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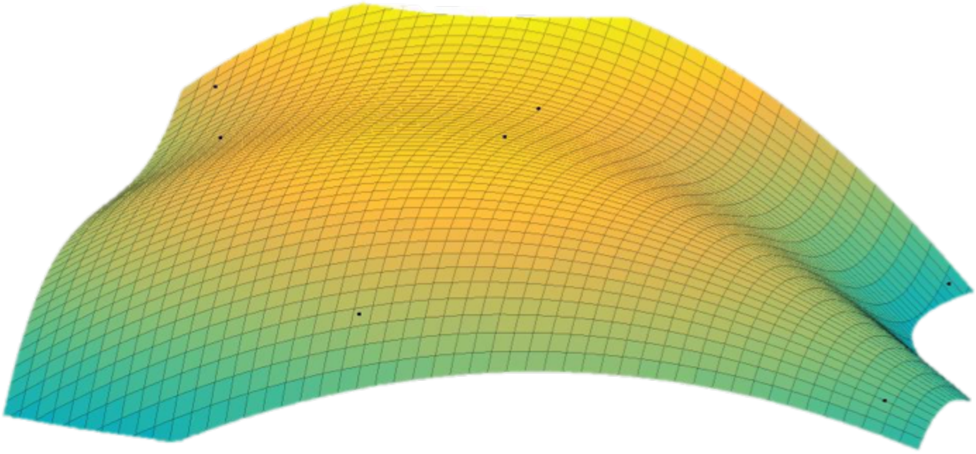

Dynamic Temperature Model of an Automatic Transmission

Master's thesis in Automotive Engineering and Mechanical Engineering

Oscar Berglund
Yao Zhang

Department of Mechanics and Maritime Sciences
CHALMERS UNIVERSITY OF TECHNOLOGY
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MASTER'S THESIS 2019:04

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Oscar Berglund
Yao Zhang

Supervisor: Raoul Rinaldo, Volvo Cars
Per-Arne Malm, Volvo Cars
Examiner: Lucien Koopmans, Department of Mechanics and Maritime Sciences, Chalmers University of Technology
Sharon Kao-Walter, Department of Mechanical Engineering, Blekinge Institute of Technology
Wureguli Reheman, Department of Mechanical Engineering, Blekinge Institute of Technology

Master's Thesis 2019:04
Department of Mechanics and Maritime Sciences
Division Combustion and Propulsion Systems
Chalmers University of Technology
SE-412 96 Gothenburg
Telephone +46 31 772 1000

Cover: Heat transfer coefficient surface constructed in Matlab, showing the heat transfer coefficient for different temperatures of oil and engine speeds.

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Abstract

This report presents the development of a dynamic temperature model for an automatic transmission in a Volvo Cars passenger vehicle. The model should simulate the oil to cooler temperature and flow from the transmission. A mathematical approach to use lumped masses for different parts of the transmission was used. These lumped masses were the oil, moving parts and house. To tune the response of the lumped masses and heat transfer coefficients, temperature measurements were used. Some existing measurements at Volvo Cars were tested, but these measurements were not accurate enough. New improved measurements on a vehicle in a chassis dyno were then performed. A decision to do measurements on all eight gears were made since the power losses were different on every gear. However the torque converter was open on the first gear, and could not be closed, which led to the decision to only test the remaining gears since the first gear will only be used during short times during most driving scenarios.
To verify the model, simple drive cycles were performed with temperature measurement in the same chassis dyno and on the same vehicle.
The verification on the model shows that the model can simulate the behaviour of a transmission with an error of 2.5 °C during normal behaviour and 6.5 °C for a few minutes, when a sudden change in the temperature from the cooler have a large transient increase. Because of this, the model is considered to be fairly accurate. However, in order to make the model compatible with Volvo Cars existing simulation software, Vsim, a "cooler model" has to be created.

Keywords: Model, Dynamic, Transmission, Simulation, Gearbox, Thesis, Temperature, Lumped mass, Drive cycle, Oil.

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Oscar Berglund, Yao Zhang, Gothenburg, January 2019

Nomenclature

α	Distribution constant describing how much, in percent, of the power losses are going into the oil and rotating parts respectively [-]
$\dot{m}_{oil}$	Mass flow of the oil [$\frac{kg}{s}$]
$\dot{Q}_h$	Heat from/to the housing [W]
$\dot{Q}_m$	Heat from/to the moving parts [W]
$\dot{Q}_{amb}$	Heat from/to the ambient air [W]
ϵ	Emissivity [-]
μ	Dynamic viscosity [$\frac{Ns}{m^2}$]
ν	Kinematic viscosity [$\frac{m^2}{s}$]
σ	The Stefan Boltzmann constant, $5.67 * 10^{-8}$ [$\frac{W}{m^2 * K^4}$]
A	Area [m^2]
c	Specific heat [$\frac{J}{kg*K}$]
c_h	Specific heat of the housing [$\frac{J}{kg*K}$]
c_m	Specific heat of the moving parts [$\frac{J}{kg*K}$]
c_{oil}	Specific heat of the oil [$\frac{J}{kg*K}$]
h	Convective heat transfer coefficient [$\frac{W}{m^2 K}$]
h_h	Convective heat transfer coefficient between the oil in the transmission and the internal area of the transmission [$\frac{W}{m^2 K}$]
h_m	Convective heat transfer coefficient between the oil in the transmission and the moving parts in the transmission [$\frac{W}{m^2 K}$]
h_{amb}	Convective heat transfer coefficient between the external area of the transmission and the ambient air [$\frac{W}{m^2 K}$]
k	Thermal conductivity, expresses the heat-transfer rate in conduction [$\frac{W}{mK}$]
L	Length [m]
m_h	Mass of the housing [kg]
m_m	Mass of the moving parts [kg]
m_{oil}	Mass of the oil [kg]
P_{loss}	Power loss [W]
Pr	Prandtl number [-]
q	Emitted radiation [W]
rpm	Revolutions Per Minute, rotational speed
T_h	Temperature of the house [°C]
T_m	Temperature of the moving parts [°C]
T_{amb}	Temperature of the ambient air [°C]
$T_{oil,in}$	Temperature of the oil from the cooler [°C]
$T_{oil,out}$	Temperature of the oil to the cooler [°C]

Contents

List of Figures

1 Introduction

To set up the project background, aim, limitations and specification of issue under investigation were done. A literature review was also made to see what has previously been done in the field.

1.1 Background

The department of Propulsion Transmission Systems, which is a part of Volvo Cars Research and Development organization, are dedicated to keep their powertrains competitive. Development times need to be shorter and one way of doing this is to do more of the work in the early phases with CAE tools instead of testing physical parts. At Volvo Cars the CAE team are refining their dynamic coolant temperature model and need better input data representing the transmission since too many simplifications have been made in the existing model. The existing model does only model the transmission warm up behaviour with the heat coming from the power losses generated inside the transmission.
In reality, during start-up when the transmission is cold, the engine and cooling liquid are heated up faster than the transmission. This enables the cooling liquid to help in heating up the transmission oil to working temperature. The CAE team would like this to be added and modelled to better capture the behavior of the transmission during a driving cycle.
Today the cooling department uses look-up tables for the output heat of the oil in steady state, and the existing model does only give the oil to cooler temperature as output. The cooling department would like to have a dynamic model which gives output of the transmission oil to cooler as temperature and flow during a driving cycle.

1.2 Aim

The aim of the project is to build a dynamic temperature model which can run any drive cycle as input and predict the temperature and flow of the oil to cooler from a Volvo Cars transmission. In figure 1, the available inputs and the desired outputs are shown (output "Ambient heat" is a consequence of the input ambient temperature and will not be used for anything). The model should also be able to handle the start up behaviour of the more modern transmission where the oil from cooler will assist in the heat up of the transmission during the first minutes of a cold start. A report will be produced which will be presented in late January 2019.

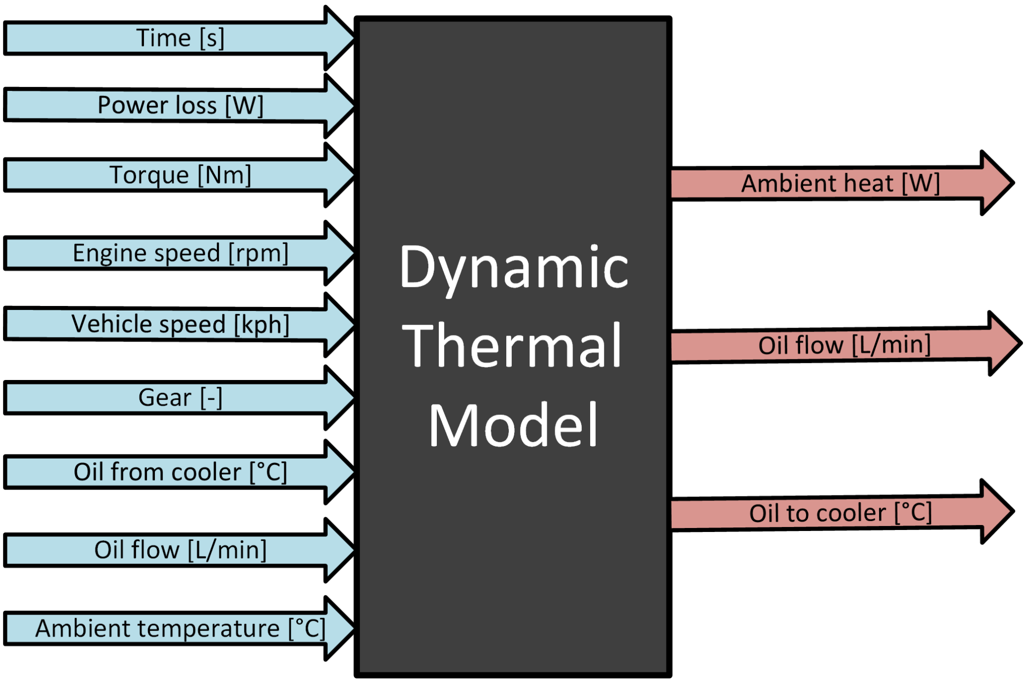

Figure 1: Black box

1.3 Limitations

The heat up process of a transmission is complex, and to accurately know how fast the oil in the transmission is heated up or cooled down requires a 3D Computational Fluid Dynamics (CFD) investigation. Volvo Cars does not have all the necessary models and coefficients of the transmission that is required for a 3D CFD simulation. It would also be too time demanding for this project and the model would also require too much computational power to co-simulate in Volvo Cars simulation software Vsim (an in-house software based on Matlab/Simulink). Instead an approach where the transmission is seen as a black box is used. This means that the physical structure of the transmission will not be simulated but only the thermal behaviour of the oil in the transmission. This also means that the resulting black box will not be physically correct but more like a curve fitting where the black box model will try to be made to act as the real system.
The model will ignore the conduction within the metal parts and oil. Instead an assumption that all different parts will have a uniform temperature distribution will be made, meaning that the same temperature will be assumed on all places in the different parts. The conduction between the engine and the transmission will also be ignored since it is considered to have a small impact on the oil temperature in the transmission.
To further simplify the model, radiation inside and from the transmission (between the outer-surface and the air) will be ignored as a starting approach but may be included if there is enough time before the end of the project. The torque converter will be considered to always be closed and not open. This consideration will be done since the torque converter will only be in open mode for a relatively short period of time in most real world driving scenarios.
A logic figure of the transmission with the cooler can be seen in figure 2. Objects inside

the green line indicates what will be included of the physical parts in the figure. This means that the cooler and cooling liquid will not be regarded in this project.

