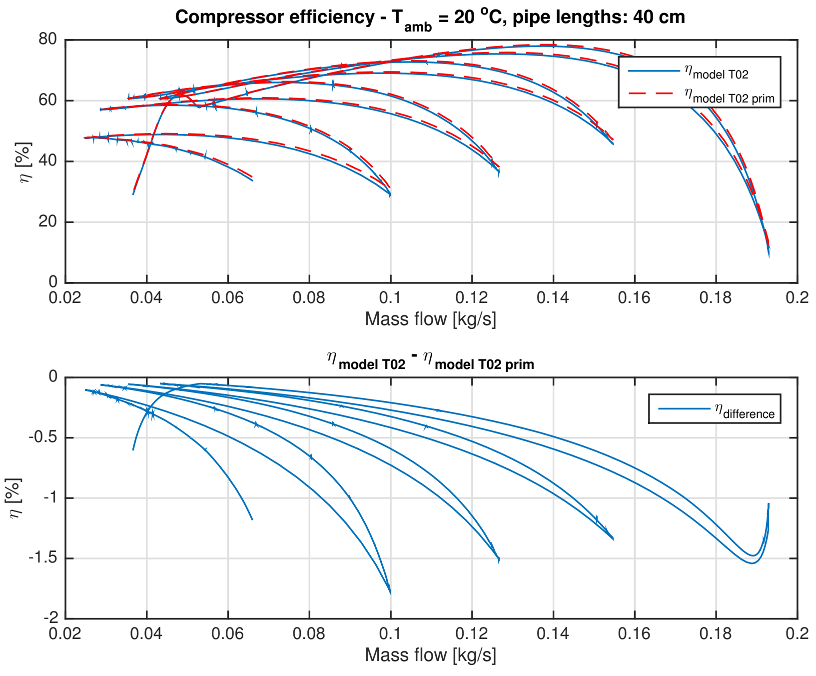

Figure 3.8: *Compressor efficiency with and without the pipes. The efficiency is calculated with measured values before and after the inlet and outlet pipes ($\eta_{model\ T_{02}}$) and the values directly before and after the compressor ($\eta_{model\ T_{02}\ prim}$). The pipe lengths are 40 cm.*

3.3.2 Temperature dependent effect

The temperature change of the air when traveling through the inlet and outlet pipes affects the compressor efficiency. The inlet air is heated and the outlet air is both heated and cooled when traveling through the pipes. If neglecting the pressure loss in the pipes, the air in the pipes is affected by temperature change only. Figures 3.10 and 3.11 show the compressor efficiency at two different ambient temperatures. With increasing ambient temperature, the efficiency is clearly affected. The increase of 10 ^oC ambient temperature is affecting the efficiency at low shaft speed and low mass flow up to 0.2%.

3.4 Inlet and outlet pipe data

The temperature change of the air entering and leaving the inlet and outlet pipes are shown in figure 3.12. At low mass flows, the temperature increase in the inlet pipe is higher than at higher mass flows, the same is for the outlet pipe, but in

