Modelling and Validation of a Truck Cooling System

Master's thesis performed in Vehicular Systems
by

Erik Nordlander

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Modelling and Validation of a Truck Cooling System

Modellering och validering av kylsystemet i en lastbil.

Författare

Erik Nordlander

Abstract

In the future, new challenges will occur during the product development in the vehicular industry when emission legislations getting tighter. This will also affect the truck cooling system and therefore increase needs for analysing the system at different levels of the product development. Volvo 3P wishes for these reasons to examine the possibility to use AMESim as a future 1D analysis tool. This tool can be used as a complement to existing analysis methods at Volvo 3P. It should be possible to simulate pressure, flow and heat transfer both steady state and transient.

In this thesis work a cooling system of a FH31 MD13 520hp truck with an engine driven coolant pump is studied. Further a model of the cooling system is built in AMESim together with necessary auxiliary system such as oil circuits. The model is validated using experimental data that have been produced by Volvo 3P at the Gothenburg facility.

The results from validation and other simulations show that the model gives a good picture of the cooling system. It also gives information about pressure, flow and heat transfer in steady state conditions. Further a design modification is done, showing how a change affects the flow in the cooling system.

The conclusion is that a truck cooling system can be built and simulated in AMESim. Further, it shows that AMESim meets the requirements Volvo 3P in Gothenburg has set up for the future 1D analysis tool and thereby AMESim is a good complement to the already existing analysis method.

Nyckelord
Cooling system, Heat transfer, AMESim, Vehicular Systems, 1D Simulation

Abstract

In the future, new challenges will occur during the product development in the vehicular industry when emission legislations getting tighter. This will also affect the truck cooling system and therefore increase the need for analysing the system at different stages of the product development. Volvo 3P wishes for these reasons to examine the possibility to use AMESim as a future 1D analysis tool. This tool can be used as a complement to existing analysis methods at Volvo 3P. It should be possible to simulate pressure, flow and heat transfer both steady state and transient.

In this thesis work a cooling system of a FH31 MD13 520hp truck with an engine driven coolant pump is studied. Further a model of the cooling system is built in AMESim together with necessary auxiliary systems such as oil circuits. The model is validated using experimental data that have been produced by Volvo 3P at the Gothenburg facility.

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Sammanfattning

I framtiden kommer det att ställas allt större krav på utvecklingen av fordon bland annat på grund av lagstiftning. Detta kommer även att påverka kylsystemet och därmed öka behovet av analysering i olika delar av produktutvecklingen. Volvo önskar av denna anledning undersöka möjligheten att använda programmet AMESim som ett verktyg i framtidens 1D analys. Det skall då kunna fungera som ett komplement till de analysmetoder som används idag. Verktyget skall kunna klara av att simulera tryck flöde och värmetransport både statisk och transient.

I detta examensarbete studeras kylvätskesystem i en FH31 MD13 520 hk med en motordriven kylvätskepump. Vidare byggs en modell av detta kylvätskesystem upp i AMESim tillsammans med nödvändiga grannsystem såsom vissa oljekretsar. Den framtagna modellen har sedan validerats mot mätdata som har tagits fram i en mätrigg internt på Volvo 3P i Göteborg.

Resultatet från validering och ytterligare mätningar visar att modellen återspeglar det verkliga system bra. Den kan också ge de svar som önskas för både flöde, tryck och värmetransport i systemet. Detta har visats genom att ändra designen för kylvätskekretsen i form av borttagande av komponenter och ändring i geometri.

Slutsatserna av detta examensarbete visar att ett verkligt kylsystem kan byggas upp och simuleras i AMESim. Ytterligare går det att säga att AMESim uppfyller de önskemål Volvo 3P i Göteborg har av framtidens 1D verktyg och att AMESim därmed kompletterar redan använda analysmetoder.

Preface

This master thesis was performed at Volvo Group AB, 3P in Gothenburg and at the Department of Electrical Engineering, division of Vehicular Systems, at Linköping University.

The Volvo Group is one of the leading suppliers of commercial transport solutions providing products such as trucks, buses, construction equipment, and drive systems for marine and industrial applications as well as aircraft engine components. The Volvo Group has about 100 000 employees around the world. Volvo 3P is a business unit within the Volvo Group. Volvo 3P combines the resources of the truck companies Volvo Trucks, Renault Trucks, Mack Trucks and Nissan Diesel in the areas of product planning, product development, purchasing and product range management. Volvo 3P works in partnership with the truck companies to ensure a powerful and strong competitive offer for each brand. The thesis work has been carried out at Cooling Analysis within Powertrain Installation, which is a section within the Chassis Department at Volvo 3P in Gothenburg.

Acknowledgement

I would like to thank everyone who helped me with this thesis work both in Gothenburg and Linköping. There are some people that I will always be special grateful to. Especially I would like to thank my supervisor at Volvo 3P, Maria Krantz, for giving me the opportunity to do this thesis work and supporting me through these months. There have been a privilege knowing you and I am looking forward working with you as a college at Volvo 3P. I would also like to thank my wife Hanna and my daughter Amanda that have been very patient with me throughout my studies and especially during this thesis work. Without you this would be a more difficult journey. Finally I want to dedicate this report to my newborn daughter Linnea who was born during this work.

Linköping March 2008

Erik Nordlander

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1 Introduction

In this chapter a background for the thesis work is presented first. Thereafter the chosen approach of the problem will follow and then the method for solving it. Finally limitations and a short overview for this report are showed.

1.1 Problem Description

When a cooling system is designed the engineer must be aware of the different needs there are for each component, which in some sense can be seen as customers of the system. These needs can for example be defined as a flow at a certain engine speed but also a pressure limitation over a defined interval. The engineer must then design a system that can offer each component a flow at a specific pressure that is at least the specified. In order to design the cooling system, different types of analysis are required: both three-dimension (3D) and one-dimension (1D). For example a 3D analysis can be done using Computational Fluid Dynamics (CFD) which can solve and analyse problems that involve fluid flows. Additional 1D analysis can show if the cooling capacity is enough or how the flow will propagate and distribute around the cooling circuit. This thesis work will only cover 1D analysis and therefore 3D analysis will not be covered in this report. For these types of 1D analysis a number of data tools are available on the market for the analyst. For this thesis work a tool called AMESim has been use for 1D analysis.

There is an increasing need for more analysis including heat transfer in the cooling circuit. Today heat transfer in each component and branch is not included in the analysis because the cooling capacity is enough for the system. In the future however an increasing complexity of the vehicle engine cooling systems in combination with requirements on optimised cooling for different load conditions drive the need for fast and reliable processes for engine heat management simulations. For these reasons a new method in a one-dimension tool are required.

1.2 Thesis Objective

To be able to meet the future demand the Cooling Analysis group within Volvo 3P need to further develop cooling system simulation methods. The main task for this thesis work is therefore to build a computational model to simulate the thermal response of the cooling system of a truck under steady and transient operation. This simulation method will be developed in the one-dimension simulation tool AMESim. This thesis work result in following:

- A complete model of a truck cooling system calibrated against experimental data
- Suggestions for future development and system design
- Valuation of AMESim as a future one-dimension tool for the Cooling Analysis group.

1.3 Limitations

This report will only handle the cooling system of a FH31 MD13 EURO5, 520hp truck. Because of this reason only components in this cooling system will be modelled and other types will not be considered. Also a more detailed study of the cooling package will not be done in this thesis work. Additional only the 1D tool AMESim has been studied and other 1D and 3D tools have been ignored.

1.4 Method

In order to fulfill the mission of this thesis work a number of steps have to be done. Each step contains one or more important tasks that need to be done before moving to the next step. In Figure 1 these steps and tasks are presented to help the reader understanding what has been done.

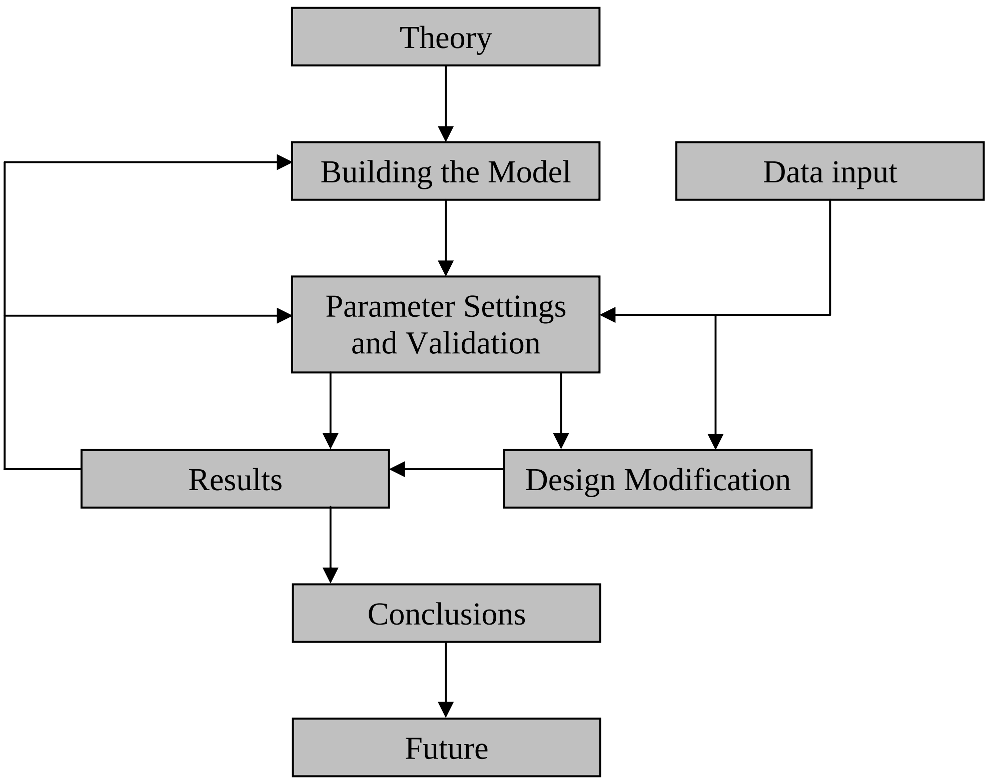

Figure 1. Thesis work research method.

1.5 Report Overview

The steps and tasks illustrated in Figure 1 can be translated to 8 chapters. In Figure 1 the introduction chapter is excluded. Every chapter deals with a certain task. The chapters are

1. Introduction

   Initial discussion of the thesis work, problems and questions.

2. Theory

   In this chapter an overview of the fundamental theory that gives the equations that are needed for building and calculating the flow, pressure and heat transfer.

3. Input Data

   To be able to build the model input data is needed in order to set boundary conditions and parameters. Other data is also needed to validate the model. This chapter describes the sources of data and how it is used.

4. Building the Model

   Here is each component in the studied cooling system described and how they are implemented in AMESim. This will gives an understanding for the actual system.

5. Parameter Settings and Validation

   This chapter deals with the validation of the AMESim model and illustrates eventual problem with the model. The parameter setting is also described in this chapter in order give a distinct picture of the model building.

6. Design Modification

   This chapter gives a possibility to see how the coolant system can be modified in order to change the distribution of flow.

7. Results

   The result from building the model and the comparison done in this thesis report will be presented.

8. Conclusions

   In this chapter a discussion around the result and the thesis objectives and questions will be done.

9. Future

   At the end a discussion around the future possibilities and development using the simulation method that is the goal with this thesis project.

1.6 The Cooling System

In this report a specific cooling system (circuit) will be covered. The following Figure 2 can help understand the problem better and the terminology used in the report. More information about the circuit of study can be found in Chapter 4.

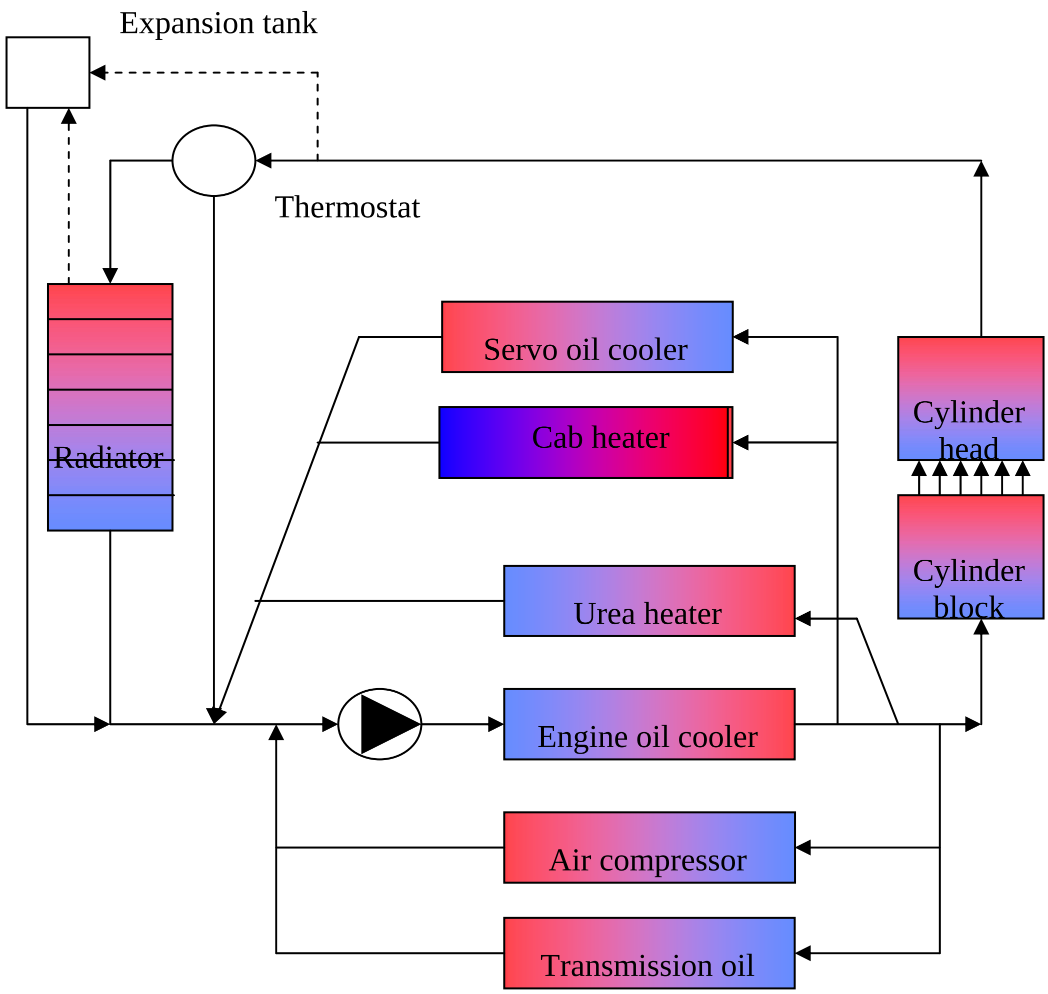

Figure 2. The cooling system.