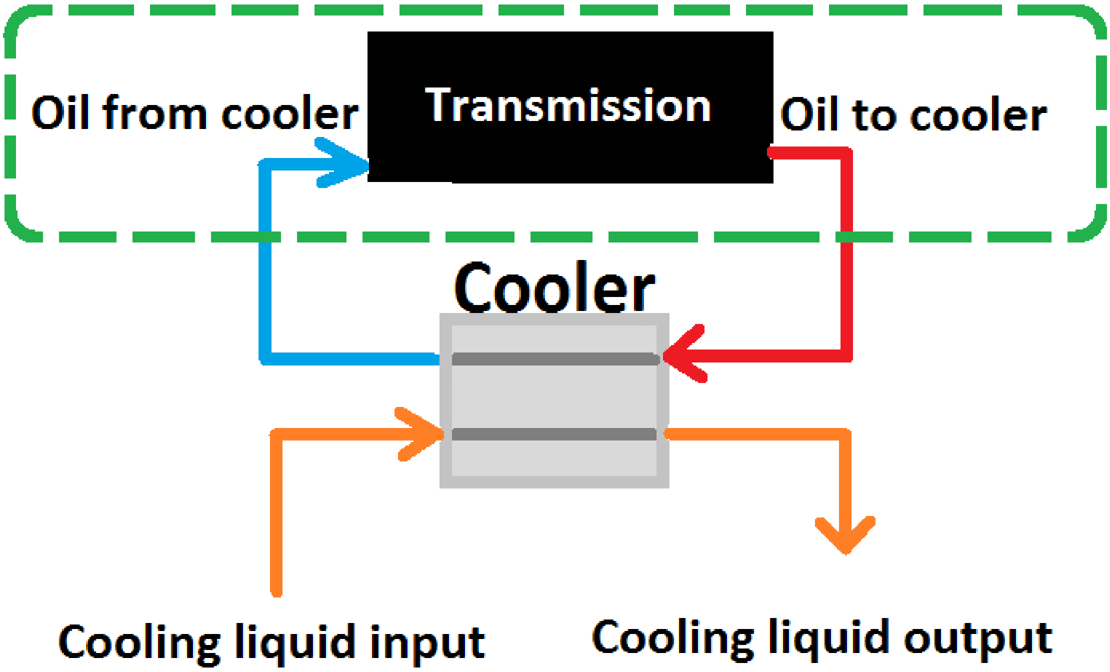

Figure 2: Figure of transmission and cooler

1.4 Specification of issue under investigation

The project aims to answer the following questions:

- Does the model predict a driving scenario similar to real world measurements?
- What ways can be used to model the system?

1.5 Literature review

The thermal behaviour of a gearbox has been studied by other projects. Most reports focus on determining the powerlosses in a specific gearbox. But the total powerloss is already known in this project. And since the full data of the transmission is unknown a more detailed analysis of where the powerlosses are occurring is not possible. In Häggström [1] the powerlosses and temperatures for some different parts are analyzed, in a Scania truck transmission, using the modelling software Amesim by specifying the geometries of the transmission parts. He states that the convective heat transfer coefficient is difficult to calculate for rotating parts (not immersed in oil) due to the unknown flow case. Häggström then uses an empirical formula (equation 59 in Häggström [1]) which is dependent on the rotational speed of the active gear. Oil cooling/warming is modelled using a linear assumption of the warm up behaviour.

In literature [2] an unknown author describes a transmission thermal model, of a single worm gear, modelled as lumped masses (which mean that the mass is assumed to be a single rigid object) where the gearbox is split into three lumped masses; the oil, gear teeth and the casing. With heat transfer equations a mathematical model is created to determine the temperatures of the different lumped masses. Heat transfer coefficients are calculated for the convective heat transfer between the oil and the casing and also between the casing and the ambient air. No oil cooling is used, but the effect of forced convection through air flowing around the casing has a cooling effect up to around 20 m/s of air speed. An even higher air speed is concluded to not have a more effective cooling.

Prakash del Valle [3] calculates the thermal behaviour on a FZG test gearbox without cooling. He uses a lumped body approach and sets up the different lumped bodies in a thermal node network, solves differential equations in Matlab, to see how the different parts interact with each other. He concludes that to improve the model a more advanced approach with a finite element approach (FEM) could be used to improve the simulation values to be more close to experimental values.

The dynamic temperature project (this project) is inspired by the lumped mass approach and to divide the transmission into oil, gear teeth (in this project moving parts) and house used in literature [2]. Prakash del Valle [3] and literature [2] gave support to that a lumped mass approach have been tested before. The project is also inspired by Häggström [1] to make the heat transfer coefficient dependent on engine speed.

2

Theory

The theory chapter introduces some elements which will be used later in the report. Key parts of the transmission are introduced and different kinds of heat transfer and thermodynamical properties. Finally a brief introduction to the existing model at Volvo Cars is done as well as an explanation to how steady state for temperatures is normally defined at Volvo Cars.

2.1 Automatic transmission basics

There are different types of automatic transmissions on the market today. The traditional automatic transmission with planetary gears and a torque converter is used in this project. It consists of two main parts, the torque converter and the gearbox which work together to change the rotational speed and torque from the engine to the driving wheels. In this report a transmission is defined as a gearbox with a torque converter and pump.

Normally a transmission can only be cooled with the oil from the cooler. But in this project the transmission have the ability to also be heated up with warm oil from the cooler to faster achieve working temperature.

2.1.1 Torque converter

Between the engine and the gearbox there is a torque converter. A torque converter made for the transmission can be seen in figure 3 and 4. This is a fluid coupling which generate a torque amplification at low vehicle speeds. It also allow a slip between the gearbox and the engine and dampens vibrations. It can simplified be described as two opposing fans (impeller and turbine) with a fixed stator in between. Between the components, oil will flow to transfer the rotational motion from the impeller to the turbine. The impeller is connected to the drive shaft from the engine and the turbine is connected to the input shaft of the transmission. The torque converter is used at low speeds, like starts, since the torque amplifications only works when there is a rotational speed difference between the impeller and the turbine. When the starting phase is over, for a vehicle, the impeller and turbine will rotate with a smaller difference in rotational speed compared to each other. In order to reduce the heat losses, created by the difference in rotational speed, the impeller and turbine will then be locked together with a clutch to have the same rotational speed.

Figure 3: Torque converter back

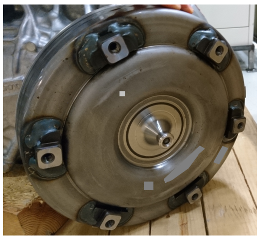

Figure 4: Torque converter front

2.1.2 Pump

There is a mechanically driven pump in the transmission. Its function is to distribute oil to gears, bearings, shafts etc. for lubrication and cooling. It also creates oil pressure used to engage and disengage clutches and brakes to change gears and to lock the torque converter. Some of the oil from the pump goes to the cooler which cools down the oil before it is recirculated back into the transmission again. It is this oil to cooler which is the output from the project described in section 1.2.

2.1.3 Gearbox

The gearbox can be seen in figure 5. The gearbox in this project has eight front gears and one reverse gear, but only the front gears will be evaluated since the vehicle will mostly be moving forward. By engaging different clutches and brakes different gear ratios will be achieved by the planetary gears amplifying or decreasing the output torque and rotational speed. However there are power losses in the gearbox, creating heat, which has to be cooled by a cooler in order not to overheat the gearbox.
By measuring the transmission housing in a CAD-model the outside surface area was estimated with or without a plastic cover and cooler to reduce the surface area.

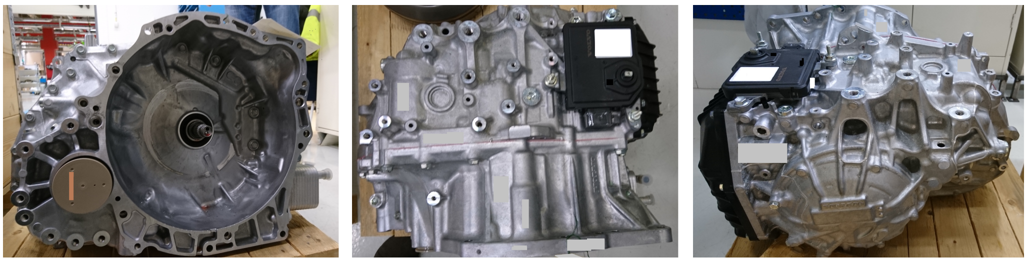

Figure 5: The gearbox from different sides. The torque converter is removed.

2.1.4 Cooler

On the outside of the transmission there is a cooler which can be seen in figure 6 and figure 2. It cools the transmission oil with cooling liquid. In the future generation gearboxes the oil will not only have the ability to be cooled by the cooler but also be heated up by the cooler during a cold start. This means the the oil from cooler can have a higher temperature than the oil to cooler during a cold start.

A small percentage of the oil is in the pipes, in and from the cooler. However this is a small portion of the total oil and according to technical expert Per-Arne Malm at Volvo Cars, this part can be neglected so that all oil is seen to be within the transmission when doing calculations.

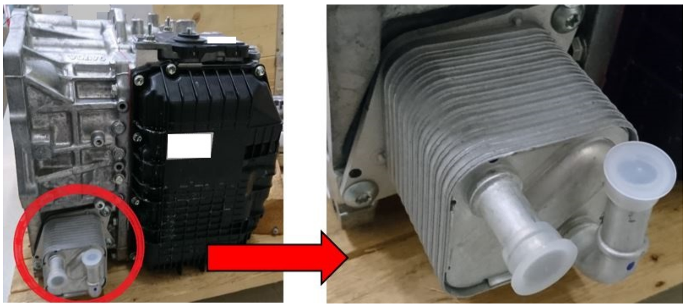

Figure 6: Cooler

2.2 Forms of heat transfer

Heat can be transferred through conduction, convection and radiation. These different types are briefly described bellow.

2.2.1 Conduction

Conduction is heat transfer through a solid or non-moving medium by molecular diffusion created by a temperature gradient. Solutions for heat transfer through conduction is based on the empirical expression, equation 1, known as Fourier´s law according to Kuehn [4].

$$\dot{Q} = -kA\frac{dt}{dx} \tag{1}$$

Where $\dot{Q}$ is heat [W] transferred through the medium. k is the thermal conductivity which expresses the heat transfer rate [$\frac{W}{mK}$]. A is the area [m^2] and $\frac{dt}{dx}$ is the change in time over the change in position.

2.2.2 Convection

Convection is the heat transfer by mixing a fluid or gas, Ekroth [5]. There are two kinds of convection: free and forced. The free convection is caused by temperature differences within the fluid which will create density differences and motion of the fluid due to gravitational forces. If the flow of the fluid is created by an external source, like a fan, pump or rotating gear wheels which is the case for the transmission when moving the oil, it is called forced convection. If the convection is between a solid surface and a fluid it can be described by equation 2 from Kuehn [4].

$$\dot{Q} = hA(T_1 - T_2) \tag{2}$$

Where h is the heat transfer coefficient $[\frac{W}{m^2K}]$ and T_1 and T_2 are the temperatures of the two mediums.
Equation 2 can be rewritten so the heat transfer coefficient and the area are a value together as seen in figure 3. This fictive $\tilde{h}$ (heat transfer coefficient) will be used alot in this report.

$$\dot{Q} = \tilde{h}(T_1 - T_2) \tag{3}$$

The equation looks simple but finding the heat transfer coefficient h or $\tilde{h}$ can be very challenging. This coefficient is more explained in section 2.3.

2.2.3 Radiation

Radiation is the emission of waves or particles through another material. The radiation emitted from a material can be calculated with Stefan Boltzmann's equation 4.

$$q = \epsilon * \sigma * A * (T_1^4 - T_2^4) \tag{4}$$

Where ϵ is the the emissivity which is the materials effectiveness to emit radiative energy ranging from 0 to 1. σ is the Stefan Boltzmann constant which is $5.67 * 10^{-8} \frac{W}{m^2 * K^4}$. q is the emitted power [W].

2.3 Thermodynamical properties

Important thermodynamic coefficients are listed bellow. The coefficients are described according to Kuehn [4].

Heat transfer coefficient

If there is a temperature difference between a connecting fluid and a solid there will be a heat transfer between these two materials. The heat transfer coefficient, h, determine how much heat [W] is transferred over $1m^2$ for 1 Kelvin of temperature difference between the materials. The heat transfer coefficient is however complex to compute since it is not constant but varies with parameters like geometry and fluid velocity. To accurately know the heat transfer coefficient, a 3D CFD simulation has to be done since the heat transfer coefficient varies at different locations. For simpler geometries like flat plates, cylinders or spheres the heat transfer coefficient can be approximated to a single value.
A simplified case with the heat transfer coefficient for forced convection over a flat plate can be seen in equation 5.

$$h = \frac{k * 0.036 Pr^{0.43}((\frac{VL}{\nu})^{0.8} - 9200)}{L} (\frac{\mu_\infty}{\mu_w})^{0.25} \tag{5}$$

Where Pr, k and ν are properties of the fluid. V is the velocity of the fluid. L is the length of the plate and μ_∞ and μ_w are viscosity of the fluid. However the calculated heat transfer coefficient is just an approximation and there are still uncertainties about this value.

Specific heat

The specific heat, c, specifies how much energy [J] is needed to raise the temperature of a material with 1 Kelvin if the mass is 1 kg. For gases the specific heat can be specified as c_p (the specific heat if the pressure is held constant) and c_v (the specific heat if the volume is held constant). For solid materials and almost incompressible liquids, like oil, these specific heat constants can be seen as equal $c = c_v = c_p$.