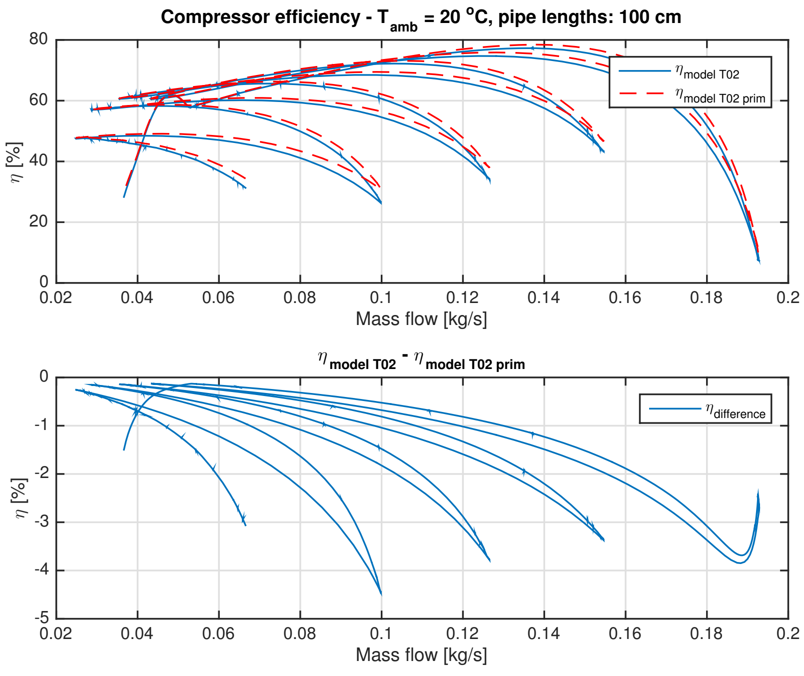

Figure 3.9: *Compressor efficiency with and without the pipes. The efficiency is calculated with measured values before and after the inlet and outlet pipes ($\eta_{model\ T_{02}}$) and the values directly before and after the compressor ($\eta_{model\ T_{02}\ prim}$). The pipe lengths are 100 cm.*

the outlet pipe the air is cooled. These small temperature changes in the pipes get visual effects on the compressor efficiency (see figure 3.10 and 3.11), in the figures the pressure drop is neglected, if looking at the lower mass flows, there is a difference in the compressor efficiency, which doesn't occur at higher mass flows. The pressure loss in the pipes increases with increasing mass flow. At low mass flows, the pressure loss in the pipe is low, at higher mass flows, the pressure loss gets higher, this can be seen in figure 3.13.

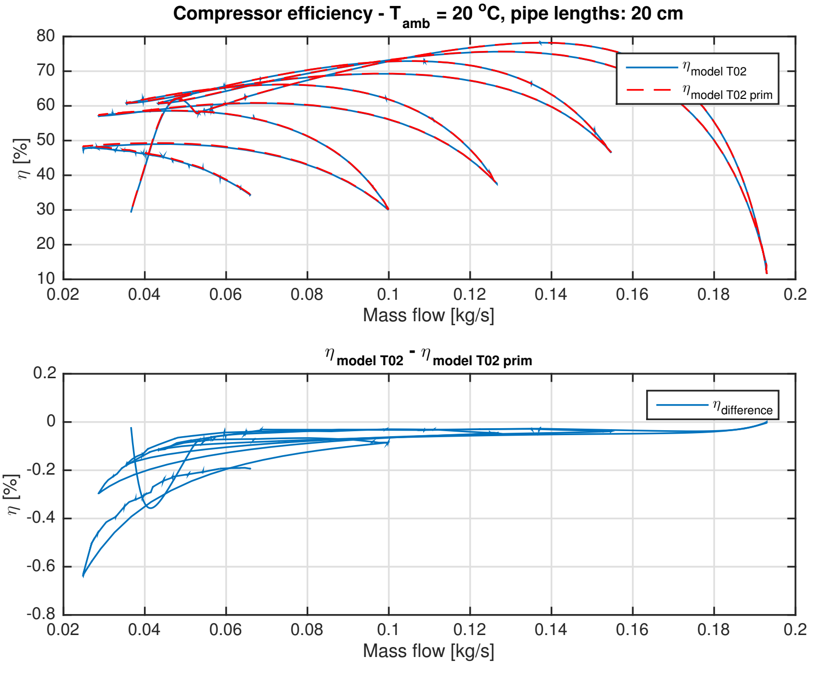

Figure 3.10: *Compressor efficiency with and without the pipes. The efficiency is calculated with measured values before and after the inlet and outlet pipes ($\eta_{model\ T_{02}}$) and the values directly before and after the compressor ($\eta_{model\ T_{02}\ prim}$). The ambient temperature is 20 ^oC.*