2 Theory

There is a need to understand the physics when designing and analysing a truck cooling system. These equations of physics can as well describe pressure changes throughout the circuit as over a unique component as the flow rate in a pipe. This thesis report also handles the heat transfer in the system and therefore equations for solving and describing the physics are needed. The theory behind the physics and the calculations will be presented in this chapter.

2.1 Pressure Drop

In a coolant circuit each component will result in a pressure drop or a pressure rise in the fluid, in this report referred as coolant. For example a heat exchanger will result in a pressure drop due to the turbulence phenomena, friction and the viscosity of the coolant. The pressure drop or rise is therefore a change in energy. Due to the conservation law the kinetic energy is constant in a simple connected circuit or control volume, but the potential energy will change into heat where it is friction with the wall. In the coolant pump a pressure rise will occur because energy is transfer to the coolant by mechanical energy from the shaft. For each coolant system (circuit) there is a specific pressure drop curve, which can be called the system curve. The pressure drop is a function of volume flow rate and can be describes as

$$\Delta p = \alpha \cdot \dot{V}^2 \qquad (1)$$

where Δp is the pressure drop, $\dot{V}$ is the volume flow rate and α is a system constant that is unique for each system. Such a curve is presented below in Figure 3. If the flow rate increases the pressure drop will also increase because friction increases with the coolant velocity and the losses are increasing. [5]

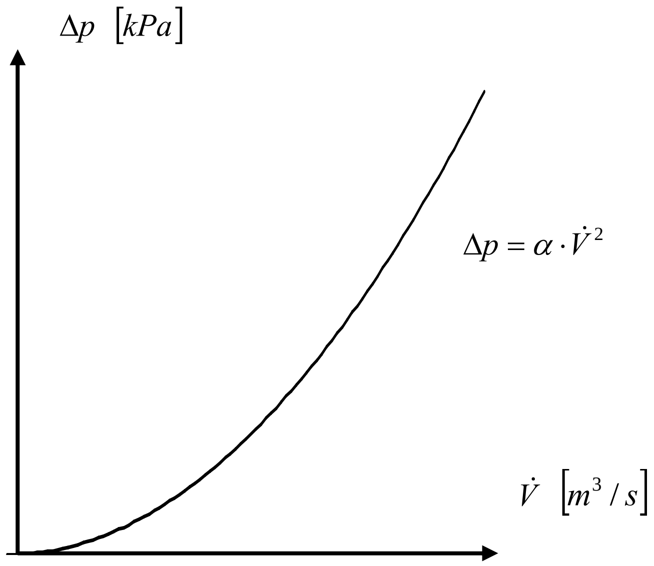

Figure 3. The system curve.

Because the coolant circuit have pipes and hoses with bends, expansion, contraction, junction and inlets there will be pressure losses due to this geometry changes. There will also be as mentioned before pressure losses due friction. To calculate these losses the following two equations can be used

$$\Delta p = K_{fric} \cdot \frac{L}{d} \cdot \frac{\rho \cdot U^2}{2} \qquad (2) \qquad\qquad \Delta p = K_{loss} \cdot \frac{\rho \cdot U^2}{2} \qquad (3)$$

where K_{fric} and K_{loss} are the friction and loss coefficients, L is the tube length, d is the tube diameters, ρ is the density of the coolant and U is the velocity of the coolant. The loss coefficients can be received from diagram and look-up tables. [5]

2.2 Calculation in Pipes

Between most components there are pipes and hoses which will affect pressure and flow. It will also affect the enthalpy flow rate. The pipes can also have bends, contraction and junctions combined with friction that will affect the system.

A pipe can be seen as a thermal - hydraulic component, which consists of capacitive and resistive elements. In resistive element enthalpy and mass flow rate can be calculated. Resistive elements can for example be bends, contractions and junctions, which are connected to each other with a capacitive element. The capacitive element is a volume, where temperature and pressure can be calculated. Each type of element will be described below for flow, pressure and enthalpy. In this report enthalpy and heat transfer will be treated as the same phenomena. The communication between the elements can also be seen in Figure 4.

2.2.1 Capacitive Element

For a coolant circuit both capacitive and resistive elements have to be used if pressure losses and flow rate distribution should be analysed. A capacitive element has the ability to store pressure energy in the volume of the coolant. This is illustrated in Figure 4.

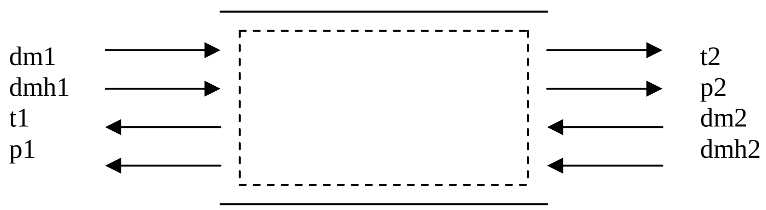

Figure 4. Capacitive element.

The element exchange information with the elements. It has two outputs and two inputs at each end. These variables are neighbouring presented in Table 1.

Port	Input/output	Variable	Unit	Variable name
1	Output	Pressure	bar	p1
1	Output	Temperature	^oC	t1
1	Input	Mass flow rate	kg/s	dm1
1	Input	Enthalpy	W	dmh1
2	Output	Pressure	bar	p2
2	Output	Temperature	^oC	t2
2	Input	Mass flow rate	kg/s	dm2
2	Input	Enthalpy	W	dmh2

Table 1**.** Input and output variables for a capacitive element.

Two of the variables in the capacitive element are state variables and these are pressure and temperature. The pressure is a state variable and is computed from the mass conservation assumption and that the properties of fluid are homogenous in a control volume marked with dot lines in Figure 4. The mass of liquid in the volume is given by

$$\dot{m} = \rho \cdot V \tag{4}$$

where m is the mass, ρ is the density and V is the volume. The continuity equation for the one-dimensional flow gives

$$\frac{dm}{dt} = dm_i - dm_o \tag{5}$$

where dm_i is the incoming mass flow rate in the volume and dm_o is the outgoing mass flow rate showed in Figure 5.

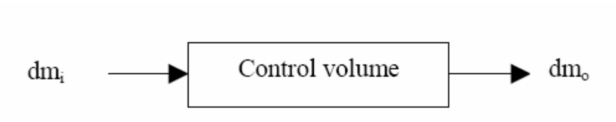

Figure 5. Mass flow rate through the control volume.

From Equation 4 and Equation 5

$$\frac{d\rho}{dt} = \frac{\frac{dm}{dt} - \rho \frac{dV}{dt}}{V} \tag{6}$$

can be derived. The density being a thermodynamic property of the liquid, it is a function of pressure p and temperature T

$$\rho = \rho(p, T) \tag{7}$$

By differentiating with respect to temperature and pressure, Equation 7 leads to

$$d\rho = \frac{\partial \rho}{\partial p} \cdot dp + \frac{d\rho}{dT} \cdot dT \tag{8}$$

which can be rewritten as

$$dp = \frac{1}{\frac{\partial \rho}{\partial p}} \left[d\rho - \frac{d\rho}{dT} \cdot dT \right] \tag{9}$$

Using the definition of the liquid properties, the pressure derivative with respect to time is given by

$$\frac{dp}{dt} = \beta \cdot \left[\frac{1}{\rho} \cdot \frac{d\rho}{dt} + \alpha \frac{dT}{dt} \right] \tag{10}$$

where

$$\beta(p,T) = \frac{\rho}{\frac{d\rho}{dp}} \tag{11}$$

is the isothermal fluid bulk modulus and the volumetric expansion coefficient is

$$\alpha(p,T) = \frac{1}{\rho} \cdot \frac{d\rho}{dT} \tag{12}$$

The temperature is also a state variable and can be computed from the energy conservation assumption. The relationship between the specific internal energy and the specific enthalpy of the liquid is

$$u = h - \frac{p}{\rho} \tag{13}$$

where u is the specific internal energy and h is the enthalpy. The energy in the control volume is then

$$E = m \cdot u + \frac{m \cdot V^2}{2} + mgz \tag{14}$$

Further the kinetic and potential energies in the control volume are neglected. This gives

$$E = m \cdot u \tag{15}$$

The derivative of the enthalpy can be written as

$$\frac{dh}{dt} = c_p \frac{dT}{dt} + \frac{(1 - T \cdot \alpha)}{\rho} \frac{dp}{dt} \qquad (16)$$

where the volume is constant. Combining Equations 13, 15 and 16 leads to the following relation

$$m \cdot c_p \frac{dT}{dt} = dmh_i - dmh_o + \dot{Q} + \frac{m \cdot \alpha \cdot T}{\rho} \frac{dp}{dt} - dm \cdot h \qquad (17)$$

where c_p is the specific heat capacity of the fluid at constant pressure, Q is the heat flow rate exchanged with the outside, dm is the mass flow rate through the volume and α is the volumetric expansion coefficient. Finally, the temperature derivative with respect to time is given by

$$\frac{dT}{dt} = \frac{dmh_i - dmh_o - dm \cdot h + \dot{Q}}{m \cdot c_p} + \frac{T \cdot \alpha}{\rho \cdot c_p} \frac{dp}{dt} \qquad (18)$$

Equation 10 and Equation 18 give the dynamics. [3]

2.2.2 Resistive Element

A capacitive element is the connection between two resistive elements. These two physically represents the coolant flow. A resistive element is illustrated in Figure 6.

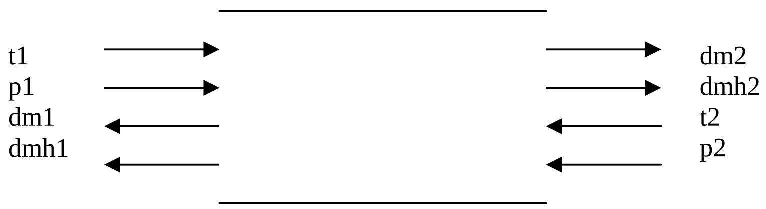

Figure 6. Resistive element.

The element exchange information with the elements. It has two outputs and two inputs at each end. These variables are neighbouring presented in Table 2.

Port	Input/output	Variable	Unit	Variable name
1	Input	Pressure	Bar	p1
1	Input	Temperature	^{o}C	T1
1	Output	Mass flow rate	kg/s	dm1
1	Output	Enthalpy	W	dmh1
2	Input	Pressure	Bar	p2
2	Input	Temperature	^{o}C	T2
2	Output	Mass flow rate	kg/s	dm2
2	Output	Enthalpy	W	dmh2

Table 2**.** Input and output variables for a resistive element.

Other elements could be seen as a special type of resistive elements. These can be for example a bend, a contraction or a junction. A common thing between these elements is a pressure loss due to the geometry and friction, which can be calculated by using a friction factor.

To calculate the mass flow rate at both ports in a resistive element the orifice law can be used which gives

$$dm = C_q \cdot A_{\min} \cdot \sqrt{2 \cdot \rho \cdot |\Delta p|} \tag{19}$$

where C_q is the flow coefficient and $A_{\min}$ is the minimum cross section area. Further the enthalpy flow rate at both ports can be deduced as a function

$$dmh = dm \cdot f(p,T) \tag{20}$$

with pressure and temperature as input. To be able to calculate pressure losses in a resistive element the pressure at each port is used. The pressure is input in the resistive element and is therefore not calculated in it. The friction factor will be showed for each type of element below. [3]

2.2.3 Bend

When modelling a pipe there will often be one or several bends. These can have different centre angle θ_b, curvature radius *r* and diameter *d*. In Figure 7 these parameters are showed. [3]

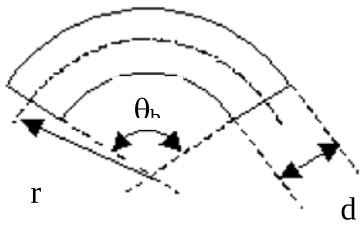

Figure 7. General bend.

The calculation of the friction factor for a bend is very complex and therefore a table or a diagram is used. These can be found in different books about flow calculation. In general the friction factor can be seen as the following function [3]

$$k_b = f(r,d,\theta_b) \tag{21}$$

Enthalpy flow rate, temperature and mass flow rate are calculated as a general resistive element.

2.2.4 Expansion/Contraction

When the diameter of a pipe change, an expansion/contraction element can be used. This is illustrated in Figure 8.

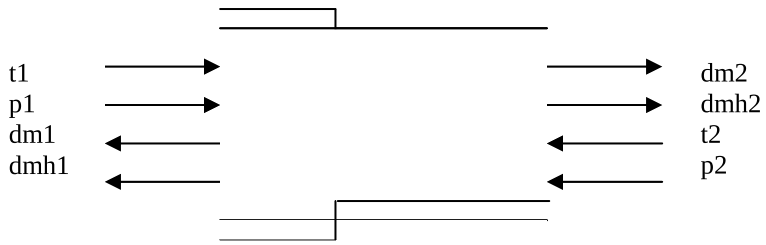

Figure 8. Expansion/contraction element.