2.4 Existing model in Vsim

Since the transmission is not made by Volvo Cars but bought from an external supplier, only limited data of the transmission is available to Volvo Cars. Because of this the transmission is seen as a hardware black box from Volvo Cars' point of view.

The existing model running in Vsim today is a software black box model seen as a single lumped mass. It uses tuned parameters of the specific heat coefficient (c) times a mass for heat up of the lumped mass and the heat transfer coefficient (h) times an area for air cooling to the environment. These two parameters and the mass of the lumped body are not real world values but only chosen to make the system oil temperature respond similar to the real oil temperature when running the driving cycle NEDC (New European Driving Cycle).

Since the model only includes air cooling and not oil cooling the system will overheat if the model run time would be longer than a NEDC cycle. The existing model is therefore not generic since it is only adapted for the specific driving cycle NEDC.

Three heat producing units are heat inputs to the model, namely the oil pump, torque converter and the gearbox. However the oil pump power loss is not included in the existing model, which has led to that the parameters of the model have been tuned to match the behaviour of the transmission without this heat. Not using the oil pump heat is a known mistake that was made when designing the model in the first place. This is due to that the inputs were historically different from what they are today and the model has not been tuned to the more modern inputs.

2.5 Steady state definition

Since many systems fluctuate there can be a need for a definition of when a system is regarded to be in steady state. At Volvo Cars a normal definition of steady state with temperatures, is when the measured systems temperature change a maximum of 1 °C over 5 minutes.

3
Method

3.1 Modelling concepts

Different model concepts were investigated in order to find the most suitable for this project.

3.1.1 CFD

A 3D CFD model of the system is the most accurate representation of the real transmission since the whole internal physical structure of the transmission is simulated and the heat transfer coefficients (h) are calculated for every element at every time step. However this requires that all physical data and CAD models of the transmission are known and since Volvo Cars has not developed the existing transmission, but have bought it from an external supplier, the physical data of the transmission is not available.
Even if the physical data and CAD models would have been available the model would have been slow since it would require a lot of computational power to solve all fluid motions inside the transmission even for just one stationary point. It is also very difficult to model moving parts like a planetary gearbox. Therefore the CFD approach was excluded at an early stage. This was also stated in the limitations in section 1.3.

3.1.2 Look-up tables

By measuring the temperature of the oil to cooler, at many different steady state points, look-up tables of the dependencies of the temperature and other parameters like engine speed [rpm], input torque to the transmission [Nm] and selected gear [-] can be created. Interpolation can then be used in the look-up table to achieve data for values in between the measured values.
Since the measurements are done in steady state, no transient behaviour of the transmission can be captured. This means that the model will not behave as the real transmission when the power levels are changing or if the vehicle is stopped with the engine running. Because the vehicle will continue to receive or give heat internally and to the ambient air.

3.1.3 Heat into oil and moving parts

When referring the transmission as a black box with three different lumped masses, which transfer heat to each other, a real world behaviour of the model could be achieved without knowing the internal physical structure of the transmission. Prakash Del Valle [3] and literature [2] gave support to that a lumped mass approach have been tested before.

Since the major materials in the transmission are Aluminum (the housing), Iron (the rotating parts) and lubrication oil these three materials can be seen as lumped masses which interact with each other.

In reality most of the heat is generated in the contact point between gear teeth and oil. The temperatures are locally much higher compared to the average temperature of the oil and rotating parts. The power losses are both transferred to the oil as well as to the rotating parts. By introducing a factor α (distribution constant) describing how much of the power losses are going into the rotating parts and oil, a behaviour more similar to reality can be achieved.

The power losses (heat) in the transmission go into the oil and the moving parts which transfer heat to the housing. The advantage of this approach is that the model can, if tuned correctly be used to describe a transient behaviour. The lumped masses can use the real masses and specific heat of the transmission, enabling the same model to be used for different transmissions by only changing the masses in the model. The disadvantage of this approach is that the model is a simplification of the reality and that the unknown factor α has to be calculated.

Volvo Cars does not have the ability to measure the temperature of the moving parts but according to literature [2] the temperature of the gear teeth are higher than the temperature of the oil.

3.1.4 Heat into rotating parts

Changing the "Heat into oil and moving parts" approach by removing the distribution constant and allowing all the power losses go into the rotating parts will simplify the model making it easier to solve. In reality the temperature distribution of the rotating parts are uneven with a high temperature at the gear contact and lower temperature at parts further away from the active gear.

3.1.5 Heat into oil

Changing the "Heat into oil and moving parts"-approach by removing the distribution constant and letting all the power losses go into the oil will simplify the model making it easier to solve. This has the advantage, compared to the "Heat into rotating parts" approach, that the oil temperature responds faster to a change in power loss. The disadvantage is that this approach is further away from the reality compared to the "Heat into rotating parts" approach. The "Heat into oil" approach was the approach chosen since it was seen to be relatively easy mathematically and since it was believed to be able to capture the transient behaviour. Since the thermal lumped body approach had not been tested at the transmission department before it was decided to use the simplest approach to see if it even was possible to use a lumped body approach with satisfactory results.

3.2 Heat transfer

3.2.1 Radiation

Both the internal radiation inside the transmission and the radiation from the outside of the transmission housing to the ambient air are neglected. The internal radiation is neglected due to the small temperature differences between the oil, moving parts and the housing. The temperature differences between the different internal parts are in the region of 15 K between the oil and the housing. With a steady state temperature of the oil of 90°C and 75°C of the housing the radiation can be calculated to $118\frac{W}{m^2}$ with equation 4 and $\epsilon = 0.77$ which is the highest value for aluminum according to literature [6], and hence it is probably lower in reality for the oil coated transmission walls.
The area of the oil in the oil sump and the internal area of the housing are both smaller than $1m^2$, since the outside area of the transmission seen in figure 5. This also will generate a radiation value lower than 118W which generally is less than one tenth of the powerlosses.
However the effect of the internal radiation will somewhat be included in the calculations by its' effect on the internal heat transfer coefficients (heat transfer coefficient for housing and moving parts) even though the actual radiation values are unknown. The external radiation from the transmission housing is neglected due to the uncertainties in view factors and which temperature the warm surrounding objects in the engine bay has. However the effect of the external radiation is assumed to be captured by the ambient heat transfer coefficient.

3.2.2 Conduction

Convection was neglected as heat transfer and all the three different parts (oil, moving parts and housing) in the transmission was considered to have uniform temperature distribution. Convection was neglected since the internal layout of the transmission is unknown and only the total power losses are known, without knowing from what specific parts different amount of power losses are coming from.

3.2.3 Convection

The interaction between the three different parts (oil, moving parts and housing) will be the oil for the internal parts (oil, oil on the moving parts and oil on the internal area of the housing) and the surrounding air for the external area of the housing. Convection was the only heat transfer mode chosen for the model and the ambition was that it would be enough to capture the oil temperature behaviour close enough to the reality.

3.3 Mathematical model

Three different differential equations (equation 6 - 8) were set up. One for each part of the black box. All the equations are set up as: if the part receives heat then the sign is positive and if the part loses heat then the sign is negative.
For the moving parts:

$$m_m c_m \frac{dT_m}{dt} = \dot{Q}_m \tag{6}$$

Where m_m is the mass of the moving parts [kg], c_m is the specific heat of the moving parts [$\frac{J}{kg*K}$], $\frac{dT_m}{dt}$ is the change in temperature of the moving part over time and $\dot{Q}_m$ is the heat to/from the moving part [W].
For the housing:

$$m_h c_h \frac{dT_h}{dt} = \dot{Q}_h - \dot{Q}_{amb} \tag{7}$$

Where m_h is the mass of the house [kg], c_h is the specific heat of the house [$\frac{J}{kg*K}$], $\frac{dT_h}{dt}$ is the change in temperature of the house over time, $\dot{Q}_h$ is the heat to/from the house [W] and $\dot{Q}_{amb}$ is the heat to/from the ambient air [W]
For the oil:

$$m_{oil} c_{oil} \frac{dT_{oil,output}}{dt} = P_{loss} - \dot{Q}_m - \dot{Q}_h - \dot{m}_{oil} c_{oil} (T_{oil,out} - T_{oil,in}) \tag{8}$$

Where m_{oil} is the mass of the oil [kg], c_{oil} is the specific heat of the oil [$\frac{J}{kg*K}$], $\frac{dT_{oil,output}}{dt}$ is the change in temperature of the oil over time, P_{loss} is the power loss [W], $\dot{m}_{oil}$ is the mass flow of the oil [$\frac{kg}{s}$], $T_{oil,out}$ is the temperature of the oil to the cooler [°C] and $T_{oil,in}$ is the temperature of the oil from the cooler [°C].
The logical layout of the mathematical model can be seen in figure 7. The power losses are here generated in the oil and the heat is going to the moving parts and the housing as well as being mixed with the oil from cooler.

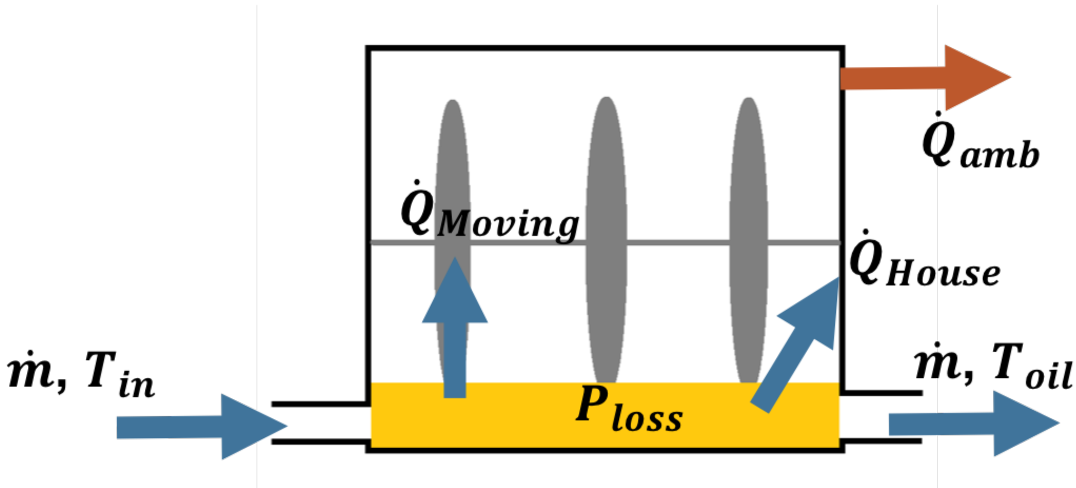

Figure 7: Mathematical model

$\dot{Q}_m$, $\dot{Q}_h$ and $\dot{Q}_{amb}$ in equation 6 - 8 can be expanded as seen in equation 9 - 11. This shows $\tilde{h}_m$, $\tilde{h}_h$ and $\tilde{h}_{amb}$. The "tilde" symbol above the h-values (heat transfer coefficients) indicates that these values are fictive with the area included in the heat transfer coefficients. The normal way of writing a heat transfer is the heat transfer coefficient (h) times an area ($h * A$) as in equation 2. But since the internal areas are unknown and it is difficult to know how much of the outside areas are used for heat transfer, all of the heat transfer coefficients have the unknown areas included in the heat transfer coefficients, like in equation 3.

$$m_m c_m \frac{dT_m}{dt} = \tilde{h}_m(T_{oil,out} - T_m) \tag{9}$$

$$m_h c_h \frac{dT_h}{dt} = \tilde{h}_h(T_{oil,out} - T_h) - \tilde{h}_{amb}(T_h - T_{amb}) \tag{10}$$

$$m_{oil} c_{oil} \frac{dT_{oil,out}}{dt} = P_{loss} - \tilde{h_m}(T_{oil,out} - T_m) - \tilde{h_h}(T_{oil,out} - T_h) - \dot{m} c_{oil}(T_{oil,out} - T_{oil,in}) \tag{11}$$

Where T_m is the temperature of the moving parts [°C], T_h is the temperature of the house [°C] and T_{amb} is the temperature of the ambient air [°C].