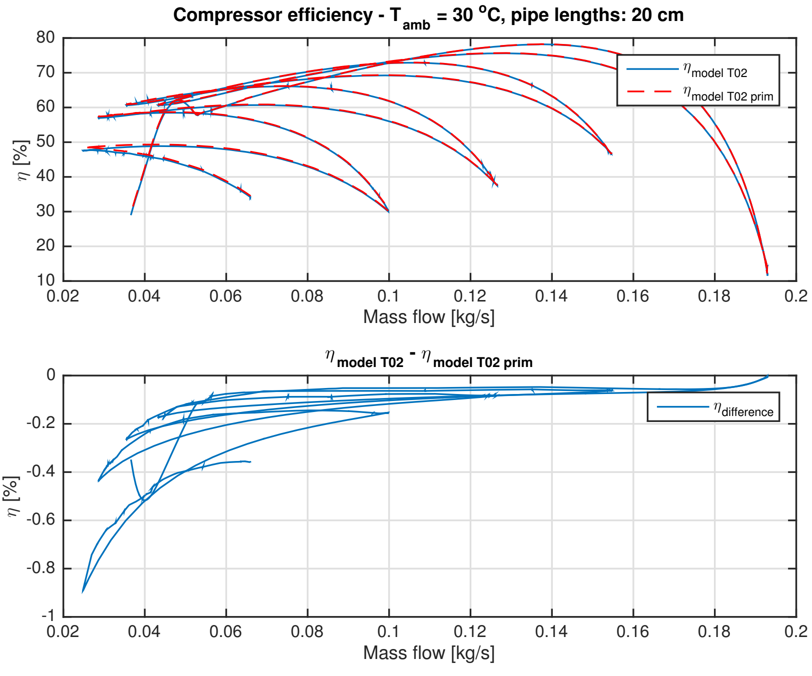

Figure 3.11: *Compressor efficiency with and without the pipes. The efficiency is calculated with measured values before and after the inlet and outlet pipes ($\eta_{model\ T_{02}}$) and the values directly before and after the compressor ($\eta_{model\ T_{02}\ prim}$). The ambient temperature is 30 ^{o}C.*

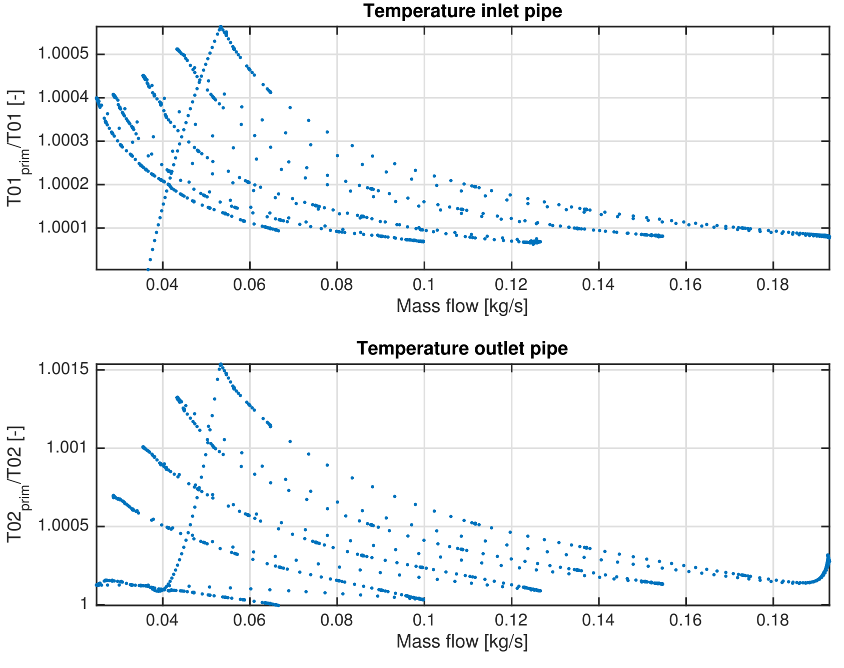

Figure 3.12: *Inlet and outlet air temperature change in pipe, plotted against mass flow. At low mass flows the air in the inlet pipe is heated and the air in the outlet pipe is cooled. The ambient temperature is 20$^o C$ and the pipe lengths are 1m.*