Because the change in diameter a pressure drop will occur due to the friction which can be expressed as

$$\Delta p = k \cdot \frac{\rho \cdot Q^2}{2 \cdot A_{\min}} \tag{22}$$

where $A_{\min}$ is the minimum cross section area and k is the friction factor. This can be calculated as

$$k_{\exp} = \left(1 - \frac{A_1}{A_2}\right)^2 \tag{23}$$

for expansion and

$$k_{cont} = \left(1 - \frac{1}{0.6 + 0.37 \cdot \left(\frac{A_1}{A_2}\right)^3}\right)^2 \tag{24}$$

for contraction, where A_1 and A_2 are the areas at port1 and port2. [3]

2.2.5 Progressive Expansion/Contraction

Sometimes a change in diameter is done as a diffuser or convergent pipe. These are types of resistive elements and for that reason several variables can therefore be calculated, as done earlier in this chapter. One thing that is specific is the friction factor and these can be written as the following function [3]

$$k_{diff} = f(\alpha, \frac{diam1^2}{diam2^2}) \tag{25}$$

and

$$k_{\text{conv}} = f(\alpha, \frac{le}{diam}) \tag{26}$$

The parameters in the function are showed in Figure 9 and Figure 10.

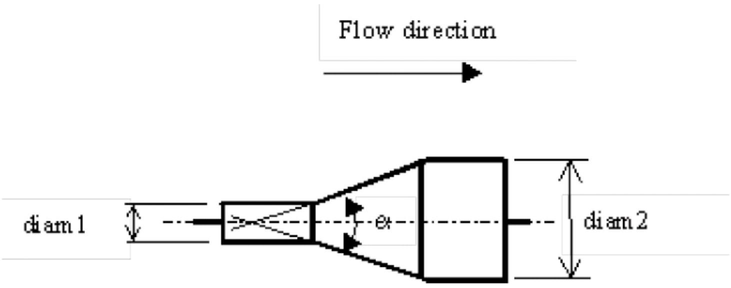

Figure 9. Diffuser.

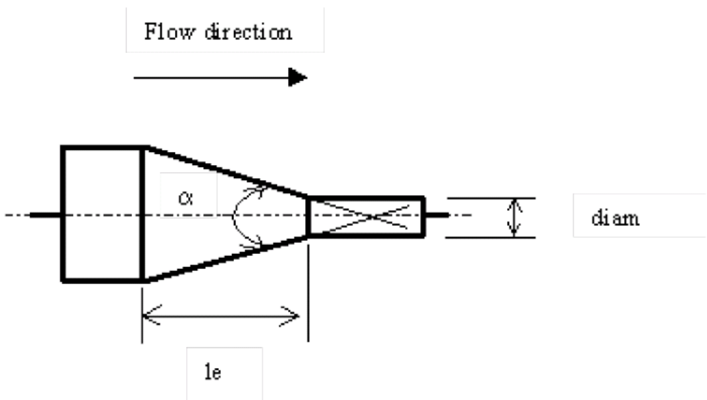

Figure 10. Convergent pipe.

2.3 Heat Transfer

In the coolant circuit there are several components that work as a heat exchanger. These components can be simple oil/coolant heat exchangers, radiators but also the coolant pump will deliver heat to the coolant. To calculate the heat that is transferred a number of equations is needed. To give a better understanding of the calculation for each component in the model the heat exchanger will be divided into two groups. The calculation for oil/coolant heat exchanger will be solved separate from the radiator and the coolant pump. In Figure 11 a general heat exchanger is showed that include basic nomenclature.

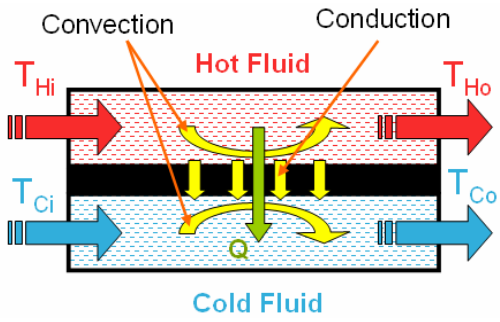

Figure 11. A general heat exchanger.

2.3.1 Heat Transfer in Radiator

Suppose that heat will be exchanged between air and a coolant with a wall that separates them from each other. Let T_c be the temperature of the coolant and T_a the temperature of the air. Let $T_c > T_a$, then the heat flow rate can be written as

$$\dot{Q} = \frac{T_b - T_a}{R} \tag{27}$$

where R is the overall thermal resistance. R can be composed of resistance due to convection at the walls and the conduction through the wall. These can be written as:

- A convective exchange between the hot fluid and the separation wall

$$R_{conv,h} = \frac{1}{h_1 \cdot A_1} \tag{28}$$

- A convective exchange between the separation wall and the cold fluid

$$R_{conv,c} = \frac{1}{h_2 \cdot A_2} \tag{29}$$

- A conductive exchange within the separation wall

$$R_{cond} = \frac{e}{k \cdot A_k} \tag{30}$$

These three can be seen as a series and R will therefore be

$$R = R_{cond} + R_{conv,c} + R_{conv,h} = \frac{e}{k \cdot A_k} + \frac{1}{h_2 \cdot A_2} + \frac{1}{h_1 \cdot A_1} \tag{31}$$

Instead of Equation 27 the following way will be convenient

$$\dot{Q} = U \cdot A \cdot \Delta T_m \tag{32}$$

where U is heat transfer coefficient, A is exchange area and ΔT_m is the average temperature difference. If Equation 27, Equation 31 and Equation 32 are combined the following equation will hold

$$U \cdot A = \frac{1}{R} = \frac{1}{\frac{e}{k \cdot A_k} + \frac{1}{h_2 \cdot A_2} + \frac{1}{h_1 \cdot A_1}} \tag{33}$$

These are not by nature always the same but the heat flux continuity gives

$$U \cdot A = U \cdot A_k = U \cdot A_1 = U \cdot A_2 \tag{34}$$

Here a dimensionless number will be presented, The Number of Transfer Units (NTU), which is widely used for heat exchanger analysis. It is defined as follows [1]

$$NTU = \frac{U \cdot A}{C_{\min}} \tag{35}$$

where $C_{\min}$ is defined as

$$C_{\min} = \min\left(\dot{m}_f \cdot Cp_f, \dot{m}_a \cdot Cp_a\right) \tag{36}$$

Furthermore the effectiveness for a heat exchanger is written as

$$\varepsilon = \frac{q_f}{q_{\max}} = \frac{C_f \cdot (T_{f,in} - T_{f,out})}{C_{\min} \cdot (T_{f,in} - T_{a,in})} \tag{37}$$

A relationship between ε and NTU can than be written as

$$\varepsilon = 1 - e^{-NTU} \tag{38}$$

which gives the following equation

$$U \cdot A = \dot{m}_f \cdot Cp_f \cdot \ln\left(\frac{1}{1-\varepsilon_f}\right) \tag{39}$$

In the model U will be calculated as a semi empirical equation with five coefficients

$$U = \frac{1}{\frac{1}{k_m} + \frac{1}{a_{air} \cdot G_a^{bair}} + \frac{1}{a_{af} \cdot G_a^{bf}}} \tag{40}$$

The coefficients k_m, a_{air}, b_{air}, a_f and b_f will be determined by regression from the heat exchanger performance measurements using an excel sheet supported by AMESim. [4]

2.3.2 Oil/Coolant Heat Exchanger

The heat exchanger outlet temperature is computed from its derivative with respect to time as shown below

$$\frac{dT}{dt} = f(\varphi_3, dm_1, dm_2) \tag{41}$$

where φ_3 is the heat flow rate, dm_1 and dm_2 are the enthalpy flow rates at inlet and outlet. The heat transfer for a heat exchanger can be calculated as

$$\dot{Q} = k \cdot A \cdot \overline{\vartheta} \tag{42}$$

where k is the convection coefficient, A is the area in the exchanger where heat transfer is done and $\overline{\vartheta}$ is the average temperature difference. The heat transferred to the cold fluid

$$\dot{Q}_h = \dot{m}_h \cdot c_{cp,h} \cdot \Delta T_h \tag{43}$$

and the heat transfer from the hot fluid

$$\dot{Q}_c = \dot{m}_h \cdot c_{cp,c} \cdot \Delta T_c \tag{44}$$

should be the same where $\dot{m}$ denotes the mass flow of the fluid and ΔT is the temperature difference between inlet and outlet for a fluid. [6]

There are several of principles of a heat exchanger design. Fundamentally three principles can be distinctive: counter flow, parallel and cross flow. The calculation of $\overline{\vartheta}$ will be different for each of these three principles. Each one will be illustrated below with the equation for the convection coefficient. [6]

- Counter flow heat exchanger

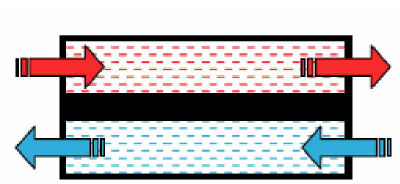

Figure 12. Counter flow heat exchanger.

$$\overline{\vartheta} = \frac{\vartheta_1 - \vartheta_2}{\ln \frac{\vartheta_1}{\vartheta_2}} \tag{45}$$

- Parallel flow heat exchanger

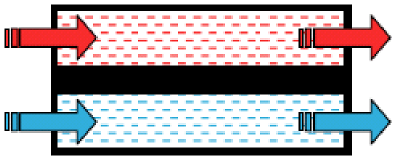

Figure 13. Parallel flow heat exchanger.

$$\overline{\vartheta} = \frac{\Theta - \vartheta}{\ln \frac{\Theta}{\vartheta}} \tag{46}$$

- Cross flow heat exchanger

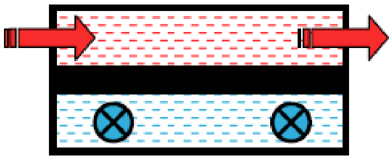

Figure 14. Cross flow heat exchanger.

It is possible to calculate the average temperature differences for a cross flow but it can be very time demanding and therefore diagrams could be used.

2.3.3 Special Heat Exchange Calculation

Sometimes the heat transfer calculations are difficult to perform for a heat exchanger because of the geometry. Maybe the area is difficult to calculate or there can be problems to get a good value for the convection coefficient. For these reasons in practical engineering measured values for the heat exchanger can be used to calculate the heat transfer. Let $T_{h,exp}$ and $T_{c,exp}$ denote the experimental inlet temperatures, which are constant in the experiment, of the heat exchanger for the hot and the cold fluid. Then the experimental temperature difference can be defined as

$$\Delta T_{\exp} = T_{h,\exp} - T_{c,\exp} \tag{47}$$

The real heat exchange can thereafter be calculated as

$$\dot{Q} = \dot{Q}_{\exp} \frac{T_{h,real} - T_{c,real}}{\Delta T_{\exp}} \tag{48}$$

where $\dot{Q}_{\exp}$ is the experimental heat exchanged for a certain mass flow $\dot{m}_h$ and $\dot{m}_c$. [10]

2.3.4 Heat Transfer in Coolant Pump

When the coolant pump works energy is transferred to the coolant as heat. The coolant temperature at the pump outlet is computed from the temperature at the pump inlet and the heat provided by the pump to the coolant. In an ideal pump, with an incompressible fluid, this energy can be written as follows

$$\dot{Q} = \frac{\dot{V}\Delta p}{n_{eff}} \tag{49}$$

where $\dot{V}$ is the volumetric flow rate, Δp is the pressure differential over the pump and η_{eff} is the efficiency parameter of the pump.

2.3.5 Heat Transfer in Transmission Oil Cooler

When driving the truck there is often a need to change gear due to changing driving conditions. This can be done in different ways, both automatic and manual. Independent of gearbox solution there will always be a heat exchange into the transmission oil due to losses in the gearbox. This heat need to be cooled down and in theory there is three ways to do this, conduction, convection and radiation. Because of the small temperature different between the rear shaft and the gearbox the conduction will be negligible if the transmission and the shaft are warm enough. To know the other two, a combined equation for the convection and radiation has been derived from

$$\dot{Q}_{convection} = h \cdot A_s \left(T - T_\infty\right) \tag{50}$$

and

$$\dot{Q}_{radiation} = \varepsilon \cdot \sigma \cdot A_s \left(T^4 - T_\infty{}^4\right) \tag{51}$$

to

$$\dot{Q}_{conv+rad} = X_1 \cdot v^a \cdot \left(T - T_\infty\right) \tag{52}$$

because the exact value of A_s and h is not known. The amount of heat that is exchanged in the cooler is the different of total loss.

3 Input Data

To be able to build the model in AMESim and later validate the model different data sources and types is needed. These data will be presented in this chapter along with how they are used in the modelling

3.1 Data Overview

During this thesis project two sources of data have been used, data sheets and experimental data. The latter can be divided in nine tests, each one is unique with different design modification. These modifications can be a different pipe, placing of inlet and outlet or removing of a component. The experimental data has been used for both parameter settings and validation. In Figure 15 the different sets data are presented and then explained how they are used.