3.3.1 Stationary

To calculate the heat transfer coefficients for the house and the ambient air, the differential equations had to be changed to be used when the system is in steady state. The resulting equations can be seen in equation 12 - 14.

$$0 = \tilde{h}_h(T_{oil,out} - T_h) - \tilde{h}_{amb}(T_h - T_{amb}) \tag{12}$$

$$\dot{Q}_{amb} = P_{loss} + mc_p(T_{oil,out} - T_{oil,in}) \tag{13}$$

$$\tilde{h}_h = \frac{\dot{Q}_{amb}}{T_{oil,out} - T_h} \tag{14}$$

3.4 Existing measurements

The equations in section 3.3.1 were used to calculate $\tilde{h}_h$ and $\tilde{h}_{amb}$. However existing measurements from Volvo Cars were not steady state enough to get reliable data, see section 4.4.1. Because of this a decision was made to make new improved measurements, see section 3.6. During the existing measurements it was noticed that the oil sump temperature and oil to cooler temperature had a temperature difference of about 0.1 °C. Since the temperature difference was so small a decision was made to treat the oil sump temperature as the oil to cooler temperature.

In the new measurements it was then seen that these measurements showed a larger temperature difference between the oil sump and the oil to cooler. The reason behind this larger temperature difference is not known. Since the differential equations were made with only one oil temperature, a decision was made to use only the oil to cooler temperature and not the oil sump temperature for the calculations. This was made since the time limit was not enough to change the equations.

3.5 Simulink

A Simulink model was created with equations 6 - 8. With this model (and with good values of the heat transfer coefficients of the house and ambient air) the heat transfer coefficient of the moving parts and the masses of the oil, moving parts and the housing was tuned to match the transient temperature response for the oil to cooler of transient measurements.

3.6 New measurements

Since the existing measurements were not steady state enough, the number of measurements were to few and had sudden variations in torque and engine speed (which are assumed to be due to that the test vehicle did not have a cruise control so the driver had to manually maintain a constant speed when driving on the road) a decision was made to make more measurements in a more controlled environment.
Three different environments were discussed:

- Real world measurements with a vehicle but with more focus on getting stable steady state measurements at the test track Hällered.
- Rig measurements with the engine and transmission in a rig.
- Dyno measurement with a vehicle.

The real world measurements have the advantage that wind resistance and wind flows are accurate. However wind speed and ambient air temperatures can change from the different test days and during the tests, making the environment less controlled.
Rig measurements have the advantage of being in a controlled environment. However the cooling system in the rig is stronger than the one the vehicles are equipped with. This gives an inaccurate cooling of the transmission making the environment less similar to reality.
Dyno measurements have the advantage of being measured in a controlled environment and that a real vehicle is tested. However, the airflow is simulated using a large fan to blow wind on the radiator to cool the vehicle. This in combination with that the vehicle is standing still without a moving ground underneath makes the airflow around the vehicle different from reality.

Of these three measurement types the dyno was chosen due to its controlled environment and that the cooler is the same as in the future production vehicles. The airflow around the gearbox can change depending on the vehicle speed and the amount of air that is allowed to come in to the engine bay through the grill.
At the beginning of the project different transmissions were subject to investigation. But since time was a limiting factor a decision was made to only focus on one of the transmissions. And the most interesting one for this project was one which have the ability to be heated up by warm oil during a cold start.

3.6.1 Test vehicle

The requirements on the test vehicle that would perform the measurements were:

- Internal combustion engine.
- Transmission which can be heated up with warm oil from the cooler during a cold start.
- Volvo Cars SPA platform

Diesel or Petrol as fuel was regarded as not important for the measurements of the transmission in order to get accurate heat transfer coefficients.
A test vehicle which could match the requirements was borrowed. This was a Volvo S90 with engine and transmission according to the requirements. The test vehicle mounted on the dyno can be seen in figure 8.

Figure 8: Volvo S90 on dyno

Temperature sensors were attached to the transmission to be able to log temperature signals. These sensors measured the temperature in/on the:

.

- Oil sump, drilled in to the bottom of the transmission.
- Oil from the transmission to the cooler.
- Oil from the cooler to the transmission.
- Housing (nr 1), close to the drive shaft.
- Housing (nr 2), on the top of the transmission in the direction of the rear of the vehicle. This point is close to the warm catalyst.
- Housing (nr 3), on top of the metal parts enclosing the torque converter. This sensor is the only housing sensor which does not have oil on the other side of the metal the sensor is attached to. Instead there is air on the other side.
- Housing (nr 4), on the side furthest away from the engine. Opposite side of nr 3.
- Housing (nr 5), on top of the transmission under a TCU (transmission computer).

The temperature sensors can be seen attached to the transmission in figure 9.

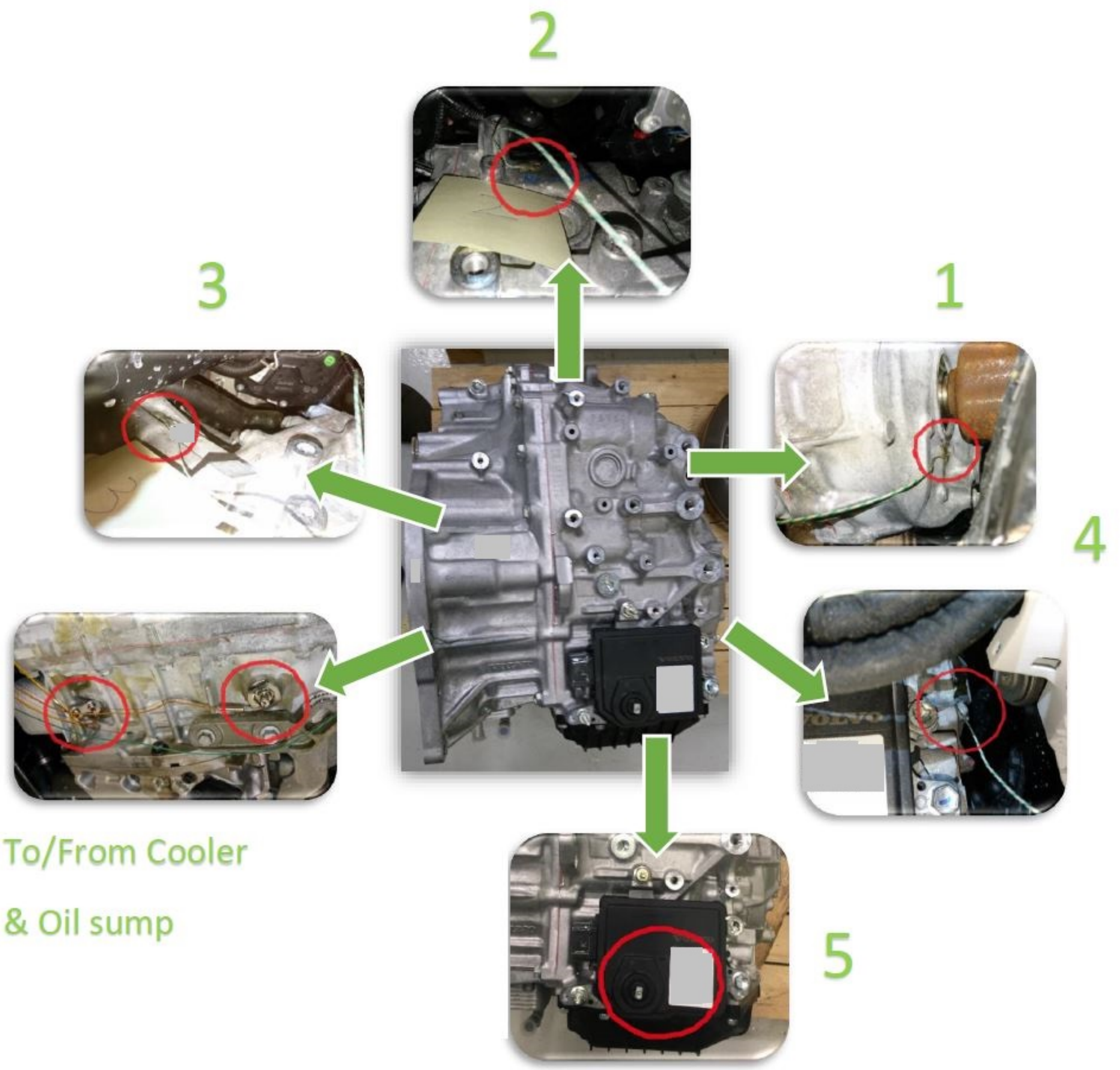

Figure 9: Temperature sensors

3.6.2 Measurement test plan

Before the measurements on the vehicle were made, a test plan was created to make sure all necessary information was obtained during the testing.
It is uncommon to drive a vehicle at high engine speeds and to reduce the risk of overheating the vehicle over a long test, a decision was made to test the vehicle between 1500 rpm and 3500 rpm.
The vehicle was considered to be in steady state when the oil to cooler temperature and housing (nr 5) temperature varied a maximum of 1 °C over 5 minutes. Both a technical definition in combination with a judgment was used to state if the temperatures were steady state or not. The judgmental aspect was introduced since the measurement results showed that the vehicle temperature sometimes could vary more than 1 °C over 5 minutes with a cycling repeating behaviour. The cycling behaviour is believed to be due to that the measurements were done on a vehicle which lacked cruise control, so the driver had to adjust the vehicle speed.

Nine measurement points are needed to get a function of the heat transfer coefficient of the house since it was assumed that it was a function of engine speed and oil temperature. Since the internal oil flow most likely is different when different gears are engaged an idea was raised to use different heat transfer functions for every gear. This would mean 8 functions for the house heat transfer coefficients since there are 8 gears. 9 measurement points used for every gear would mean a total of 72 measurement points for all gears. Since every point was estimated to take at least 20-40 min to achieve with a cool down phase of 1 hour before changing gear, the testing time of two weeks was estimated to be too short. Because of this, a method to reduce the number of gears or measurement points was evaluated.
The power loss for all gears were calculated and plotted. The behaviour is similar between different engine torques and the power losses for one engine torque can be seen in figure 10 with every gear from 1-8 having different lines. This showed that there is a difference in the powerlosses between the different gears leading to the the decision that as many gear as possible should be tested.

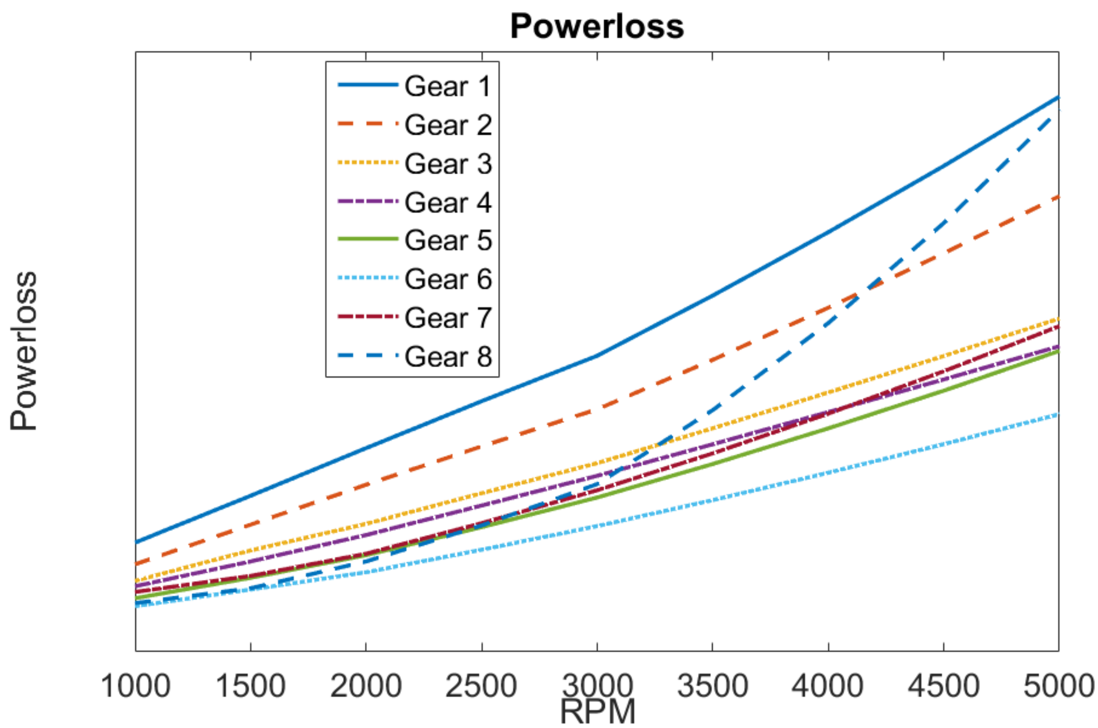

Figure 10: Transmission powerloss

Since gear 3, 4, 5 and 7 show a similar pattern (see figure 10) a discussion was made to simplify the testing and only do one of these gears in order to save testing time. However this idea was discarded since it was considered to be too important to get the correct behaviour for all gears.
Since the vehicle is stationary in a dyno, a fan was placed in front of the vehicle in order to cool down the radiator. To try to mimic the wind from real driving, the air speed was set to the same as the vehicle speed. However, the nozzle from the fan was far smaller than the frontal area of the vehicle making the flow characteristics around the vehicle not entirely realistic.