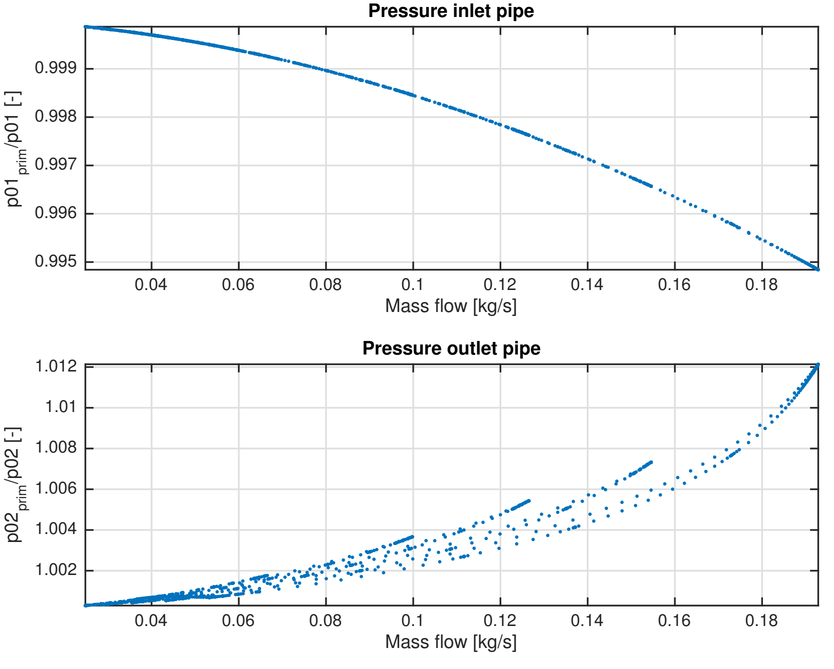

Figure 3.13: *Pressure drop in both the inlet and outlet pipes, plotted against mass flow. Both plots shows the pressure close to the compressor divided with the measured pressure at the inlet and outlet. The pressure close to the compressor inlet gets lower with increasing mass flow, the pressure close to the compressor outlet gets higher with increasing mass flow. The inlet and outlet pipe lengths are 1 m.*

3.5 Discussion

The following is discovered when studying the results:

- The inlet and outlet pipes are affecting the measured performance.
- The pressure loss in the pipes affects the measured efficiency. If using different piping setups in different gasstands during measurements, the results could be different depending on the suppliers test setup.

When analyzing the pressure data from the pipes in figure 3.13, it shows:

- The inlet pressure p'_{01} decreases with increasing mass flow.
- The outlet pressure p'_{02} increases with increasing mass flow.

The pressures p'_{01} and p'_{02} are important factors in equation (3.2). If p'_{02} increases and p'_{01} decreases, the calculated efficiency increases. The pipe surface roughness seems to be important, if the pipe surface is smoother, the pressure loss in the pipes will decrease.

The model result shows that the measurement positions have great importance on how the compressor efficiency looks, when taking the pipes into account. The difference between the measured efficiency and the efficiency based on the values actually entering and leaving the compressor, increases with increasing mass flow, the measurement error gets bigger and bigger. This seems reasonable since the pipe friction increases with increasing mass flow (according to 3.13). The model result shows that if only using gasstand data to calculate performance with the measured temperatures and pressures, and not taking the pipes into account, the efficiency could in reality be greater at both low and high mass flows. At high mass flows, when taking the pipes into account, the efficiency looks better when calculating the efficiency with the calculated values at the inlet and outlet on the compressor. At low mass flows, the compressor efficiency looks better since the inlet and outlet air is heated and cooled before the measurement positions.

4 Conclusion

When investigating the efficiency change of the compressor, with the inlet and outlet pipes taken into account, it shows that the modeled pressure drop and temperature change occurring inside the pipes actually affect the measured compressor efficiency. At low mass flows, the efficiency that is measured looks worse than the calculated efficiency with pressure and temperature data closer to the compressor. The temperature change of the air inside the pipes affect the efficiency at low mass flow, the pressure drop due to surface roughness inside the pipes affect the efficiency at high mass flow. If the pipes are taken into account and the calculated values closer to the compressor are used to calculate the compressor efficiency, the efficiency is looking better. The inlet and outlet pipes are affecting the measured efficiency in a negative way, if the pipes are not accounted for when calculating the efficiency, the result might show a compressor efficiency that is inaccurate at both high and low mass flows.

4.1 Importance of taking the pipes into account

When looking at the results, it shows that when receiving compressor maps from dealers, the results could be different depending on how the measuring setup is configured. The most important thing from the thesis is the great impact from the pipe friction at higher mass flows, since the gasstand measurements are sometimes isolated to reduce the heat exchange with the environment, the temperature change of the air inside the pipes might be less important. The results also give a hint about what one could do to reduce the efficiency measurement error due to the pressure drop in the pipes. If the input and output pipes are made with a smooth inner surface, the measurement error due to the pressure drop in the pipes should decrease, compared to if the pipes where less smooth.

4.2 Future work

These things could be made with greater accuracy if more time had been available:

1. The bearing housing needs to be investigated further, the equations that are stated in the bearing housing chapter in appendix A could be extended to make the dynamic compressor housing temperature model dependent of the oil and water that lubricates and cools the turbocharger.