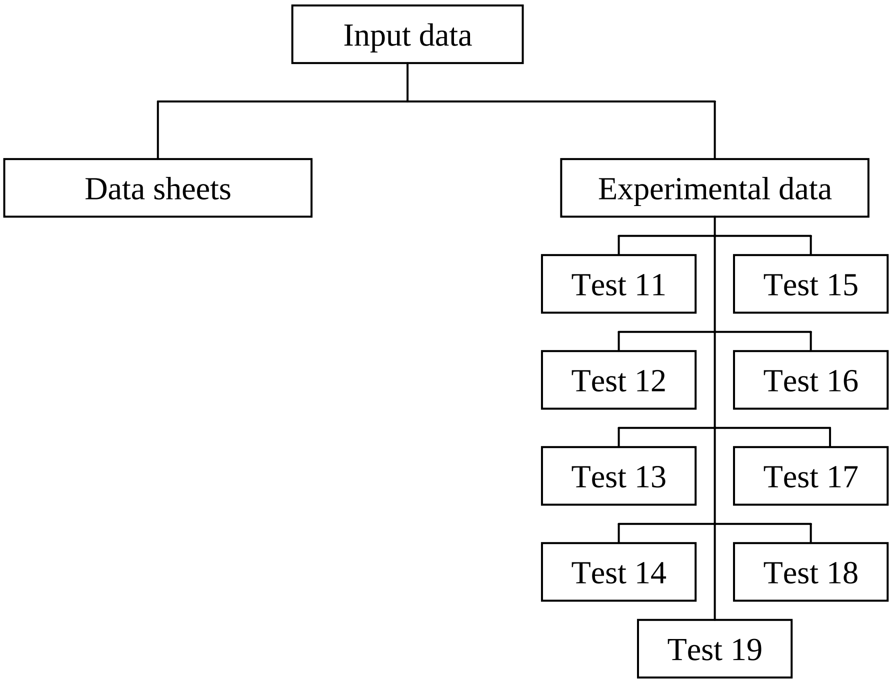

Figure 15. Input data.

The data sheets are used for building the model and Test 11 is used for parameter settings. Then Test 13 and Test 15 are used to validate the model. The other 6 tests are not used because they are slightly different from the other three. For this reason they will not contribute to the validation and are therefore not used.

3.2 Data Sheets

When building the model some components are based upon data sheets. These are used when it is very difficult to calculate the actual pressure drop, enthalpy transfer (heat) or changes in mass flow rate. Often these data sheets contain several points of measurement that could have different parameters such as mass flow rate or temperature. For a heat exchanger there will also be parameters for both the cold and hot side.

Normally the supplier of a component produces the data sheets but it can also be produced within Volvo. The data sheets are used in different places around Volvo and therefore have a high level of credibility.

3.3 Experimental Data

During the thesis work experimental data has been used for both validating the AMESim model and some parameter setting. These experimental data are as earlier explained divided in several tests, which are done separately at the test facility.

At Volvo 3P measurements are done continually on a full-scale truck in order to validate different design and concepts. The specific model done in AMESim has been chosen because it is a relevant measurement that has recently been done and it represents a common truck model. The actual truck is showed in Figure 16. The actual measurement has been done July 2007 at Volvo 3P facility in Gothenburg for project P2631 Euro 5. It is presented in an internal engineering report. [12]

Figure 16. FH31 MD13 EURO 5 520hp.

In the measurements a fully open thermostat has been used. Furthermore the coolant pump has been running at full and reduced speed. In the validation and parameter setting only full coolant pump speed has been considered and therefore only tests with reduced pump speed will be ignored. Totally nine test has been done all with different circuit designs. This report will just use three of the tests because they represent interesting situations and designs. The differences between the other test these and are small and they will therefore not contribute much to the validation. Test 11 is used for parameter settings and Test 13 and Test 15 are used for validation.

There are a number of points of measurement in the experimental data representing different physical quantities. The selected measure points have been chosen because the design has changed or they could give more information of special interest. Volume flow rate, pressure and temperature have been measured. Each test has been done when a certain coolant temperature was at 90 ^{o}C. Because of that a validation of temperature has not been done in this report. The measure points in the test were:

- Coolant flow through radiator
- Coolant flow through transmission oil cooler
- Coolant flow through air compressor
- Coolant flow through servo oil cooler
- Coolant flow through cab heater
- Coolant flow through urea heater
- Coolant pressure before coolant pump
- Coolant pressure after coolant pump
- Coolant pressure after oil cooler,
- Coolant pressure before transmission oil cooler
- Coolant pressure after transmission oil cooler
- Coolant pressure before radiator

From these measure points the total pressure drop over the cooling circuit can be calculated, which gives the system curve. Together they will give an opportunity to build and validate the AMESim model accuratly.

4 Building the Model

In this chapter an overview of specific cooling circuit for this thesis work is presented. Each component is then described and the solution in AMESim is showed.

4.1 Coolant Circuit

The design of a cooling circuit (system) can differ between truck models. Different components and different connections can be found. This report will focus at the circuit in Figure 17. The circuit is found in both FH and FM trucks with different engine sizes.

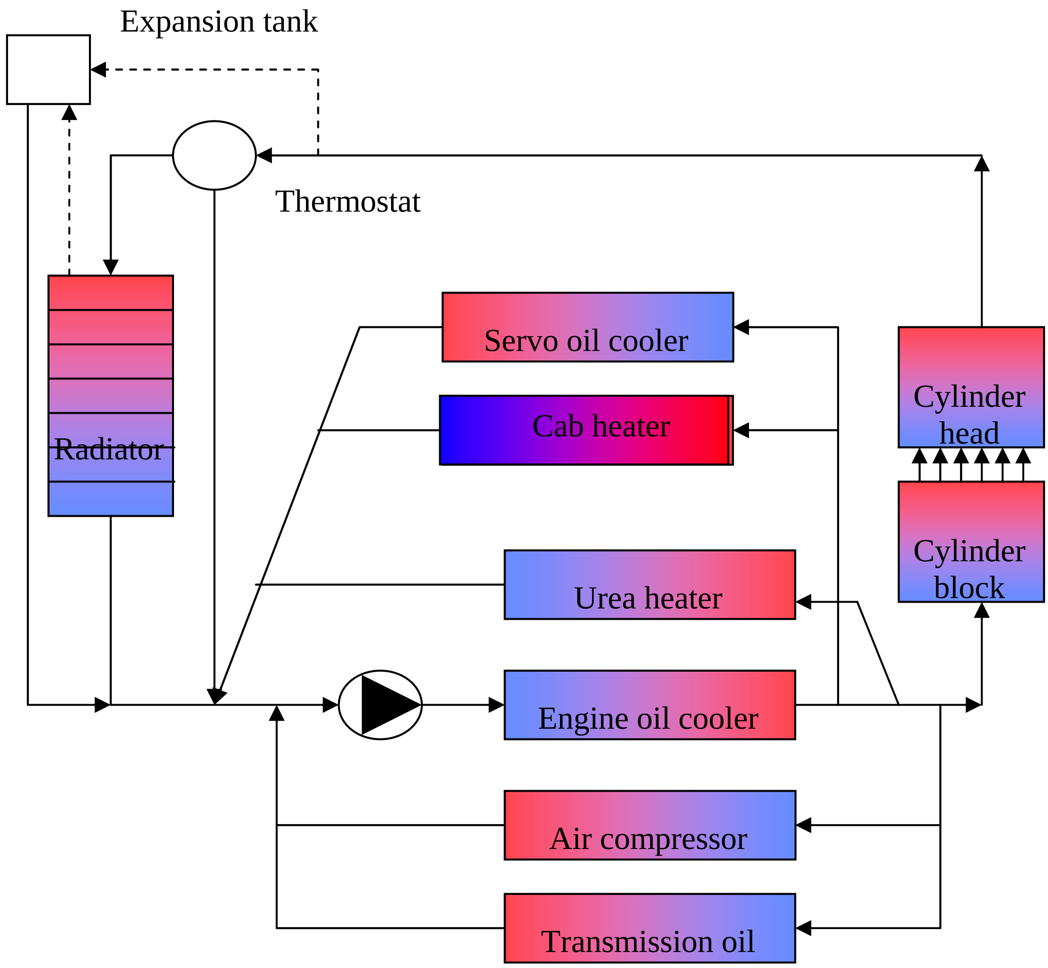

Figure 17. The coolant circuit.

When building a model in AMESim the user must be aware of the physics behind the components. Each component has a special task in the cooling system. For example these tasks can either be to transfer heat from a component to the coolant or give the coolant energy to be able to supply the circuit with the flow that is needed. Between the components there is often a system of pipe and hoses. If the diameters or length of these change the flow or pressure drop in that branch of the cooling system will be affected. Also the position of the inlet and outlet will effect flow and pressure drop. Therefore pipes and hoses can give the designer the possibility to calibrate the cooling, giving each component the flow that is specified for their individual needs.

Every component in the system change between different truck models depending on the area and market they will be used in. Because Volvo is a global company with a wide range of models and is present on all continents the combination of components will be big. In this report a truck used in Europe will be used which are called FH31 MD13 EURO5 520. The coolant circuit have the following components

- Coolant pump
- Thermostat
- Cab heater
- Urea heater
- Air compressor cooler
- Radiator
- Cylinder head and block
- Servo oil cooler
- Transmission oil cooler
- Engine oil cooler
- Expansion tank

Each one of these components will be described below in terms of function, task and how they are built in AMESim. After that examples of pipe simulation and realisation will be showed.

4.2 The Coolant Pump

The coolant pump is the device that procures the necessary energy to the coolant to make it flow through all the components and overcome the total pressure drop exerted by the circuit.
It is crucial to give the coolant pump a minimum of flow that can guarantee right cooling capacity as stated in previous chapter.

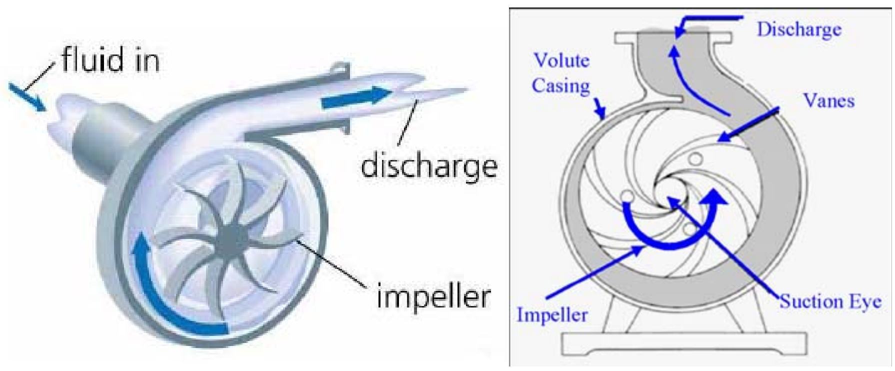

Figure 18. Centrifugal pump.

In this model the cooling pump is a motor driven pump with a certain ratio to the engine speed. This is can be seen in Figure 18. To increase the pressure of the fluid the centrifugal pump is using a rotating impeller.

The purpose of the centrifugal pump is to convert energy of the engine, first into velocity or kinetic energy and then into pressure energy of a coolant that is being pumped. The energy changes occur by virtue of two main parts of the pump, the impeller and the volute or diffuser. The impeller is the rotating part that converts driver energy into the kinetic energy. The volute or diffuser is the stationary part that converts the kinetic energy into pressure energy. This action is described by Bernoulli's principle. The amount of energy given to the coolant is proportional to the coolant velocity at the edge or vane tip of the impeller. When the impeller revolves faster or the bigger the impeller is, the higher the velocity of the coolant at the vane tip and the greater the energy imparted to the coolant will be. The kinetic energy of the coolant coming out of an impeller is harnessed by creating a resistance to the flow. The resistance is created by the pump when the fluid hits the casing and slowing down. In the discharge nozzle, the coolant further decelerates and its velocity is converted to pressure according to Bernoulli’s principle. [7]

The energy that is transfer to the coolant will convert into pressure and kinetic energy. It means that the pump makes the coolant flowing through the circuit with a certain flow rate and pressure. The pump characteristic curve defines the balance between pressure and flow rate.

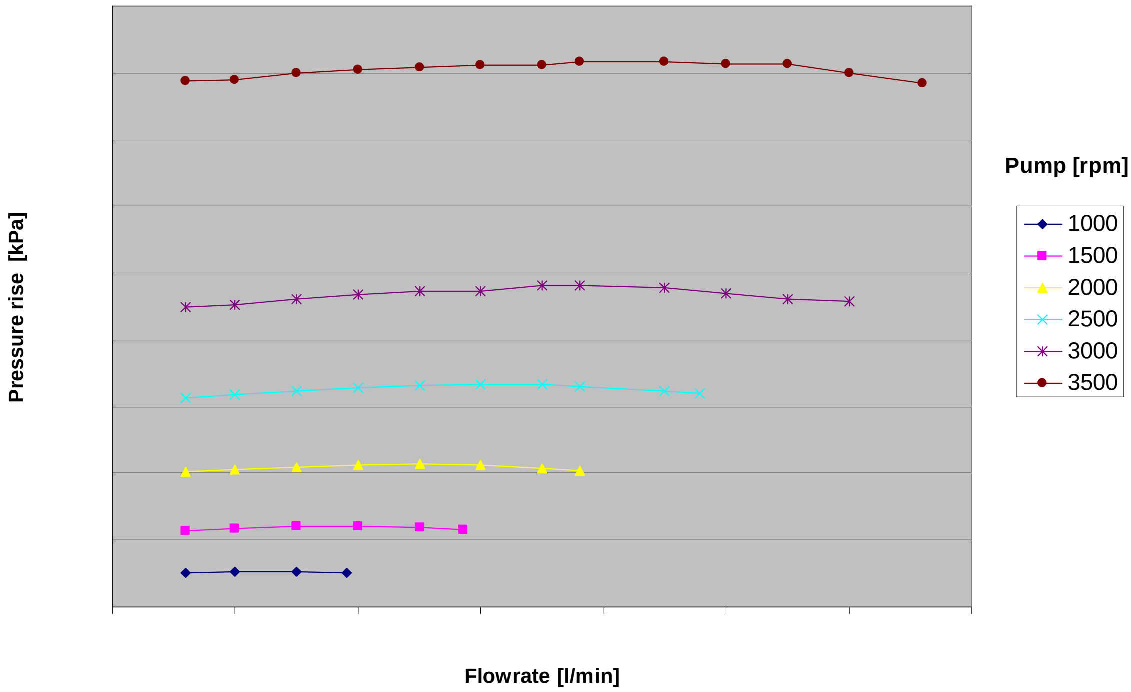

Figure 19. Coolant pump characteristic curve for a MD13 EURO5.

As mention in Chapter 1 each unique coolant circuit will result in a pressure drop curve, also called system curve. If Figure 3 and Figure 19 is combined the Figure 20 will occur (here is *Op1* and *Op2* two operating points).