To verify the tuned simulink model, three simple driving cycles were created to check if the model showed a similar temperature response as the vehicle transmission. One for low speed, one for normal driving conditions and one driving cycle for high vehicle speed. The use of three driving cycles were made to test if the model could be used for different kinds of loads. Every driving cycle was made up of 5 sections with 5 minutes, each with different vehicle speeds. The low- and normal speed driving cycles was done from both cold start and warm temperature (idling vehicle with neutral gear until the temperatures were steady state). The high speed drive cycle was only done from a warm temperature due to limited time with the test dyno. When the vehicle had the vehicle speed 0 km/h the transmission was set to "Parking-mode" with the engine still running.

The low speed driving cycle was made up of:

- 30 km/h for 5 minutes.
- 20 km/h for 5 minutes.
- 0 km/h for 5 minutes.
- 50 km/h for 5 minutes.
- 30 km/h for 5 minutes.

The normal speed driving cycle was made up of:

- 30 km/h for 5 minutes.
- 50 km/h for 5 minutes.
- 0 km/h for 5 minutes.
- 120 km/h for 5 minutes.
- 70 km/h for 5 minutes.

The high speed driving cycle was made up of:

- 100 km/h for 5 minutes.
- 200 km/h for 5 minutes.
- 100 km/h for 5 minutes.
- 150 km/h for 5 minutes.
- 170 km/h for 5 minutes.

During the test of gear 1, a difference in the output speed of the transmission and input speed to the transmission was noticed. This meant that the torque converter was open. There was no possibility in locking the torque converter. Since the added power losses in an open torque converter would disrupt the measurement values for gear 1, and in combination with that gear 1 only is used for a few seconds during normal driving, a decision was made to neglect gear 1. Instead the measurement values for gear 2 were chosen to be used for gear 1. The values from gear 2 for the house heat transfer coefficient and ambient heat transfer coefficient were also used when the vehicle was set to idle. The logical structure for the house heat transfer coefficient (which is gear dependent compared to the ambient heat transfer coefficient which only is dependent on vehicle speed and hence only uses one function) can be seen in figure 18.

Due to time limitations in the dyno it was decided to make most tests at gear 4, 5 and 6 which were regarded to be used often in normal driving. For all gears tests were performed at an engine speed of 1500, 2100, 2800 and 3500 rpm with a single an engine load. For gear 4, 5 and 6 the engine load was measured at more torque levels (at the same four engine speeds for every engine load) to get a wider spread in temperatures in the transmission.

3.6.3 Oil flow

It is difficult to install a device measuring the oil flow from the transmission to the cooler and vice versa since there is a risk of disturbing the flow. Because of this, the flow was calculated with the help of a look-up table (from the oil pump manufacturer) which uses known values for the oil flow at different torque levels and engine speeds. To get the oil flow between the known values and outside of known range, linear interpolation and linear extrapolation was used.

4

Results

The results from calculations, simulations, tuning and verification are presented in the following chapter. Three iterations were done to find good heat transfer coefficients between the oil and the housing ($\tilde{h}_h$) and between the housing and the ambient air ($\tilde{h}_{amb}$).

4.1 Logical model test

The behaviour of the model was tested with both known steady state values and transient steps to see how the model responded. The outputs from the model were oil to cooler temperature, temperature of the housing and temperature of the moving parts (there are no measurements of what the temperature of the moving parts are, since it is very difficult to measure, so this temperature is only a result of how the differential equations are made). In figure 11 the steady state behaviour was tested and regarded as reasonable since the output temperatures of the model should not and did not change with the known steady state input values.

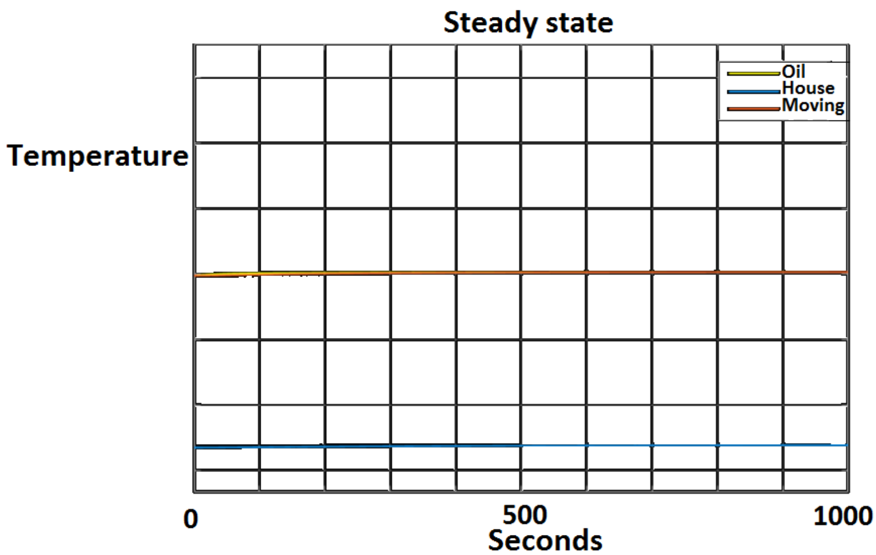

Figure 11: Test of steady state

In figure 12 the system is first at steady state and after 500 seconds there is a decrease in the power loss. The model responds with a steep decrease in the oil temperature followed by a smoother change of the temperature of the housing and the moving parts. This is logical since the power losses are modelled to happen in the oil and because of this the response is fastest in the oil.

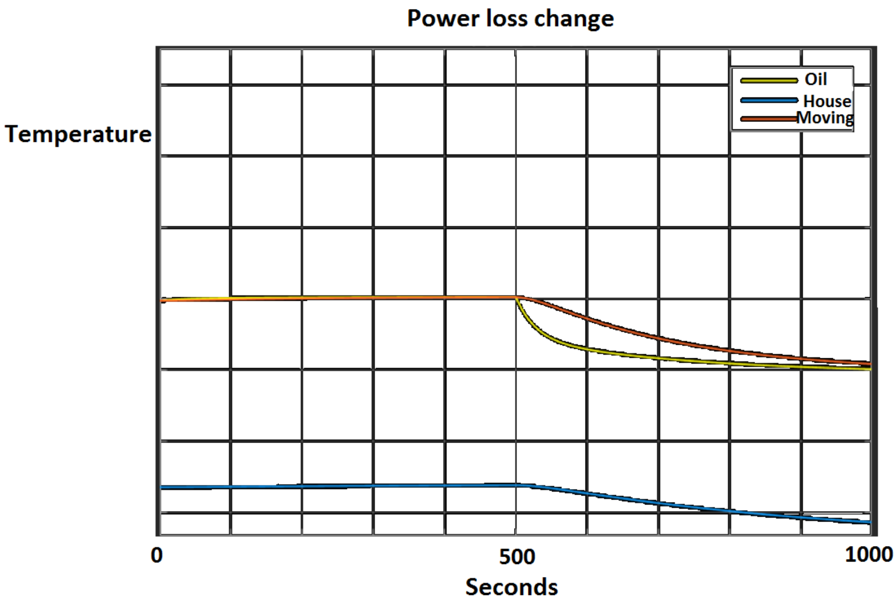

Figure 12: Test with power loss change

In figure 13 the system is first at steady state and after 500 seconds the oil flow, cooling to the system, is reduced to 0 kg/s. The oil temperature then rises immediately with the other temperatures following in a smoother increase. This is representative to how the system is expected to respond in reality.

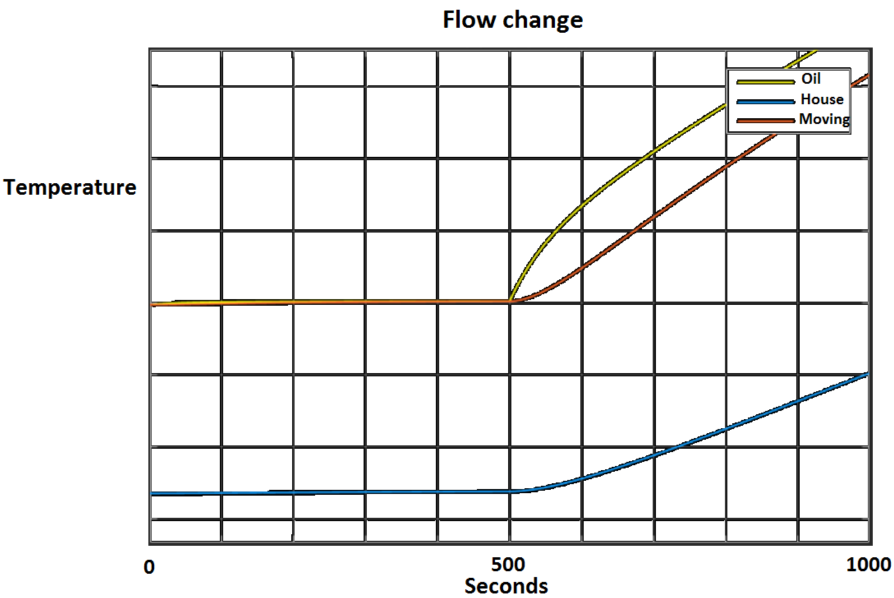

Figure 13: Test with oil flow

In figure 14 and 15 the initial temperature was the same with the only difference between the figures that the ambient heat transfer coefficient was smaller in figure 14. The temperature of the housing is smaller i figure 15 since the ambient cooling is more effective. It can also be noticed the the oil temperature is lower as well, due to the better cooling. This is representative to how the system is expected to respond in reality.

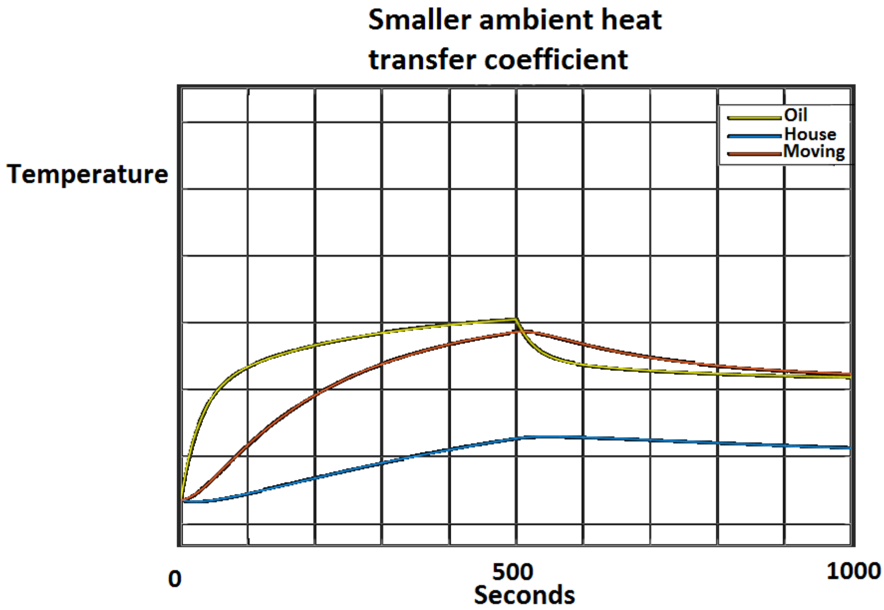

Figure 14: Test with smaller ambient heat transfer coefficient

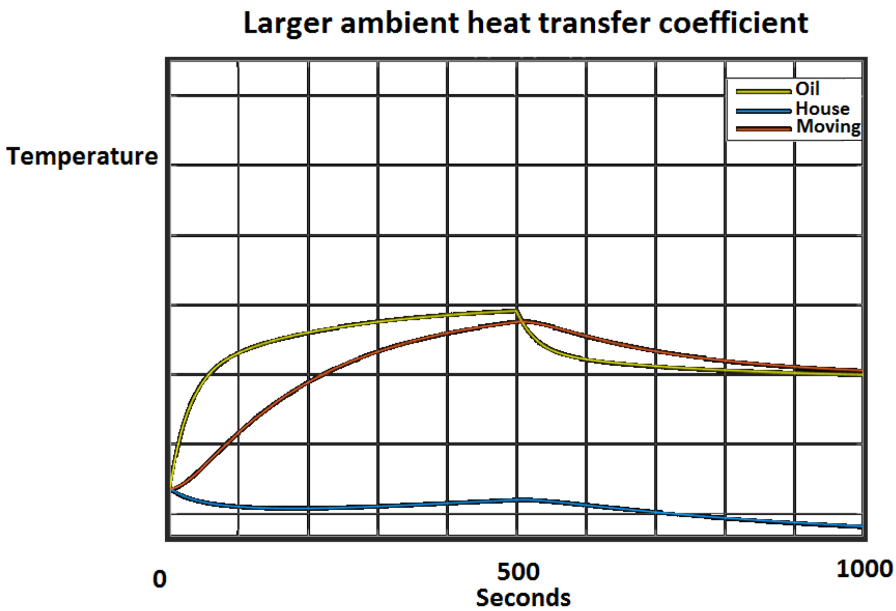

Figure 15: Test with larger ambient heat transfer coefficient

4.2 Sensitivity Test

In order to get a good picture of how the input parameters influence the output of the model a sensitivity test was performed.