2. The heat conducted through the shaft has not been accounted for since the compressor housing temperature is assumed to be known. The conduction through the shaft is included with the total energy from the bearing housing in the calculations made in appendix A, but may be investigated separately.

3. The compressor may be divided into more parts, like nozzles and restrictions when the air is entering or leaving the compressor into pipes with different sizes. These connectors and such could change the pressure drops in the pipes even more.

4. The equation used to calculate the surface roughness might be changed, to get a more accurate value on the friction factor, since the pressure losses in the pipes are important when calculating the compressor efficiency.

Appendix

A

Bearing housing model

The bearing housing is a very significant part in the turbo. The tasks are to cool the turbocharger, reduce the amount of heat transfered to the compressor and also keep the shaft connecting the turbine and the compressor lubricated. The following chapter shows performed calculations, where the heat conducted from the turbine housing into the compressor housing is of interest.

A.1 Modeling heat transfer in bearing housing

There are four main heat transfers in the bearing housing, heat conducted from the turbine, energy transported from the bearing through water and oil and heat conducted to the compressor. There is also convection and radiation but since the gas stand measurements are winded in low conductive cord, these effects are neglected. The heat transfered from the turbine affects the compressor, by conduction from the bearing housing wall directly into the compressor housing. To keep the heat conducted into the compressor at a low level, the bearing housing is cooled with both water and oil. These four major heat transfers are described separately in the following chapter.

A.1.1 Heat conducted from the bearing housing to the compressor

To calculate the energy transfered from the bearing house to the compressor, the following formula is used:

$$\dot{Q}_{comp} = 15.8(T_{bearing} - T_{compressor}) \tag{A.1}$$

The value 15.8 W/K is given from a parallel master thesis, the author have managed to calculate the energy conducted from the bearing house to the compressor house. The master thesis is written by Mikael Bengtsson at Scania in Södertälje. Equation (A.1) will be used in the SIMULINK model, since the following section about bearing house calculations gave a resulting conducted heat flux that was too high (see A.2.4).

A.2 Bearing housing calculations

In this section it will be described how to calculate the amount of energy conducted from the turbine to the compressor. The following models will not be integrated in the SIMULINK model. The following models are described to get a knowledge of how the bearing housing could be modeled. This model gives a hint about the energy transport through the bearing housing until the cooling oil and cooling water are added in the last stage of the modeling. The models describing shaft friction and effective work on shaft, give good model fits, see figures A.1 and A.2.

A.2.1 Water cooling

The heat transfered from the bearing to the colling water is calculated by:

$$\dot{Q}_{water} = \dot{m}_{water} c_{p,water} (T_{wat,\ out} - T_{wat,\ in}) \tag{A.2}$$

The mass flow ($\dot{m}_{water}$) of the cooling water and the inlet and outlet temperatures (T_{in} and T_{out}) are needed to calculate the energy transported from the bearing house.

A.2.2 Oil lubrication and cooling

The oil in the bearing is used both as a lubricant and a coolant fluid. The energy transported by the oil is calculated in the same way as the energy transported by the water:

$$\dot{Q}_{oil} = \dot{m}_{oil} c_{p,oil} (T_{oil,\ out} - T_{oil,\ in}) \tag{A.3}$$

A.2.3 Heat conducted from the turbine housing to the bearing housing

The heat conducted from the turbine to the compressor is heavily reduced thanks to the cooling oil and water. The energy that is left after the cooling oil and water is conducted to the backplate, that is mounted on the backside of the compressor housing. The temperature on the backplate depends on both the turbine temperature and the compressor temperature. The energy transfered from the bearing housing to the compressor housing can be calculated according to:

$$\dot{Q}_{turbine} = K(T_{turbine} - T_{bearing}) \tag{A.4}$$

To find the heat flux $\dot{Q}_{turbine}$ a special test from the gasstand is used. The test is an adiabatic map where the turbine inlet temperature to the turbine is controlled to be as close as possible to the compressor outlet temperature. In that way, the heat transfered by conduction through the bearing housing is minimized. When making the following calculations, the heat transfer to the environment is neglected because the turbine and compressor are isolated with a low conductive cord during measurements. When the turbine and compressor are isolated, the radiation and convection to the surrounding air are minimized.