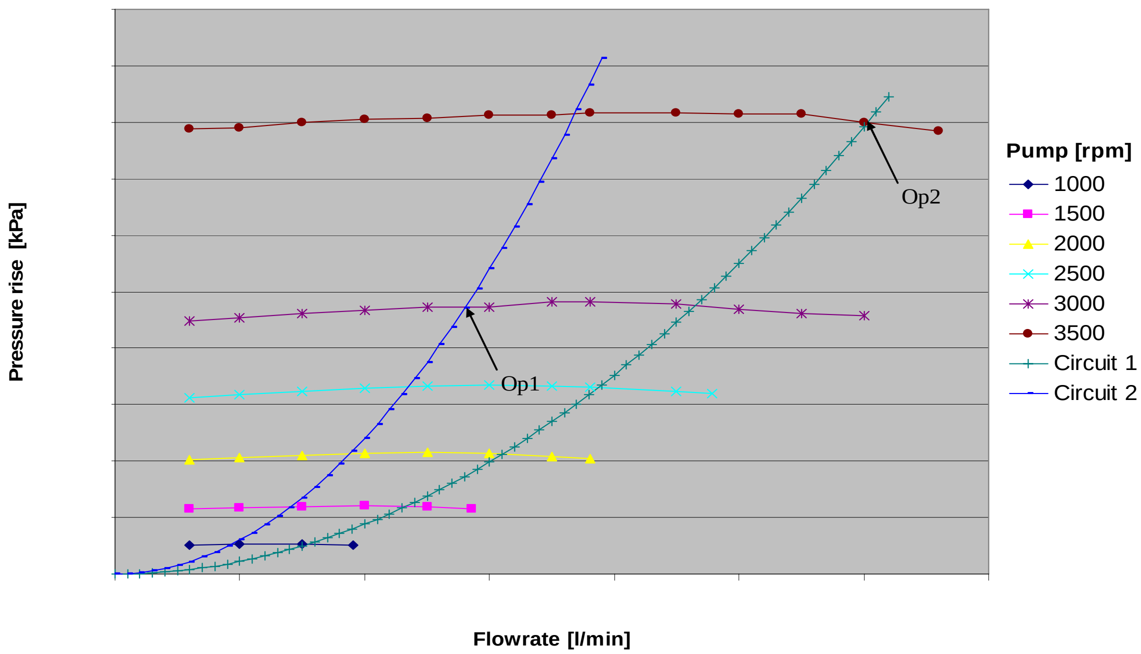

Figure 20. Coolant pump characteristic curve with system curves.

The coolant pump built in AMESim is showed in Figure 21.

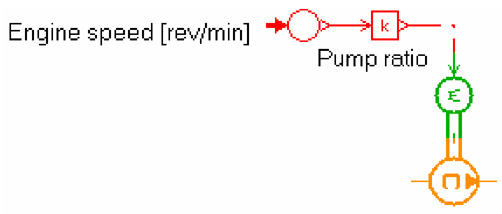

Figure 21. Coolant pump in AMESim.

Engine speed is a global parameter input which can change during the simulation if for example the purpose is to drive a certain road or under certain conditions. A ratio is also set with a constant value for each truck and configuration. Then a variable conversion is done by converting a mathematical signal into an angular velocity. The actual pump is showed in orange and has the previous characteristic curve as input. The pump will therefore deliver a certain pressure rise depending one the engine speed. There will also be small energy transfer, which can not be neglected.

4.3 Thermostat

The purpose of having a thermostat is to be able to split the flow from the radiator. A design goal is to not give away more heat from the cooling system than necessary. The main strategy to transfer heat from the system is to use the radiator, which will transfer heat from the coolant to the ambient air. The task of the thermostat is to not let more coolant go through the radiator than necessary. A wax body will expand or shrink due to the wax temperature, which is closely related to the coolant temperature. The size of the wax will then control the opening to the radiator and the by-pass. The fraction opening will follow the curves illustrated in Figure 22, one for increasing and one for decreasing wax temperature. As Figure 23 shows all coolant will go through the radiator and not in the by pass when the thermostat is full open.

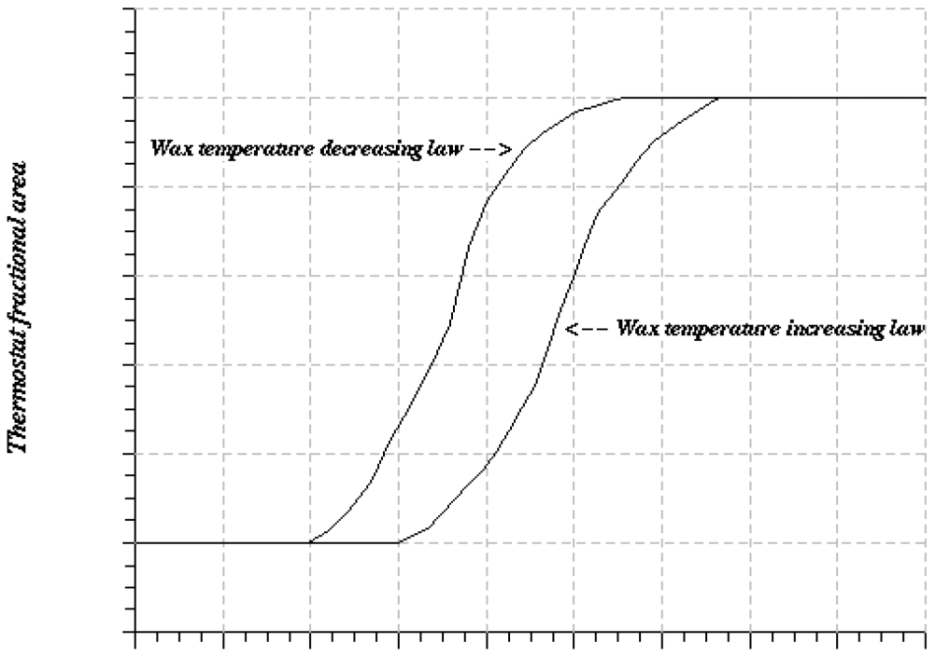

Figure 22. Characteristics curves for the thermostat.

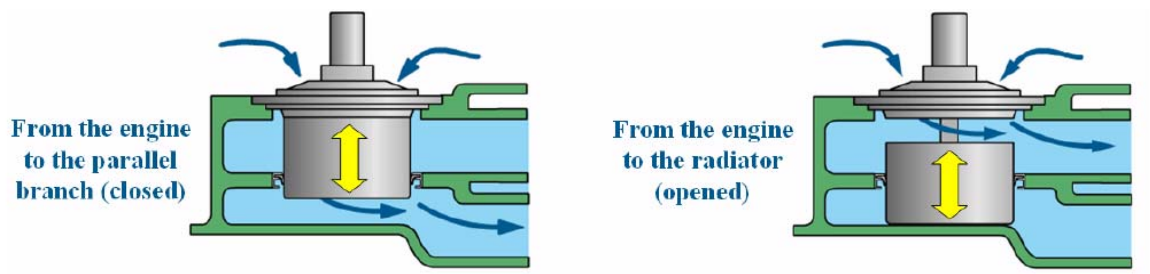

Figure 23. Closed and fully open thermostat.

The thermostat modelled in AMESim is presented in Figure 24 and as a supercomponent in Figure 25. In both ways the fraction of opening is controlled by a first order lag and with a relay with hysteresis. The user must set the time constant, gain and the numerical hysteresis.

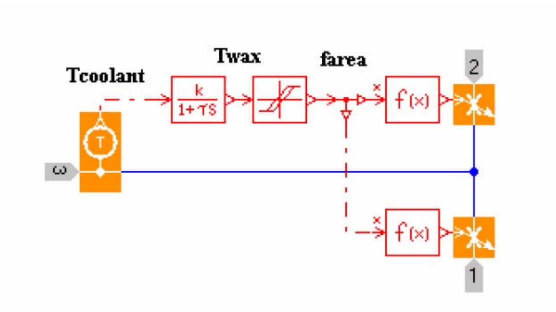

Figure 24 Thermostat.

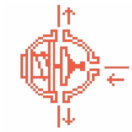

Figure 25 Thermostat as a supercomponent.

4.4 Cab Heater

In the coolant circuit there are some components that will not transfer heat to the coolant but rather need the heat at certain times. The cab heater is one component that needs as much heat as possible, for example at a cold start or when the truck is not moving. This is sometimes not easy to do because the engine could be cold and the ambient temperature could be under zero ^{o}C. A solution for this problem is to pre heat the engine and the cab with a device called parking heater, also called phenca. The device works like a small combustion engine with a combustion chamber, a fuel and air supply system. After the combustion the burned gases will be transported away in special pipes. The heat is transferred to the coolant that will be transported around the cooling system supported by a small water pump. This pump is mounted onto the phenca, and could be turned of when it is not needed any more. This extra parking heater system can be turned on in advance by a timer, giving the system time to prepare the truck. The reason to warm up the engine is lower emission levels due to better working points.

A pressure valve and a temperature valve also control the coolant flow to the cab heater. The pressure valve guarantees a low pressure at the cab heater in order to protect it and extend the component lifetime. When the pressure difference is too large a by pass will be opened which releases pressure by letting some coolant passing direct to the coolant pump. Both the ambient temperature and the temperature in the cab govern the temperature valve. The cab heater system is showed in Figure 26.

The system can be divided into four parts, a heat exchanger, valves, phenca and additional pipes and hoses. Each one will result in a pressure drop, which will lower the flow through the circuit. In the basic AMESim model the valves are fully open and will therefore not affect the flow. In later models the valves will be included in the situations where they make a difference.

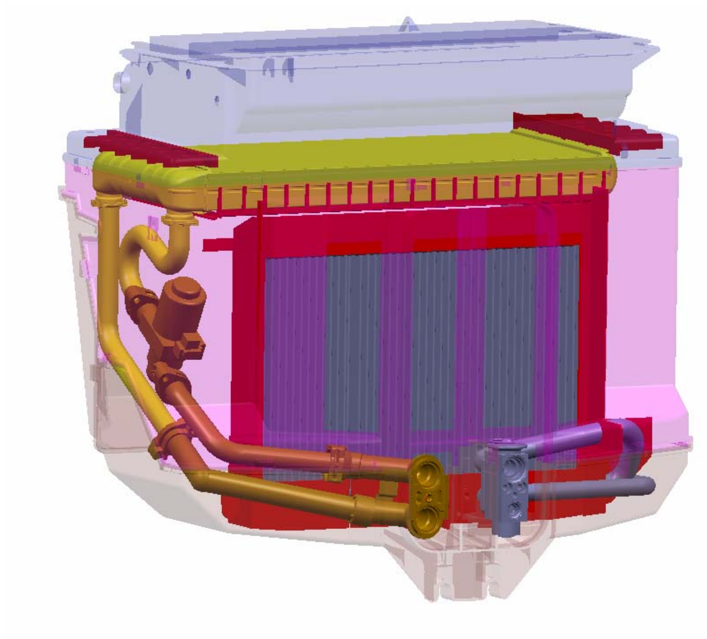

Figure 26. The cab heating system.

4.5 Urea Heater

Another component that needs a lot of heat when the ambient temperature is low is the urea heater. The main task of this component is to heat up the urea to the temperature where its effect is optimised. Urea is a solution to lower the NOx emission, which has a bad impact on the environment. Therefore governments in Europe, North America and Japan have legislated limits for emission that the truck and car manufactures need to manage. One of these emissions is NOx and to lower the emission urea can be used. Urea it self is an organic compound with the chemical formula (NH2)CO, which is a type of ammonia. The urea is a part of the SCR (Selective catalytic reduction).

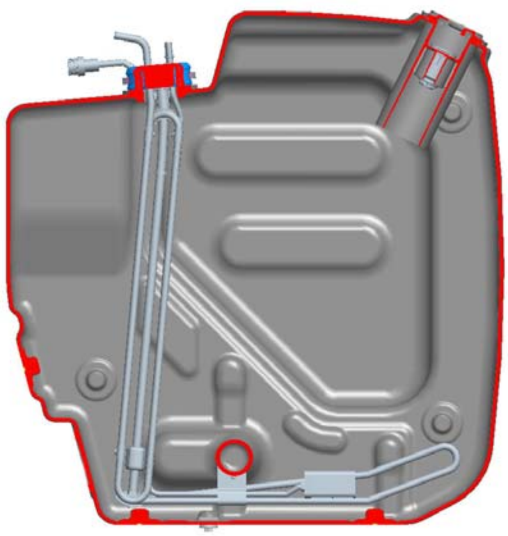

Figure 27. The urea heater and the tank.

The principle of the urea heater is to lead the coolant in several pipes that are inside the urea tank. The warmer coolant will therefore exchange heat with the urea and its temperature rise. When the temperature in the urea tank has reach a threshold, an electrical valve is closed and the coolant can not reach the urea heater. In the AMESim model the electrical valve is ignored because the coolant can pass freely through the urea heater. A valve can later be included in the model to give the behaviour. In AMESim the urea heater is designed and built as a long pipe with bends, measured after the real component and it is showed in Figure 28. The heat transfer is done in one of the pipes and this can be seen as an aggregating of all the pipes.

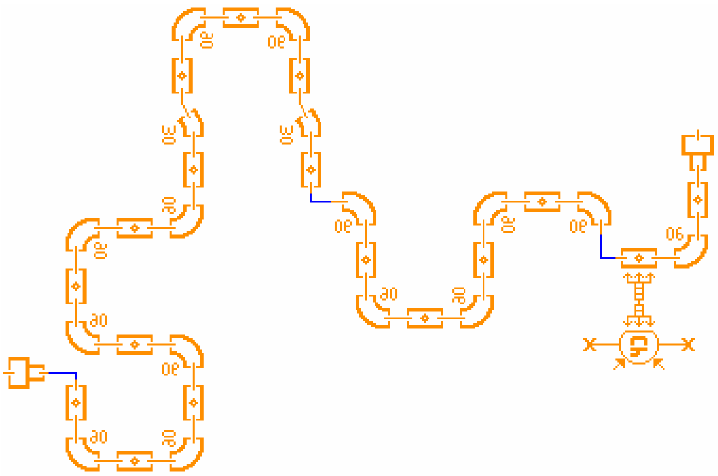

Figure 28. The AMESim model of urea heater.