4.2.1 Sensitivity test for heat transfer coefficient

The sensitivity test showed that the model is more sensitive to the ambient heat transfer coefficient compared to the house heat transfer coefficient. This means that a change in the temperature difference between the house and the ambient air transfer more heat compared to the temperature difference between the oil to cooler and house.

4.2.2 Sensitivity test for different mass

By changing the masses of the lumped masses the response time until the system was at 70% of a steady state value was evaluated.

The sensitivity test showed that the system is more sensitive to a change in the oil mass and mass of the moving parts than the mass of the housing.

4.3 Power loss

The power losses in the transmission can not be directly measured from measurements. A simulink model with powerloss look-up tables, identical to the existing ones in Vsim for the oil pump and gear box, was created. This ensured that the calculated values for the powerloss, which would be used to calculate the heat transfer coefficients, would be the same that would have been calculated in Vsim with the same input values for torque, engine speed, gear and oil temperature.

4.4 Measurements

Existing measurements were evaluated and new measurements were performed.

4.4.1 Existing measurements

In existing measurements the oil to cooler temperature was unstable and many times too short for the system to reach steady state. An example of a good existing measurement can be seen in figure 16. Here the inconsistent behaviour of the oil to cooler temperature can be seen around 1500 seconds for the left picture and the system not really reaching steady state at 2500 seconds. The right picture shows the inconsistent behaviour of the existing measurements with oil and different housing temperatures.

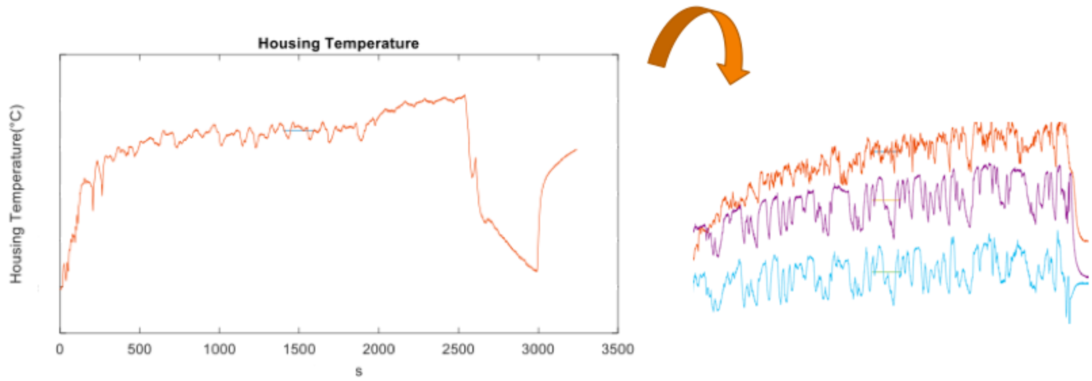

Figure 16: Existing measurement

4.4.2 New measurements

Figure 17 shows one of the new measurements that were performed in the dyno. One gear was run with a constant engine torque but with changing rpm on the engine, which gives different vehicle speeds. The different colors of the lines are described below:

- Green: Vehicle speed
- Teal: Housing 1 temperature
- Pink: Oil sump temperature
- Orange: Oil from cooler temperature
- Blue: Oil to cooler temperature

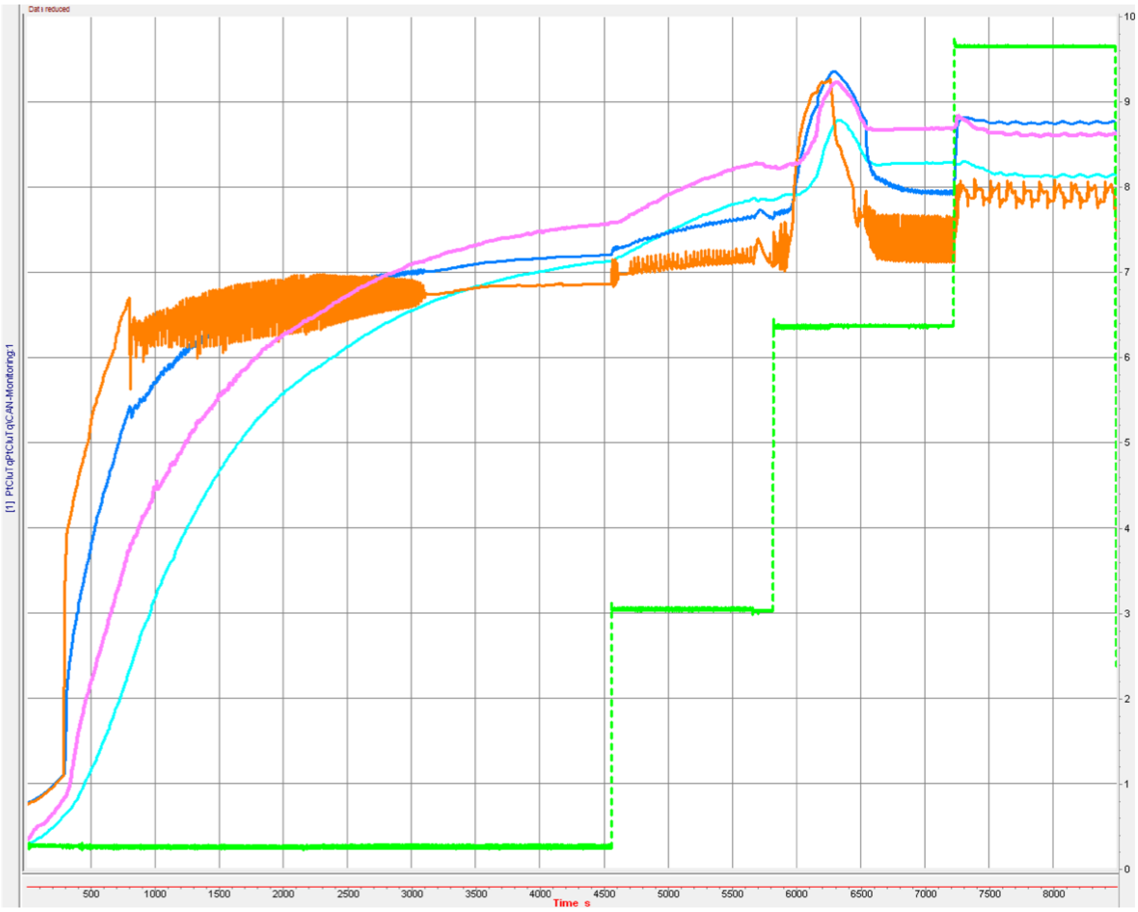

Figure 17: Dyno test for a gear

All the housing temperatures are not presented in this figure to make the picture more clear. The unstable temperature of the orange line is difficult to explain since temperature measurements of the cooling liquid was skipped. Not measuring the temperature of the cooling liquid was a mistake.

It can be seen in figure 17 that the time until the system is at steady state is the longest for the first measurement since the system is cold when the measurements start. This enables the warm up behaviour to be logged.

During the tests it was noticed that the grill shutter in the grill was open during all time when the vehicle was running. This is believed to be due to the relatively warm surrounding air which requires good cooling and hence open grill shutters.

The outside temperature sensor on the vehicle was attached to the inside of one of the rear-view mirrors. This showed for the most part an outside temperature 5-10°C higher than the ambient air. This is thought to be due to that the temperature sensor is affected by the heat from the engine.

Figure 18 shows how the different ambient heat transfer coefficient functions was structured in Simulink. Since gear one does not have any measurements data, an assumption that gear one have the same ambient heat transfer coefficient value as gear two was proposed and when no gear works it has the same ambient heat transfer coefficient value as gear two as well.

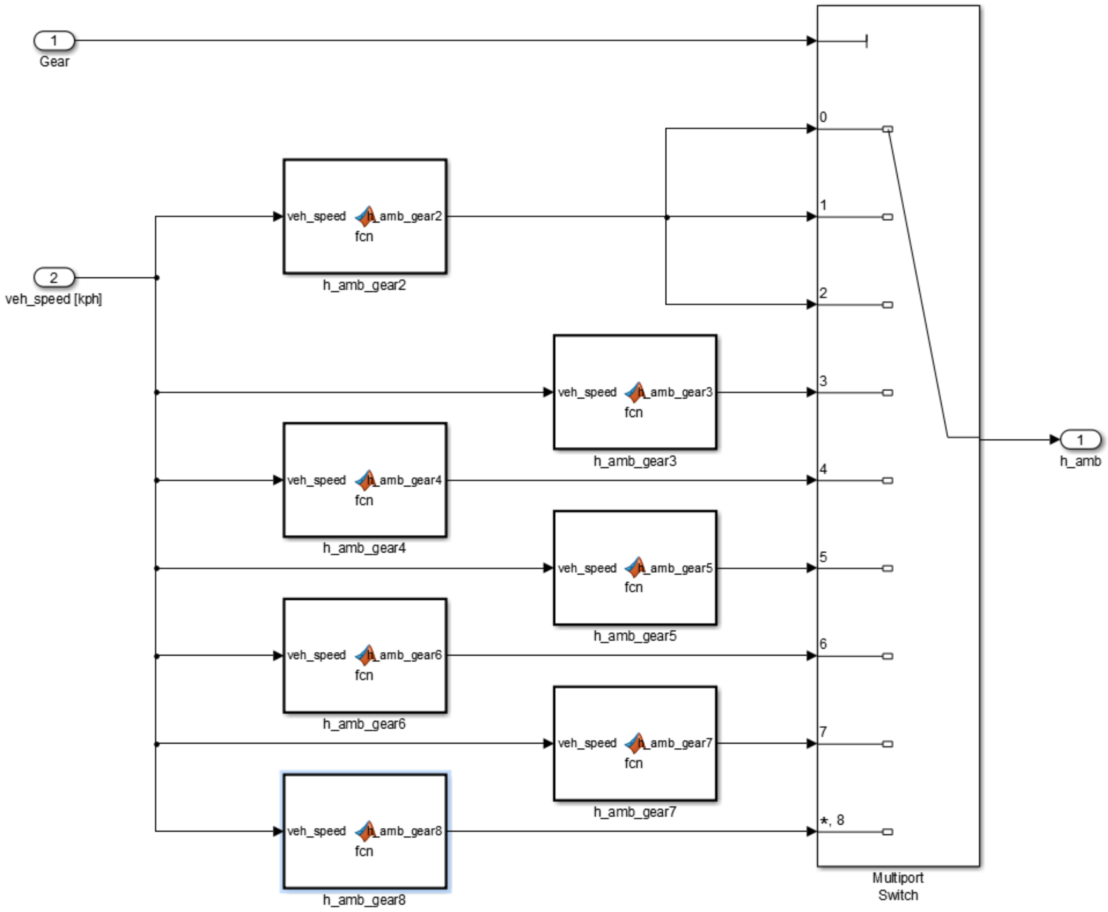

Figure 18: Structure ambient heat transfer coefficient functions

4.5 First attempt heat transfer coefficients

Three iterations to find good house heat transfer coefficient and ambient heat transfer coefficient values were done. The first attempt was with the existing measurements. Figure 19 shows the surface created by the function dependent on oil temperature (the oil sump and the oil to cooler temperature which at this state of time of the project were considered to be the same temperature) and engine speed (rpm). Häggström [1] inspired to make the housing heat transfer coefficient a function of engine speed (rpm). Since equation 5 is dependent on fluid speed (which is dependent on rotational speed which is engine speed) and viscosity (which is dependent on temperature), temperature of the oil was chosen to be a variable in creating the heat transfer surface.