Shaft friction

Amount of work in turbine:

$$\dot{W}_t = \dot{m}_t c_{p,t}(T_{03} - T_{04}) \tag{A.5a}$$

Amount of work in compressor

$$\dot{W}_c = \dot{m}_c c_{p,c}(T_{02} - T_{01}) \tag{A.5b}$$

The friction in the shaft is calculated with:

$$\dot{W}_{f,shaft} = \dot{W}_t - \dot{W}_c \tag{A.5c}$$

The function for shaft friction is stated as:

$$\dot{W}_{f,shaft\ model} = f_f(N_{tc}) \tag{A.5d}$$

A function is created to describe the shaft friction in the bearing housing. The friction is dependent of mainly the shaft speed. Mechanical friction is linear to the shaft speed, but since there are coolant fluids surrounding the shaft, a quadratic relation to shaft speed is expected. Figure A.1 shows the calculated shaft friction. When studying the calculated power loss due to friction (in figure A.1), it is noticeable that the model fit isn't that good. There is probably some parameter missing to describe the shaft friction in a better way. Its reasonable to believe that the pressure drop over the turbine (Π_t) or the mass flow ($\dot{m}_t$) through the turbine that is missing. The measured data is spread over a smaller interval at each shaft speed (seen in figure A.1), still this model is used.

Effective shaft work

The function for effective shaft work (the shaft work when heat transfer is neglected) is also calculated using the adiabatic map. The following equation is used:

$$\dot{W}_{effective} = \dot{W}_t - \dot{W}_{f,shaft} \tag{A.6a}$$

The model for $\dot{W}_{shaft}$ is given by:

$$\dot{W}_{effective,model} = f_{eff}(\Pi_t) \tag{A.6b}$$

A model is created to describe $\dot{W}_{effective}$. The model depends on relative pressure difference over the turbine, Π_t.

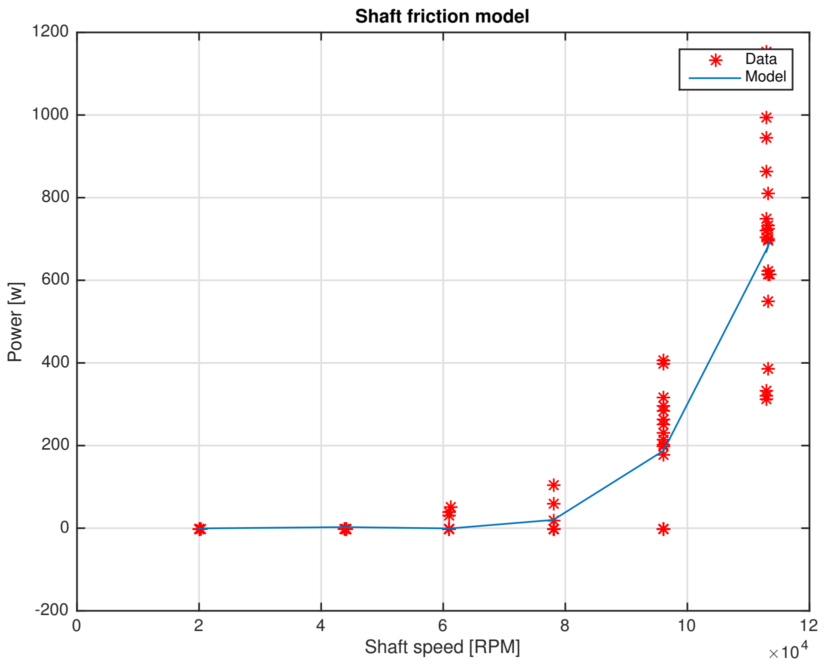

Figure A.1: *Power loss due to shaft friction plotted against the shaft speed. The model is equation (A.5d), the data is calculated according to equation (A.5c). Clearly there is some trend in the data that isn't included in the model.*

Heat conducted from turbine housing to bearing housing

With the friction power and the effective work modeled, the heat conducted from the turbine housing into the bearing housing can be calculated. Both models are parametrized to fit the adiabatic map, so the models does not model heat transfer. A hot map (gasstand measurement with turbine inlet temperature $T_{03} = 750\ {}^oC$) is used to include the affects of heat transfer. If subtracting equation (A.5d) and (A.6b) from equation (A.7a), this should be equal to the heat transfered into the bearing housing.