4.6 Air Compressor Cooler

To generate air pressure for braking, suspension system and auxiliary equipment an air compressor is used. The compressors task is therefore to provide the vehicle with clean compressed air. If the air is not clean enough components can brake down and a failure can occur. The speed of the air compressor is related to the engine speed with a certain ratio. The component is showed in Figure 29.

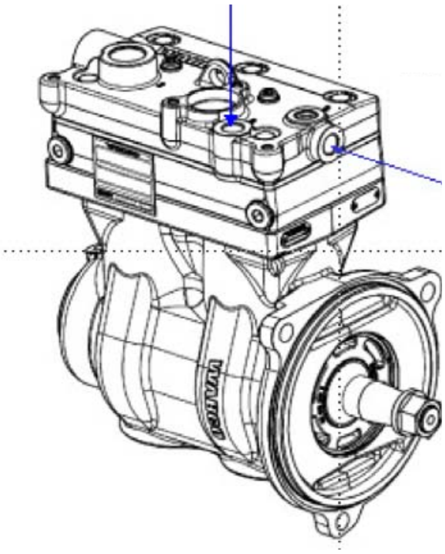

Figure 29. Two cylinders air compressor.

When a gas is compressed the heat and temperature will rise and cooling of the compressor is needed. This is done by letting coolant flow over under around the goods, which lead to a heat transfer from the air compressor to the coolant. The amount of heat needed to be transfer from the air compressor depends on the pressure in the compressed air. Higher pressure means a larger amount of heat to the coolant. In Figure 30 the air compressor cooler modelled in AMESim is presented.

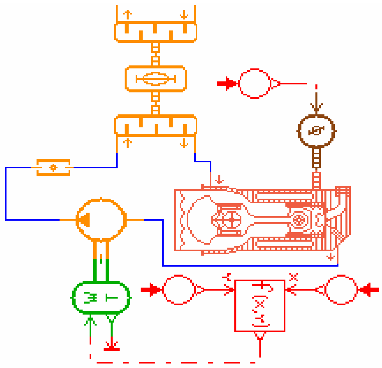

Figure 30. AMESim model of air compressor.

4.7 Radiator

Under a normal long drive most of the components will transfer heat into the coolant circuit and the radiator exchange heat with the ambient air. To be able to realise the amount of heat into the air the geometry of the radiator will change between different types of trucks. Also the grid and cab design will affect the cooling capacity of the radiator.

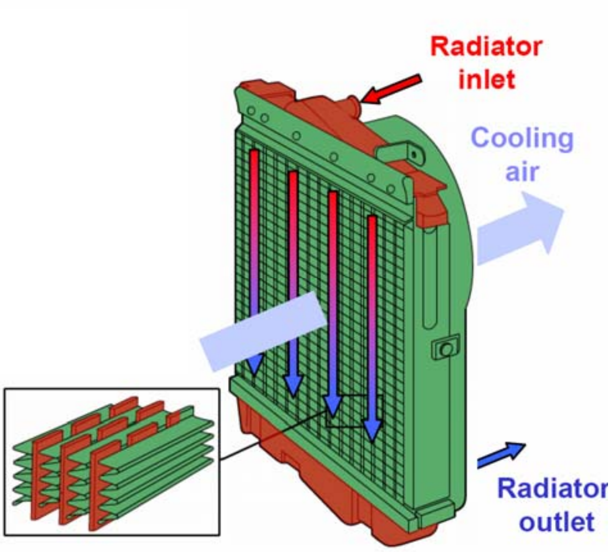

Figure 31. Principal sketch of a radiator.

The radiator used in the model is a cross-flow heat exchanger where the coolant flows from the top tank to the bottom tank like a waterfall. The air will pass the radiator horizontal as showed in Figure 31. The flow through the radiator can in AMESim be different over the front area that enables the user to change the flow due to different cab and grid designs. The heat exchange between the coolant and the ambient air is modelled from the input data collected from the supplier. Thereafter the heat flow rate in the model is calculated with a NTU-based correlation using the five coefficients k_m, a_{air}, b_{air}, a_f and b_f determined by regression. This calculation is presented in Chapter 2.

Normally there will also be a fan behind the radiator. This is not included in the model because it was not used during the measurement in the Volvo 3P facility. The fan can quite easily be inserted into the model and simulated.

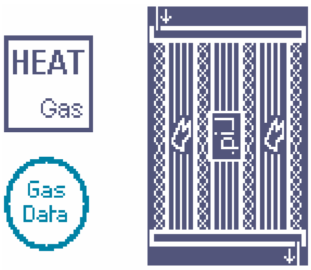

Figure 32. The radiator in AMESim.

4.8 Cylinder Head and Block

When the coolant has passed the oil cooler it will go round the cylinder block and over the cylinder head in order to cool the engine. A picture of this flow is presented in Appendix 3. To build a model and simulate this is very difficult which impels to make a rough approximation. By research done at Volvo Powertrain the amount of heat going from the engine to the coolant is known. Because of these a heat function with two variable, load and engine speed, can be used. By using data from this function in the model a accurate result will be achieved. Because of this the model is very simple and can be seen in Figure 33.

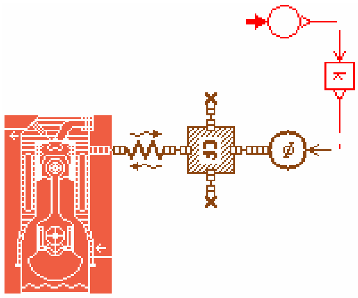

Figure 33. Engine in AMESim.

The engine has also a mass that is included in the model, which gives the model a time delay. The pressure losses in the cylinder head and block are also very complex and the model is therefore built with supplier data as well and will be interpolated from a table of pressure loss and flow rate.

4.9 General Heat Exchanger Oil/Coolant

There are a number of components in the coolant circuit that have oil that works as a friction damper and coolant fluid. In order to transfer heat from the oil to the coolant circuit a liquid/liquid heat exchanger. In the actual coolant circuit there are three such heat exchangers, engine oil cooler, transmission oil cooler and servo oil cooler. The first differ much from the two others in design because the coolant will pass through cooler rather than inside. This is not a problem for the model because these two designs will have the same look. A heat exchanger is presented in Figure 34.

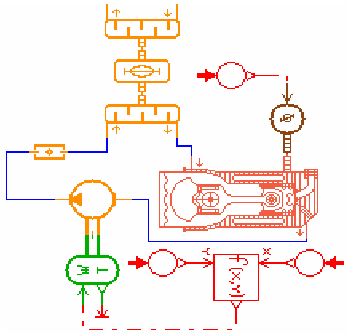

Figure 34. AMESim model of general heat exchanger.

This solution is possible because of the complexity in pressure drop over the component and the heat transfer. Due to this the model calculates the transferred heat from supplier data both for pressure and heat. The heat can also be calculated if the mass flow rate is known and the temperature differences between inlet and outlet. How this is done is in detail described in Chapter 2. Each one of the heat exchangers will be explain below.

4.9.1 Engine Oil Cooler

The engine cooler is placed after the coolant pump and is mounted under the oil cover. The main task for the engine oil is essentially the lubrication of the turning parts. It will therefore flow around the engine lubricating the moving and rolling parts. There is a gear pump, driven by the distribution, which permits the oil to flow correctly through the oil circuit. During the circulation friction between different parts will be created and it means a higher temperature in the oil and the engine during the circulation. At the same time the oil has contact with hot parts which will increase the oil temperature even more. To ensure a correct behaviour, efficiency and lifetime of the engine, the oil need to cool down and this is done by the heat exchanger. The location of the heat exchanger can be seen in Figure 35.

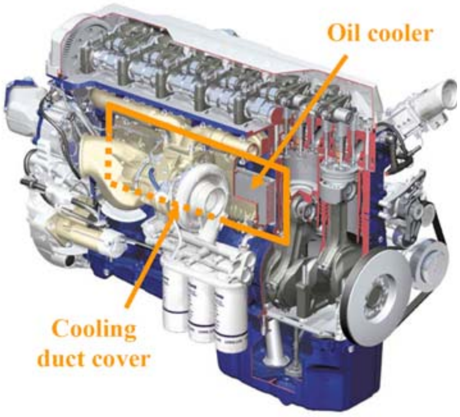

Figure 35. Oil cooling position.

4.9.2 Transmission Oil Cooler

One of the main parts in a truck is the gearbox mounted behind the engine. It permits to transfer the torque delivered by the engine to the propeller shaft with a suitable ratio for the driving conditions. The gearbox in the model is I-shift gearbox with 12 gears, which give the driver a powerful tool that can help him in his work.

In the gearbox a lubrication system is used to enhance the friction coefficient between the gears. The friction leads to that energy from the oil needs to exchange while it is in contact with all these turning parts. To transfer the heat further a transmission oil cooler is mounted on the exterior side of the gearbox. A small gear pump allows the oil to flow through the cooler. The oil temperature increases significantly only during special driving conditions with high load, for example climbing a hill, and in most cases the oil temperature just follows the coolant temperature. A gearbox is showed in Figure 36.

Figure 36. Gearbox.

4.9.3 Servo Oil Cooler

When steering a truck, energy is transferred to the servo oil. This will then exchange to the coolant in a heat exchanger. The temperature in the servo oil could be different depending where and how the truck is used. In some markets the component may be more crucial than others, for example rough terrain.

4.10 Building a Oil Circuit

Building an oil circuit has been done by a using a mathematical function with a number of inputs, depending of which component it is, and the output specified in an ASCII data file. In AMESim this function is illustrated in Figure 37.

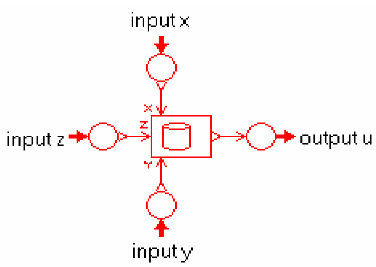

Figure 37. Input and output from an ASCII data file.

When creating an oil circuit the speed of the oil must be set. By using a pump with variable angular velocity ω the speed of the oil can be set. This depends on how the vehicle is being driven. The amount of transferred heat into the oil circuit from the surrounding parts is simulated as an engine that is leaving heat to the oil. The last part of the oil circuit is the actual heat exchanger that has been described in Chapter 2. These solutions can be seen in Figure 38 to 40. The reason modelling oil circuits is that the heat transfer to the coolant from the oil will depend on oil flow and oil temperature. Because of this it is vital to have a rough model of the oil circuit.

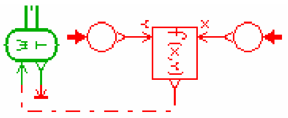

Figure 38. Flow speed setting.

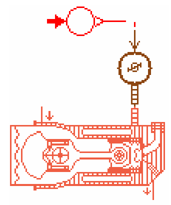

Figure 39. Heat insert.

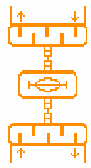

Figure 40. Heat exchanger.

5 Parameter Settings and Validation

It is important to validate a model to know if corresponds well with the cooling system. This chapter will in describe the result of parameter settings and validation of the model. Three test has been used for this purpose, Test 11, Test 13 and Test15. Each test has it own layout, which is showed in Appendix 2

5.1 General Discussion

The model built in AMESim that is described in Chapter 4 needs to be validated to show that it models the cooling system well. The model in this thesis work is validated for engine speeds between 1000 – 2000 rpm and outside this interval it not valid. The model will therefore only be used in this interval and for other application and situation the model may not be able to give a rightful response and result. For these cases another model may be used or the model from this thesis work must be further developed.

The differences between the model and the cooling system can be as high as 10% without the model loosing its reliability. The reason for this is that the values from experimental data differ due to uncertain measuring methods, wrong chosen sensors and approximation in the model. The measuring methods have been questioned because the results from different experiments gives different values for the pressure. The experimental data can be considered better than others because it is the newest. Some sensors introduce a large fault in the data because they are designed to give an accurate result at a certain interval. Outside this interval they could give a value that could not or should not be used because of the uncertainty. This happens mostly for the servo oil cooler and other components that have a low flow. In the tables this is showed by "------". The approximations of the data can also be sources for the differences between the model and the cooling system. Approximation is mostly done in the heat exchangers but also in the pipes. It can come from interpolation because of lack of data but also material parameters and constants. Finally it is possible to say that data for the volume flow rate are better than the pressure data.

The result of the discussion above is that the volume flow rates is the main targets because these have a higher reliability than the pressure. Therefore the model could be good even if the differences in pressure are high.

At first the model need to have some parameters set. This has been done with experimental data from Test 11 introduced in Chapter 3. The method used for the parameter setting is the Method of Least Square. This will give the parameter values if

$$A \cdot X = B \tag{53}$$

is used. Here X is the parameter column and A and B is known matrices from several measurements. The method and the process for getting these values will not be explained in details in this report. By doing a validation it is possible to see if the parameter values give a satisfying result.

For validation of the model two other tests from the experimental data has been used. These can has early stated give good validation of the model because they represent different design and the other tests from the experimental data can be seen as special cases of theses two.

In this chapter the result of the AMESim model is presented together with the experimental data. Each one will be presented in curves where the volume flow rate or pressure is plotted against engine speed. The engine speed is between 1000 rpm and 2000 rpm. As explained before engine speed outside this interval is not interesting for this model because of too many uncertainties. At first the parameter settings are done using values from Test 11 and thereafter are the validation done with values from Test 13 and Test 15.