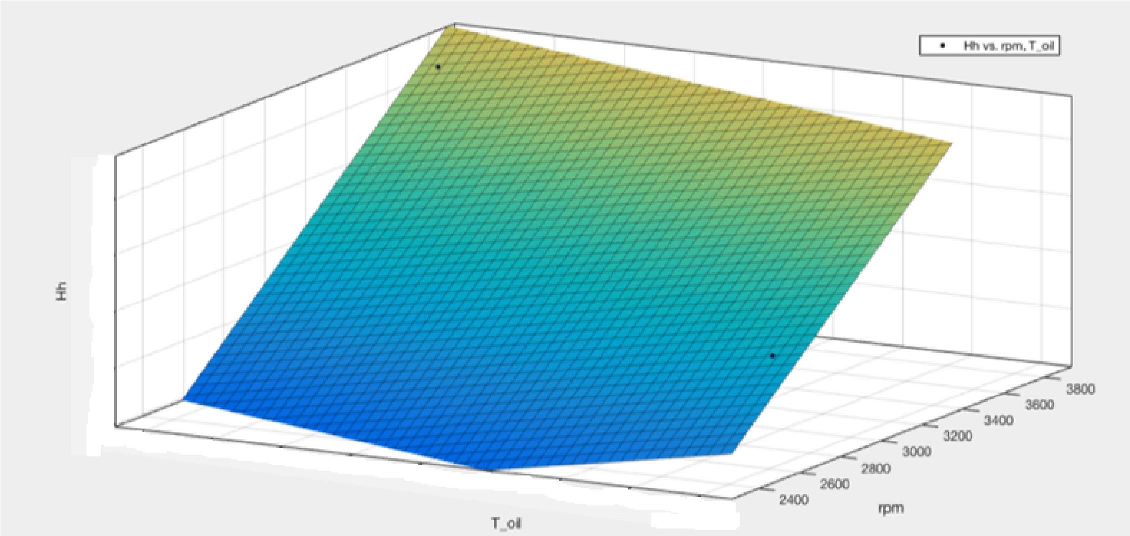

Figure 19: House heat transfer coefficient surface

There are two problems with this house heat transfer coefficient surface:

- Too many vibrations exist in the measurement data; there is not really a steady state as can be seen in figure 16. A rough mean value was then taken as a "steady state" value.
- Only 4 measurements were available, with the gear selector set to "Drive" which produced to few measurements with a too small range in the tested gears. This was not enough to make a good curve surface.

4.6 Second attempt heat transfer coefficients

The second attempt, to get good house heat transfer coefficient and ambient heat transfer coefficient values, were done with the new measurements.

4.6.1 Calculating ambient heat transfer coefficient

The ambient heat transfer coefficient was chosen to be dependent on vehicle speed since the heat transfer coefficient is dependent on the speed of the fluid flow (in this case the air).
During the calculation of the ambient heat transfer coefficient it was noticed that there was a spread for the ambient heat transfer coefficient at the same speed of the ambient heat transfer coefficient, see figure 20. This should not be possible since the ambient heat transfer coefficient should be the same for the same speed. There are even negative ambient heat transfer coefficient values which are physically not possible since this would mean that heat is transferred from a colder to a warmer medium. It was also seen that different ambient heat transfer coefficient values was given for different gears which should also not be possible since the outside of the transmission is not affected by the selected gear. Possible reasons for the spread is discussed in section 5.

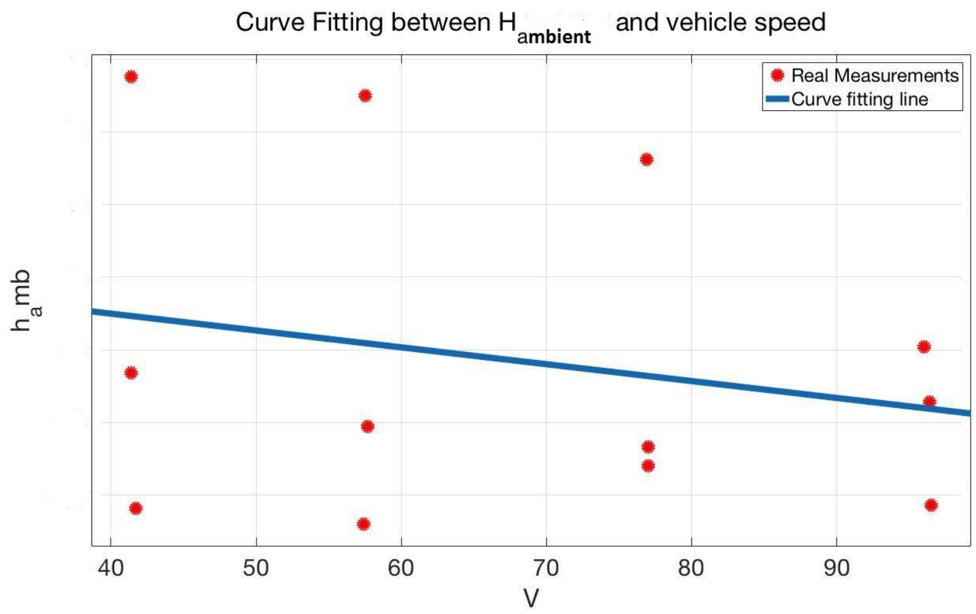

Figure 20: Curve fitting between ambient heat transfer coefficient and vehicle speed for one gear

An approach to use a constant value ambient heat transfer coefficient was discussed and tested. This was later changed to instead make different functions depending on vehicle speed for different gears. The reason to the change was that, due to the findings when doing the sensitivity test, it was seen that the model was quite sensitive to the ambient heat transfer coefficient and to get the best possible results; functions for different gears were chosen, even though this contradicts how the physics work in real life.

4.6.2 Calculating house heat transfer coefficient

Since the new measurements had a lot more measurement points a curve fit with a non-linear surface was done to hopefully be closer to reality. This can be seen for the surface in figure 21.

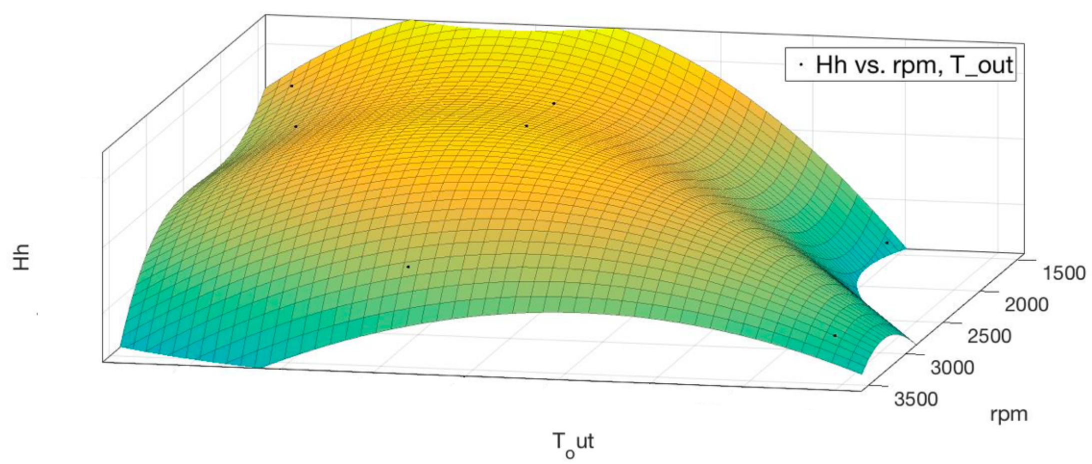

Figure 21: House heat transfer coefficient surface for gear 4, 5 or 6

For gear 2, 3, 7 and 8 not enough measurement point were measured to create a non-linear curve. Because of this the surfaces for these gears are flat as in figure 22.

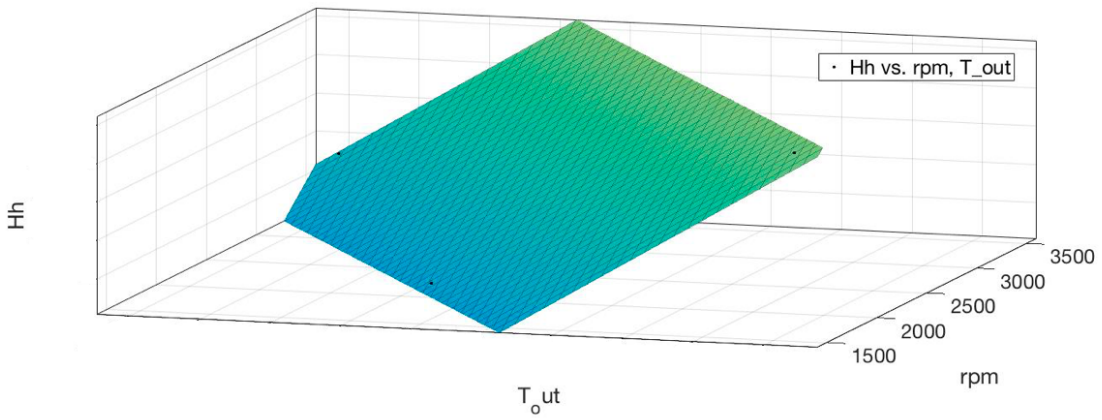

Figure 22: House heat transfer coefficient surface for gear 2, 3, 7, or 8

The house heat transfer coefficient surface values for gear 4, 5, and 6 does only work well in the range where the measurements were done. Since the surface (example is the surface in figure 21) has a non-linear behaviour, the house heat transfer coefficient values will be very low outside the surface, when extrapolation is being used. The consequence of these low house heat transfer coefficient values can be seen in figure 23 where the simulink model (blue line) does not follow the measured values (red line) at all on lower temperatures, but work well on higher temperatures where the measurements were made. Because of the inaccurate low-temperature results, new house heat transfer coefficient values were calculated for gear 4, 5, and 6.

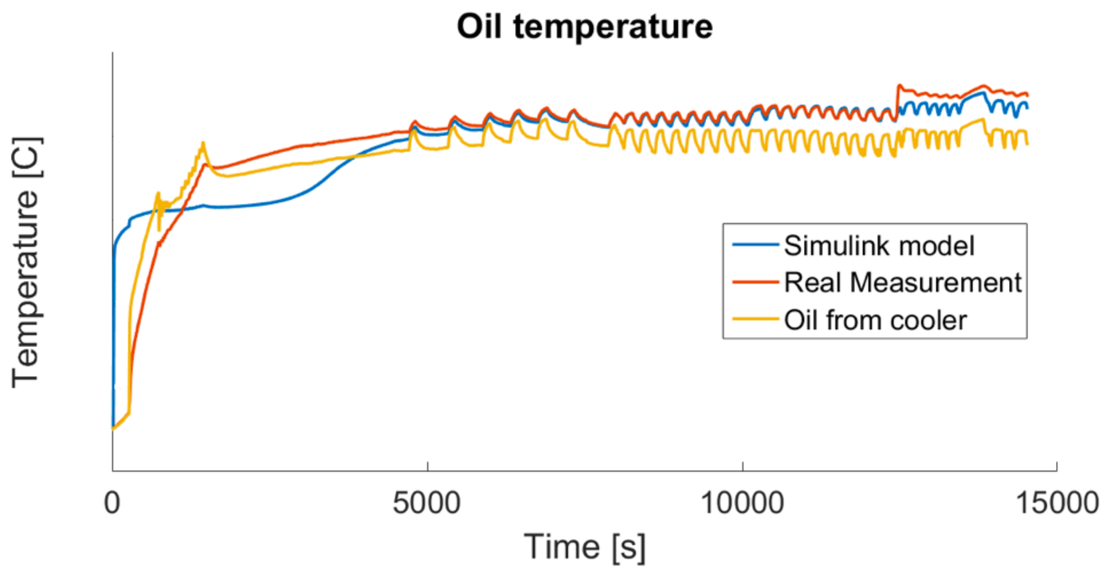

Figure 23: House heat transfer coefficient for high oil temp for a gear

4.7 Third attempt house heat transfer coefficients

A third attempt with house heat transfer coefficient-values were done with the same measurements as in the second attempt (with the new measurements). To get better simulation results for low oil to cooler temperatures, a linear surface was used as in figure 24.