$$\dot{W}_t = \dot{m}_t c_{p,t}(T_{03} - T_{04}) \tag{A.7a}$$

$$\dot{Q}_{bearing} = \dot{W}_t - \dot{W}_{f,shaft} - \dot{W}_{effective} \implies \tag{A.7b}$$

$$\dot{Q}_{bearing} = \dot{W}_t - f_{eff}(\Pi_t) - f_f(N_{tc}) \tag{A.7c}$$

The heat conducted into the bearing housing is shown in figure A.3. The model gives a hint about the heat conducted from the turbine to the bearing house, but is not very accurate, this is probably due to the heat losses from the turbine to the environment when adapting the models (A.5d) and (A.6b) to the adiabatic

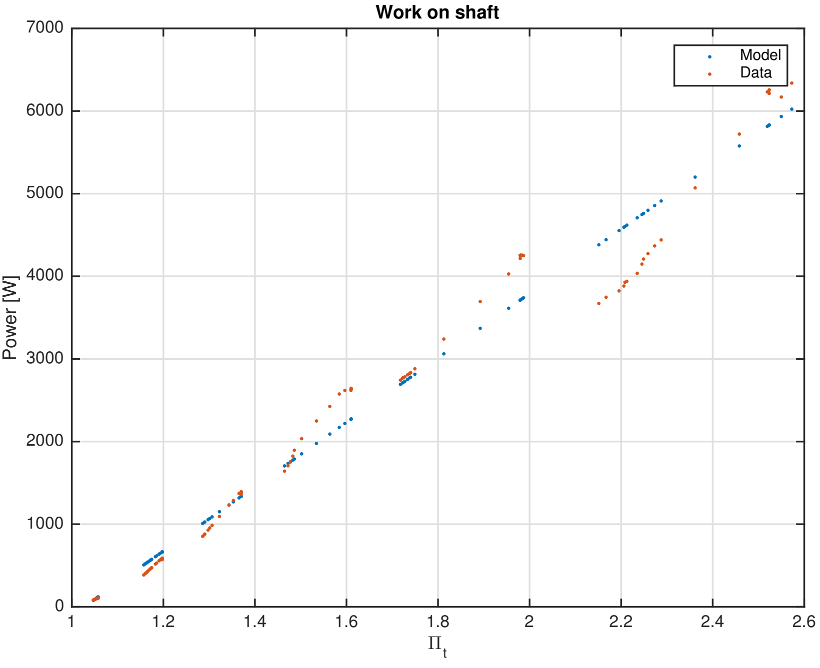

Figure A.2: *Effective power plotted against Π_t calculated with equation (A.6b) and compared to the measurements. The model agrees acceptably with the measured data.*

map. If the heat losses in the turbine had been accounted for, the model result may have been better.

Heat conducted from bearing housing to compressor housing

The amount of heat conducted into the compressor housing form the bearing housing should be given if subtracting the effects of the cooling water and oil in equation (A.2) and (A.3) from $\dot{Q}_{bearing}$ in equation (A.7b), as:

$$\dot{Q}_{comp} = \dot{Q}_{bearing} - \dot{Q}_{wat} - \dot{Q}_{oil} \tag{A.8}$$

A.2.4 Rejection of bearing housing model

The developed bearing housing heat transfer models are rejected since the final energy Q_{comp} in equation (A.8) does not seem to be accurate (see the compared models in figure A.4). There is clearly some trend that is not modeled. The missing model could be the heat loss that should take place at the turbine. In figure A.4, the developed models are denoted $Q\ calculated$, these are compared with the heat transfered according to equation (A.1). Q_{MB} seems a lot more reasonable,