5.2 Parameter Settings, Test 11

When building the model most of the components have some type of data sheets and additional physical characteristics. This has been considered enough in a model point of view and therefore no tuning or extra parameter setting have been done. But sometimes lack of input data has been a problem and to be able to build a model, parameter setting has been done. For the model in this thesis work, parameter settings has been done only for volume flow and not pressure because of uncertainty in the experimental data.

One of components that had been built using the parameter settings is the urea heater. The result of the parameters setting is presented in Figure 41.

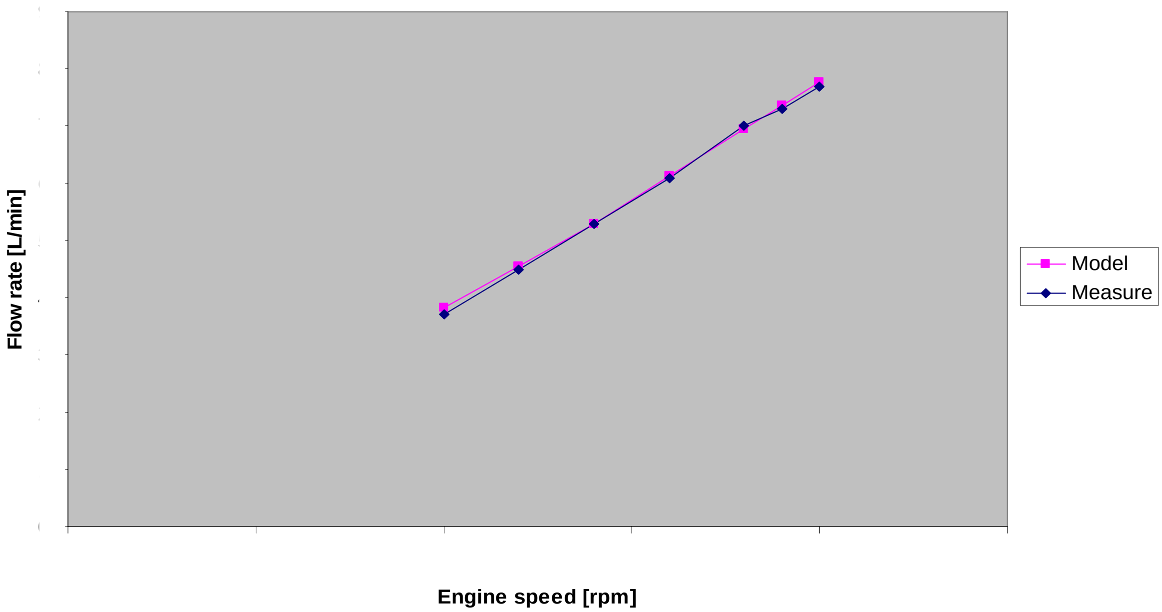

Figure 41. Flow through urea heater, Test 11.

5.3 Validation, Test 13

After the parameter settings the validation is done using values from Test 13. To see if model correspond with the cooling system, the system curve is compared with the system curve from the experimental data in Figure 42.

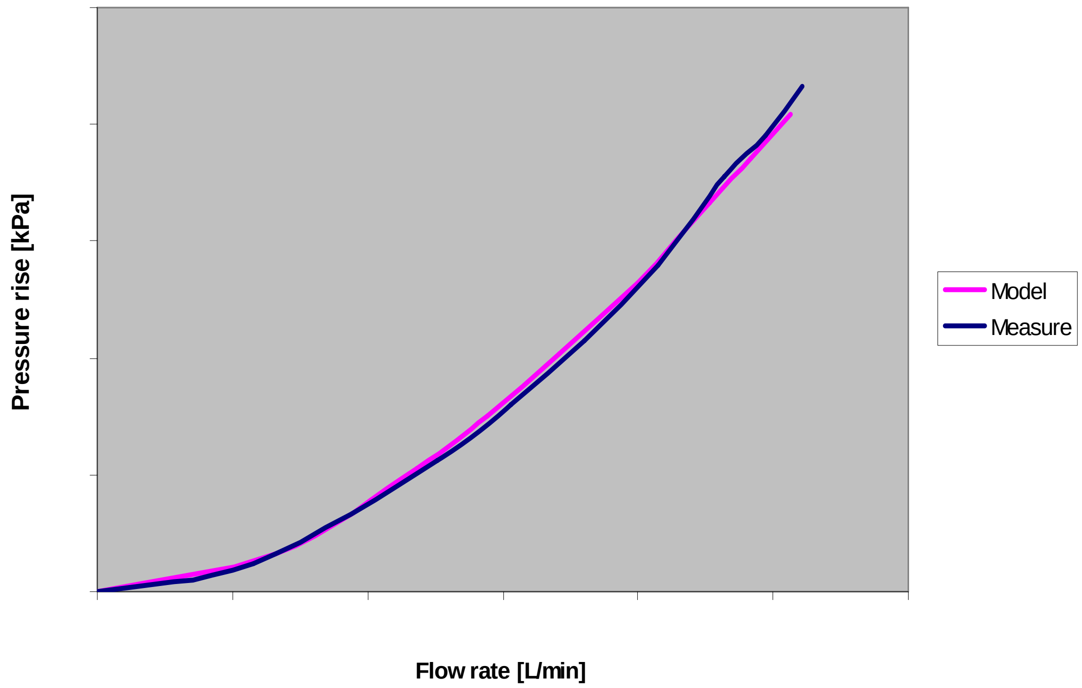

Figure 42. System curve, Test 13.

The difference between the curves is very small, under 2%. This shows that the coolant pump in the AMESim model gives nearly the same pressure rise as the real coolant pump at a specific flow. Each flow over a component is furthermore validated separately from the others. The results of these flow comparisons are displayed in Table 3 below.

	2000	1900	1800	1600	1400	1200	1000
Urea	0,10%	1,42%	0,03%	1,04%	0,61%	1,28%	5,82%
Cab heater	-0,48%	-0,80%	-0,60%	-1,25%	-2,26%	-2,53%	-3,28%
Servo	20,21%	28,54%	23,44%	24,64%	------------	------------	------------
Radiator	-2,81%	-3,72%	-3,70%	-5,42%	-5,46%	-6,82%	-7,18%
Transmission	4,40%	5,51%	5,28%	3,63%	1,81%	1,69%	3,12%
Air compressor	-1,91%	-2,35%	-2,57%	-3,07%	-3,43%	-1,90%	-0,15%
Coolant pump	-1,63%	-2,28%	-2,31%	-3,89%	-5,28%	-5,86%	-6,02%

Table 3. Differences between model and measured data in %, Test 13.

As seen in Table 3 the flow over the servo oil cooler differs a lot from the experimental data. This can be a result of the small flow through the component that makes it hard to measure. Except this the table showed that the largest difference is 7.18 %, which is ok for these types of models. Some part of the difference can be derived from difficulties in measurement and other from approximation in the model. To visualise these results two curves are presented in Figure 43 and Figure 44. The other curves are presented in Appendix1.

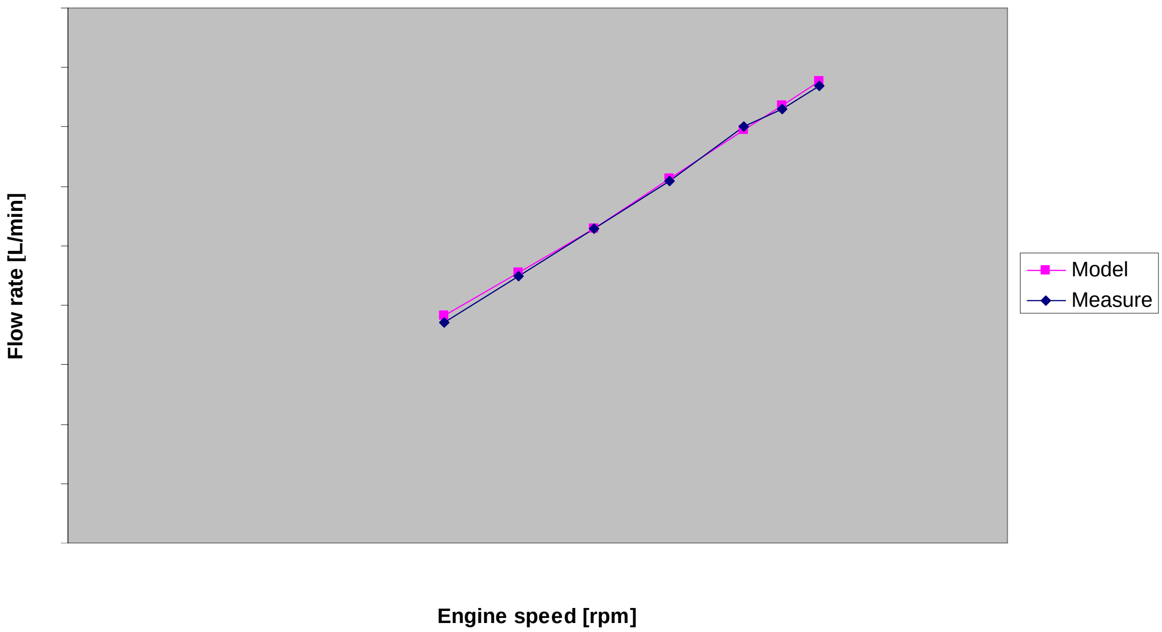

Figure 43. Flow through urea heater, Test 13.

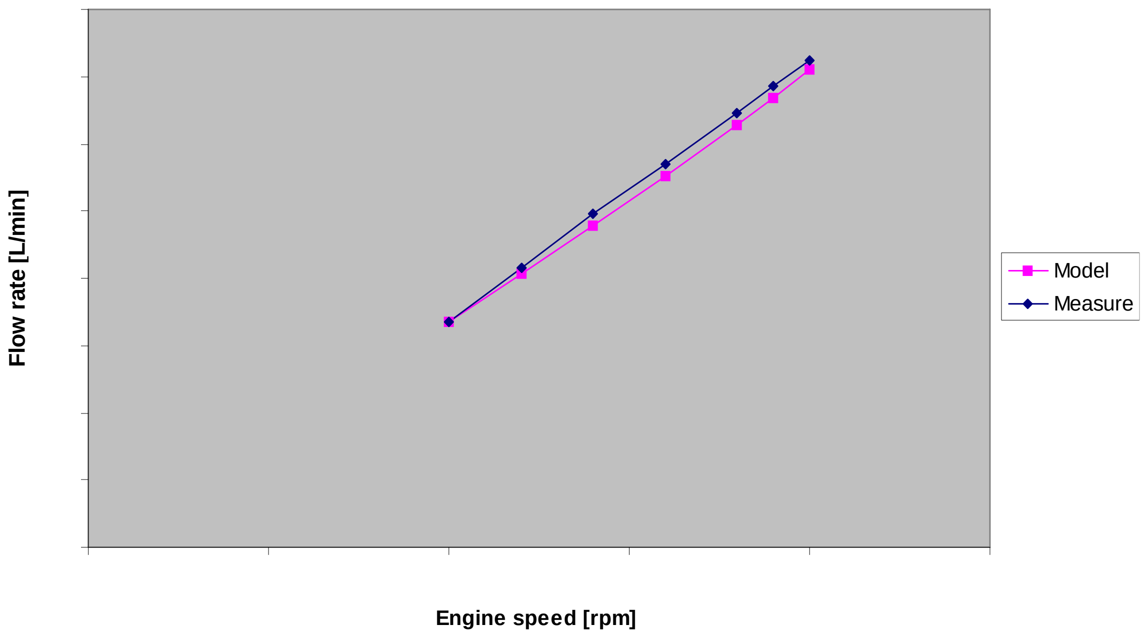

Figure 44. Flow air compressor cooler, Test 13.

The curves in Figure 43 and Figure 44 shows that the model built in AMESim correspond well with the measurement from experimental data. Together with the curve in Appendix 1 and Table 3 it shows that flow is predicted well over the interval [1000 2000] except for the flow through the servo oil cooler.

The problem with the measurement of pressure in the experimental data can be seen in Figure 45 and Figure 46. The curves from the measurement have unexplained dips over the interval. This can come from different sources as stated in Chapter 5.1, for example problem with measurement methods. For this reason the pressure validation have been difficult to do. But in order to visualise the problem the pressure before the transmission and coolant pump in Figure 45 and Figure 46.

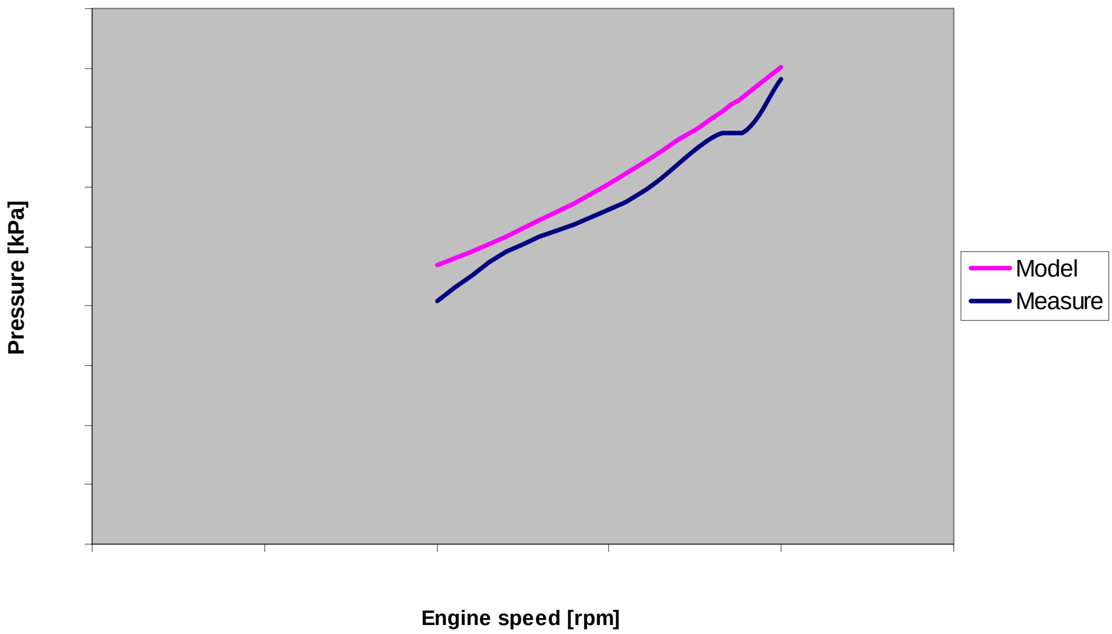

Figure 45. Pressure before transmission, Test 13.