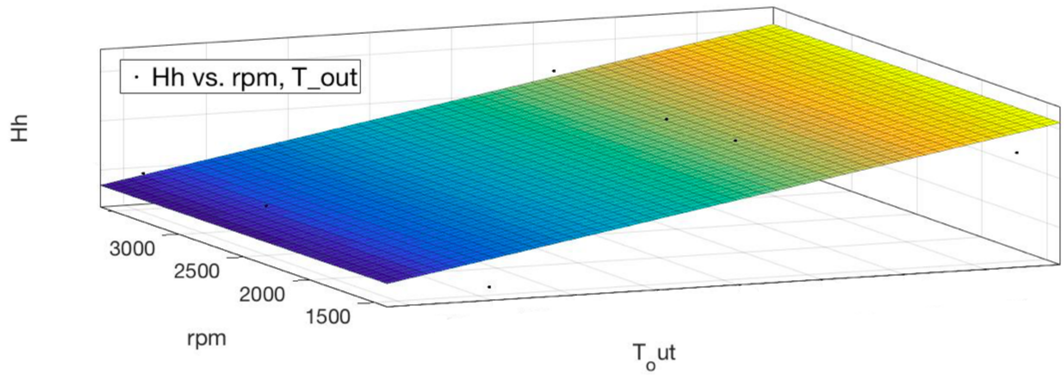

Figure 24: House heat transfer coefficient surface for gear 4, 5 or 6

The house heat transfer coefficient-values from the third attempt gave better simulation results for low oil to cooler temperatures as seen in figure 25.

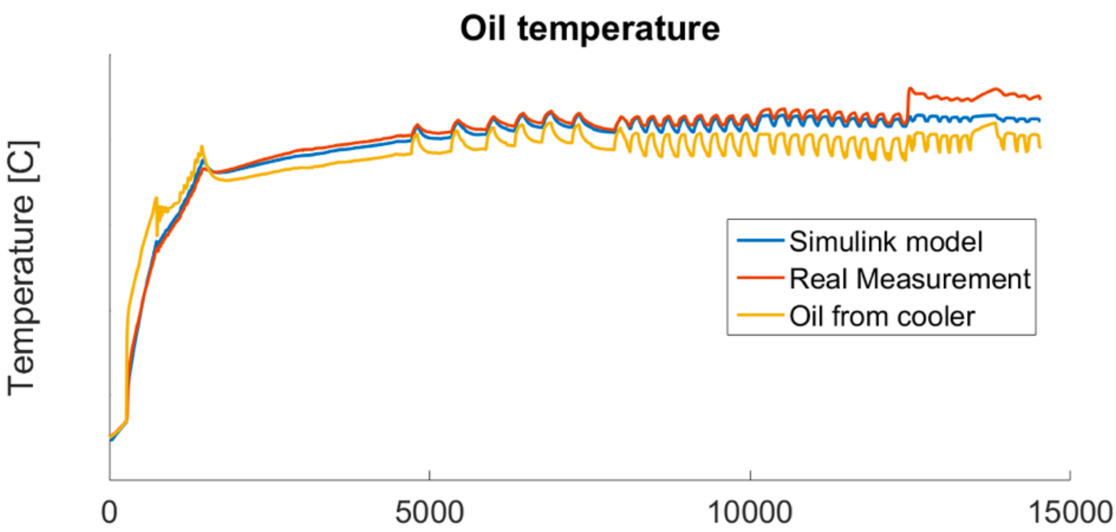

Figure 25: House heat transfer coefficient for low oil temp for a gear

However the result in figure 25 is less accurate for high oil to cooler temperatures compared to figure 23. Because of this a combination of the two house heat transfer coefficient surfaces were made with the linear surface acting for lower temperatures. The non-linear surface was used for higher oil to cooler temperatures. The simulation result of the combined house heat transfer coefficient values can be seen in figure 26. This combined house heat transfer coefficient value was the one used in the model verification in section 4.8.

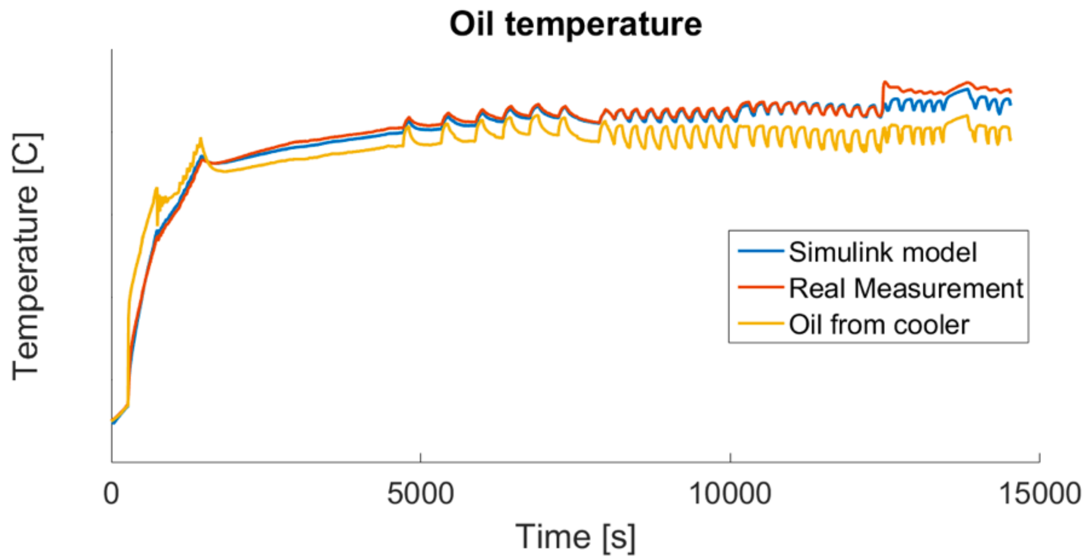

Figure 26: House heat transfer coefficient combined for a gear

4.7.1 Tuning of the heat transfer coefficient for the moving parts and masses

The heat transfer coefficient for the moving parts and masses dictate how fast the system responds to a temperature change. The system was tuned by manually iterating the heat transfer coefficient for the moving parts and the three different masses in the system (oil, moving parts and housing).

4.8 Model verification

When the heat transfer coefficient for the moving parts and masses had been tuned a drive cycle test was being performed on the model. The sample time of the installed temperature sensors (like the one measuring the temperature of the oil from the cooler) was 0.1 seconds. However the sample time from the CAN signals (like torque, engine speed) uses a shorter sample time. This difference in sample time is tricky to handle in simulink so the CAN signal sample times were adjusted to create an input signal with the same sample time of 0.1 seconds.
When a vehicle speed change was requested it was automatically adjusted within a few seconds by the dyno computer. However the road load had to be manually adjusted by the dyno-operator. This manual operation could take up 20 seconds to perform for a speed change when the vehicle was moving and up to 70 seconds when the vehicle had been idling. This was due to that the operator had to manually climb into the vehicle and go from "Neutral" to "Drive" with the transmission shifter and then from the control room adjust the road load. This made the measurements of the torque of the engine varying for the beginning of a new vehicle speed.
The drive cycle uses different speeds with real world road load, see section 3.6.2, with the transmission set to "Drive" to allow the transmission to use the most suitable gear.

The result of the normal drive cycle started from a cold start can be seen in figure 27. The difference between the red line (Real Measurement of oil to cooler) and the simulated blue line (also oil to cooler) is a maximum of 6.5 °C when the oil from cooler rises quickly and a maximum of 2.5 °C when the oil from cooler has a more steady behaviour.

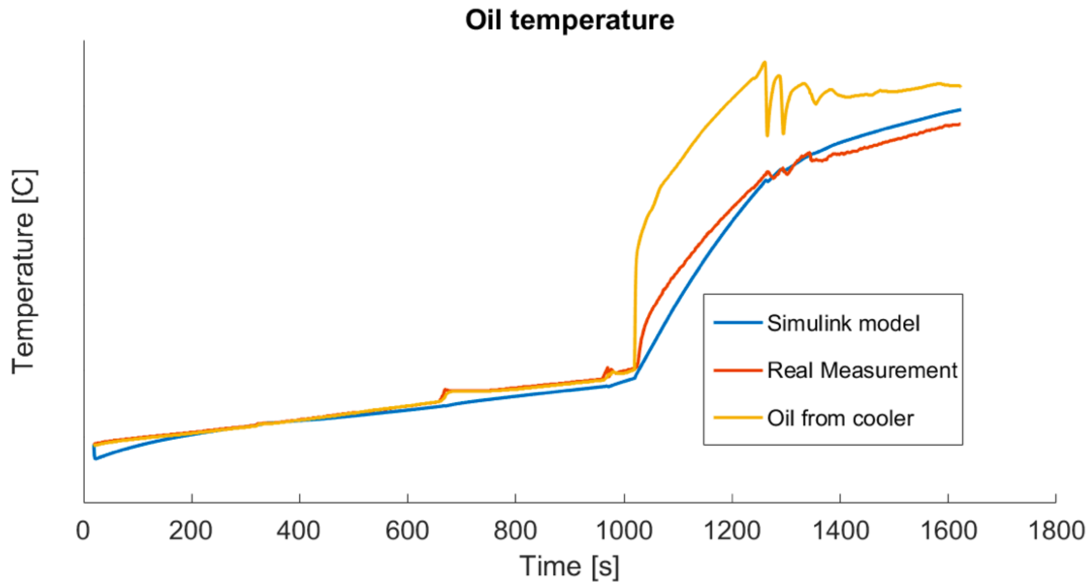

Figure 27: Normal drive cycle, cold start

5
Conclusion

The aim of the project was to build a dynamic temperature model for the automatic transmission, which should be able to handle any drive cycle and load. Especially important was the warm up behaviour since the heat up process of the transmission will be faster on the new model.

Due to time limitations only one gearbox model was chosen to be investigated. The result shows that the warm up difference between the simulated model and real measurements during a driving cycle was a maximum of 2.5 °C when the temperature of the oil from the cooler showed a slower and more normal increase. A maximum difference of 6.5 °C was noticed when the temperature of the oil from the cooler made a rapid increase in temperature. This means that the model can fairly well predict the temperature to the cooler if there is no big difference in the input oil temperature (which can happen during a warm up when the engine is warm enough and the control system allows the transmission to use the heat from the engine cooling). This bigger difference last for about 3 minutes until the system is once again having a maximum temperature difference of 2.5 °C. This means that the model is fairly accurate at predicting a driving scenario similar to real world measurements. The model gives output in oil flow [L/min] and oil to cooler temperature [°C].

During the project three different ways, when using a black box model, which can handle transient behaviour was discussed. The aim of the project is hence stated to be met.

5.0.1 Discussion

The model does only handle how the transmission behaves when there already exists a correct input temperature of the oil from the cooler. This means that the model is not yet ready to be used in Volvo Cars Vsim software since no cooler model exists.

The model can only be predictable when the grill shutter is open. This is due to that the grill shutter was only open when the measurements were done (due to that the temperature in the dyno was warm, so the vehicle needed the cooling capability).

One error in the setup could be that only one measurement equipment was used and only one transmission was tested. However the temperature sensors are reported to be consistent and have a tolerance of 0.1°C so the measured values are probably correct.

The flow and air cooling around the vehicle is probably lower than reality since the ground under the vehicle is stationary and the fan generating air to the cooler of the engine has about the same size as the grill of the vehicle and can not cover the entire vehicle with realistic air flow.

The model can not be used for very low speeds, when only gear 1 is engaged, since no measurement data of gear 1 could be measured.

The ambient heat transfer coefficient values are sometimes negative. This is not physically possible since this would mean that heat is flowing from a colder surface to a warmer surface. However even though this is not possible in reality this was left negative in the model, for some ambient heat transfer coefficient values to make the model more accurate. The negative ambient heat transfer coefficient values is probably due to the limitations of the simple mathematical model which was used. This means that heat in form of radiation and conduction is missed in the model.
Temperature sensor 2 gives varying results. This is assumed to be so since it is affected by other equipment like the catalyst which sometimes burn fuel to clean itself, which creates extra heat.

5.0.2 Future work

Future work to improve the model and to make it possible to work with Vsim are:

- Improve model with varying c_p of the oil and metals.
- Create an "oil from cooler" -model and link it to the dynamic temperature model to allow the dynamic temperature model to be used in Vsim.
- Do more dyno tests with a large range of rpm and temperature values to get more accurate heat transfer coefficient surfaces. This is especially important for gear 7 and 8, which are used more often in driving than first anticipated.
- Implement radiation between housing and environment in the mathematical model.
- Verify the model with more drive cycles (these drive cycles are already measured but not tested due to time limitations)

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