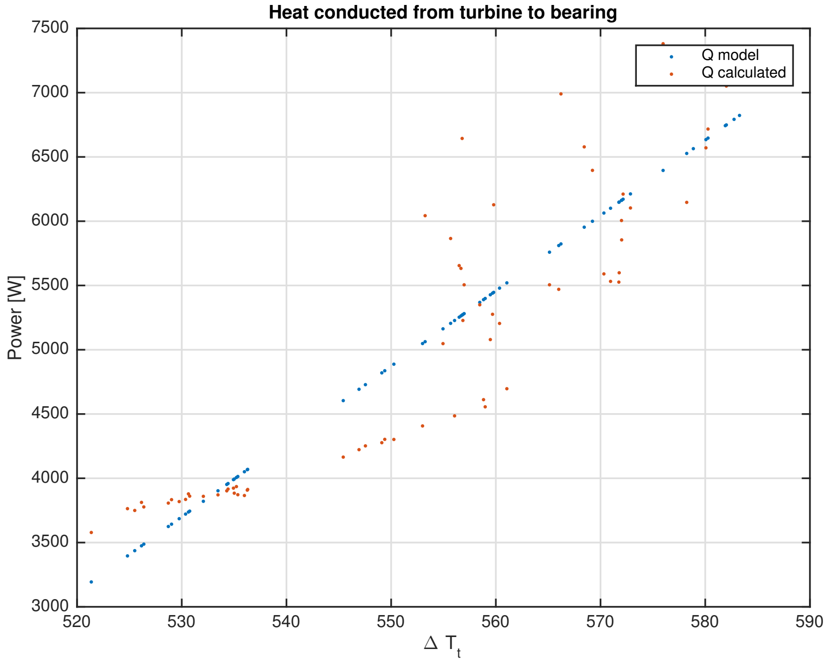

Figure A.3: *Heat conducted from the turbine to the bearing house. The model is calculated with equation (A.7c), the reference data is calculated according to equation (A.7b)*

since it is linear compared to the temperature difference (ΔT) between the two parts. The amount of heat transfered should not increase when the temperature difference (ΔT) decreases.

A.2.5 Bearing housing temperature

To calculate the bearing house temperature, the bearing housing is given a temperature state (see equation (A.9)). The heat transfer into and out from the bearing housing is modeled and the temperature is left as a state to be solved by MATLAB & SIMULINK. The mass of the bearing house is estimated to be 1 kg and the material is assumed to be iron with thermal storage capacity $c_p = 420\ J/kgK$, according to p.79, Storck et al. [2010]. The dynamic temperature model isn't getting close to any of the measured temperatures. A measurement of the temperature from the gasstand data is used instead.

$$\frac{dT_{bearing}}{dt} m_{bearing} c_{p,,bearing} = \dot{Q}_{internal} - \dot{Q}_{external} \tag{A.9}$$

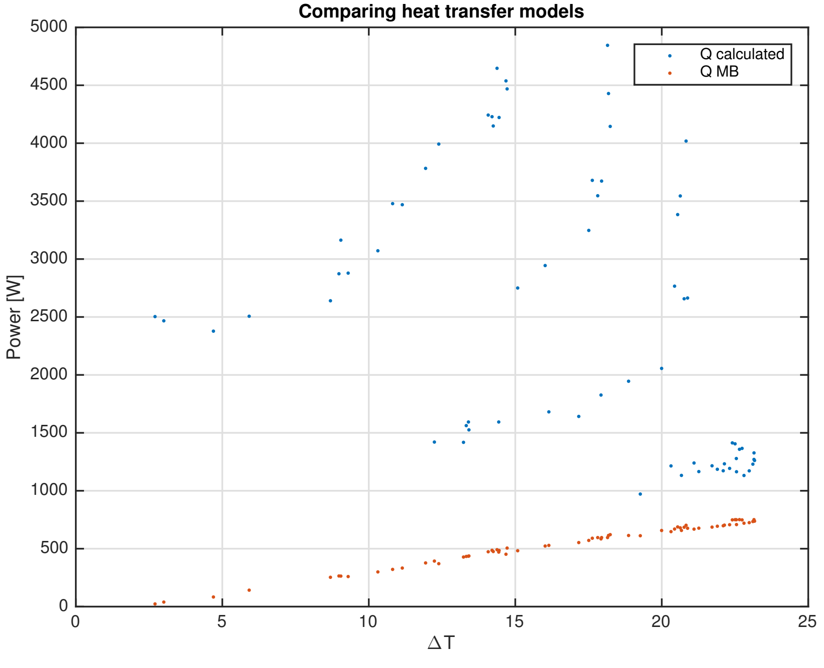

Figure A.4: *Modeled value of $\dot{Q}_{comp}$ (denoted $Q\ model$) compared to Mikael Bengtssons model $\dot{Q}_{MB}$ (equation (A.1)). ΔT is the temperature difference between the compressor housing and the bearing housing (temperature sensor on backplate).*

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