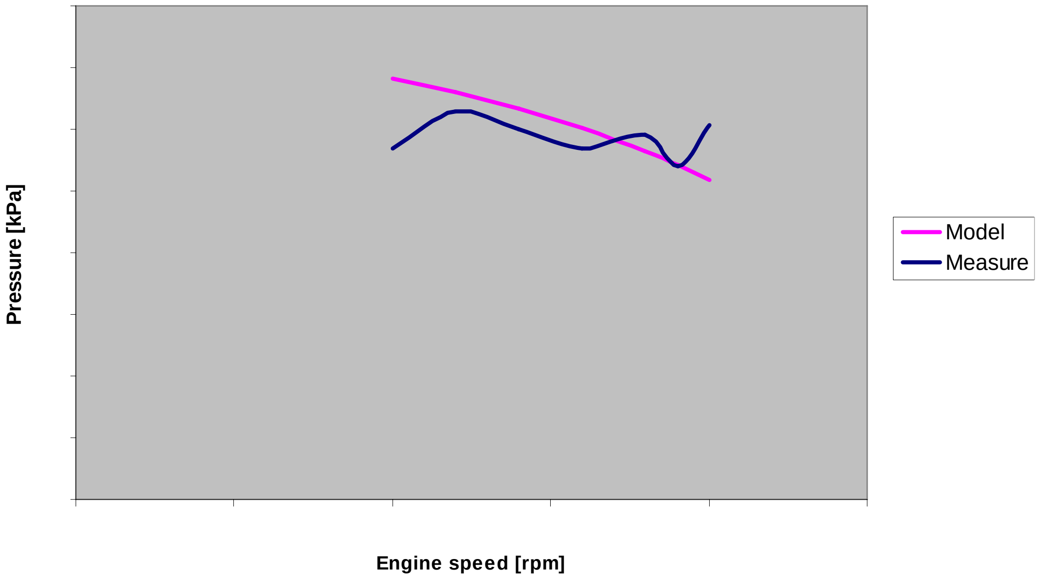

Figure 46. Pressure before coolant pump, Test 13.

5.4 Validation, Test 15

To further validate the model values from experimental data Test 15 has been used. To see if the model correspond well with the cooling system the system curve for the model is compared with the system curve for the experimental data in Figure 47.

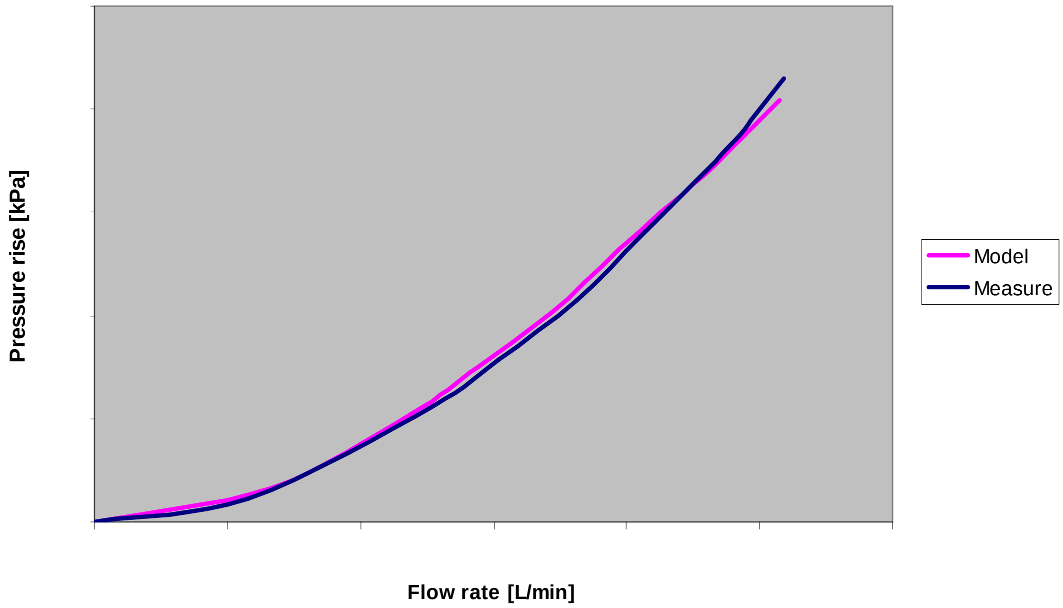

Figure 47 System curve, Test 15.

The difference between the curves is very small, under 3%. This shows that the coolant pump in the AMESim model gives nearly the same pressure rise as the real coolant pump at a specific flow. Each flow over a component is furthermore validated separately from the others. The results of these flow comparisons are displayed in Table 4 below.

	2000	1900	1800	1600	1400	1200	1000
Urea	-0,37%	-0,41%	-1,83%	-1,06%	-1,72%	0,82%	2,44%
Cab heater	4,03%	3,20%	2,96%	1,73%	0,58%	0,65%	0,40%
Servo	14,58%	18,15%	61,19%	------------	------------	------------	------------
Radiator	-2,59%	-3,71%	-4,90%	-5,89%	-6,90%	-6,35%	-8,70%
Transmission	5,89%	5,68%	5,81%	3,79%	2,04%	1,99%	2,83%
Air compressor	-0,07%	-3,28%	-4,98%	-4,96%	-3,66%	-0,93%	4,22%
Coolant pump	-0,76%	-1,81%	-2,61%	-4,40%	-5,40%	-4,82%	-6,68%

Table 4. Differences between model and measured data in %, Test 15.

Table 4 showed that the largest difference is 8.70 %, which is good for these types of models. There is still problem with the servo oil cooler because of the large differences from the experimental data. Some part of the difference can be derived from difficulties in measurement and other from approximation in the model. To visualise these results two curves are presented in Figure 48 and Figure 49. The other curves are presented in Appendix1.

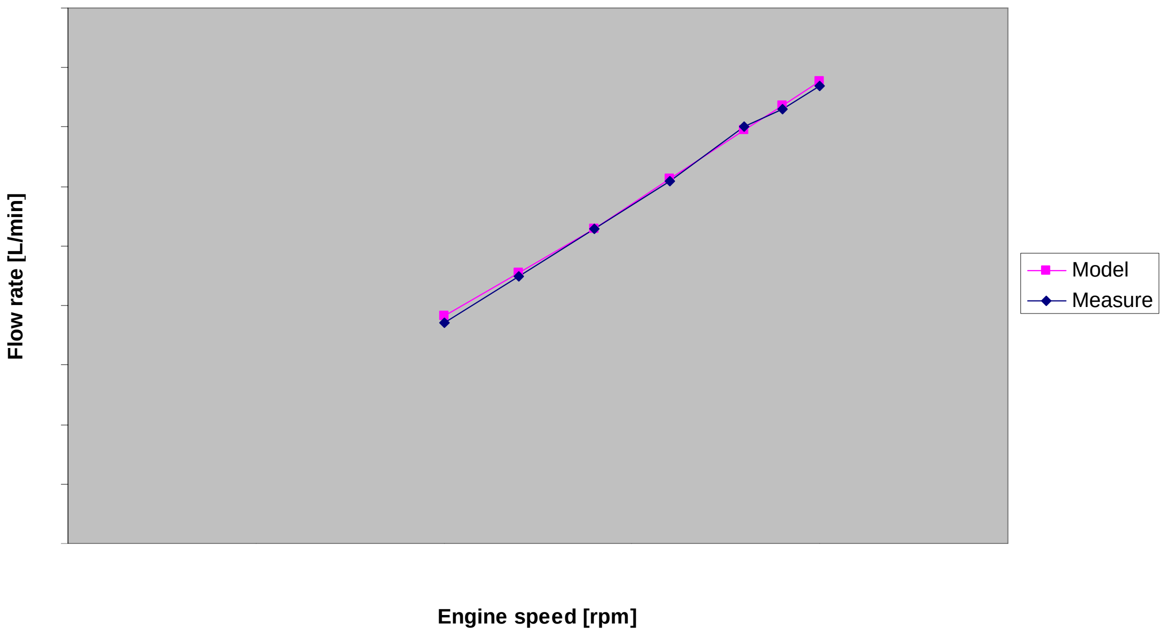

Figure 48. Flow through urea heater, Test 15.

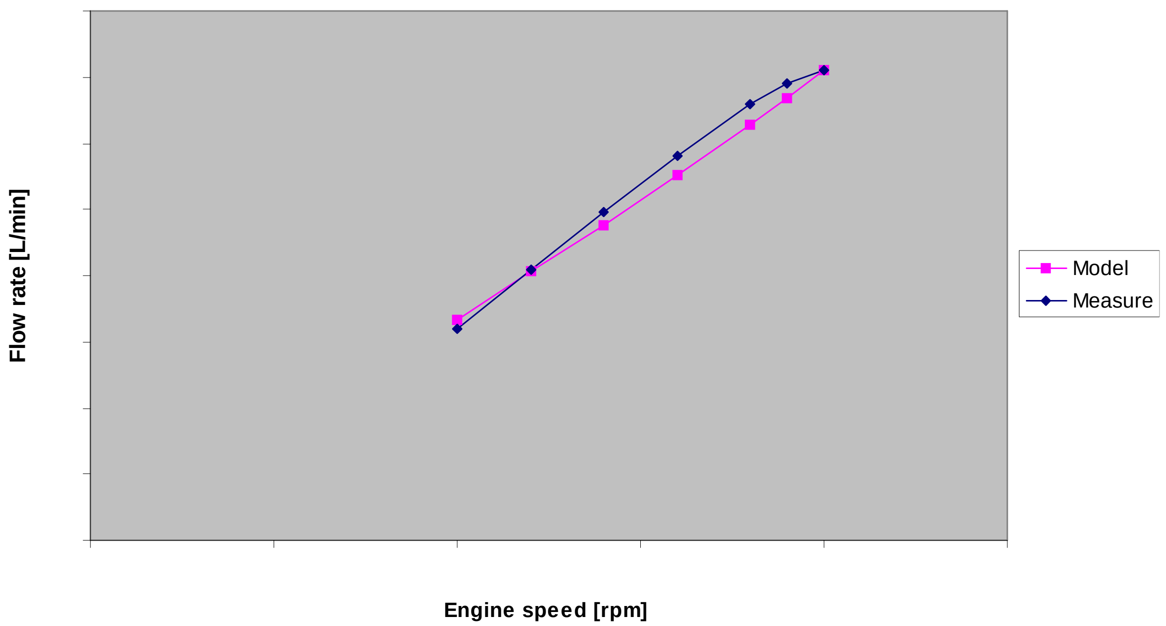

Figure 49. Flow through air compressor cooler, Test 15.

These curves show that the model built in AMESim corresponds well with the measurement from experimental data. Together with the curve in Appendix 1 and table 4 it shows that flow is predicted well over the interval [1000 2000].

The pressure validation of the model showed below in Figure 50 and Figure 51. In the latter the pressure after the coolant pump correspond well with the experimental data. In Figure 50 however the curve for measure data stop increasing and this has to do with different high-pressure release valve levels. The level is set a little higher in the model than in the cooling system.

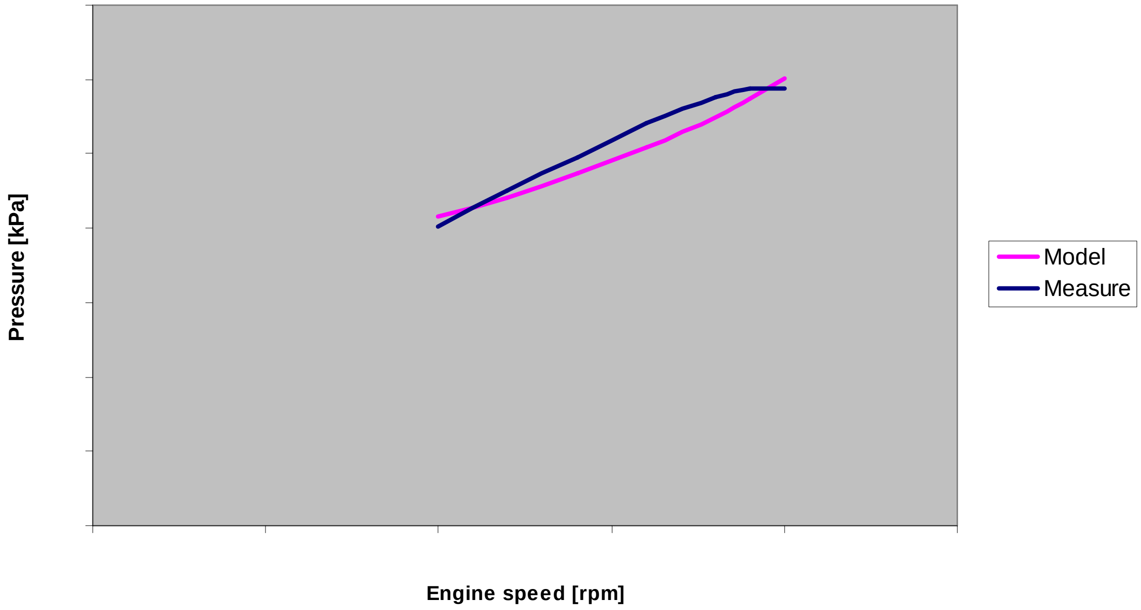

Figure 50. Pressure before radiator, Test 15.

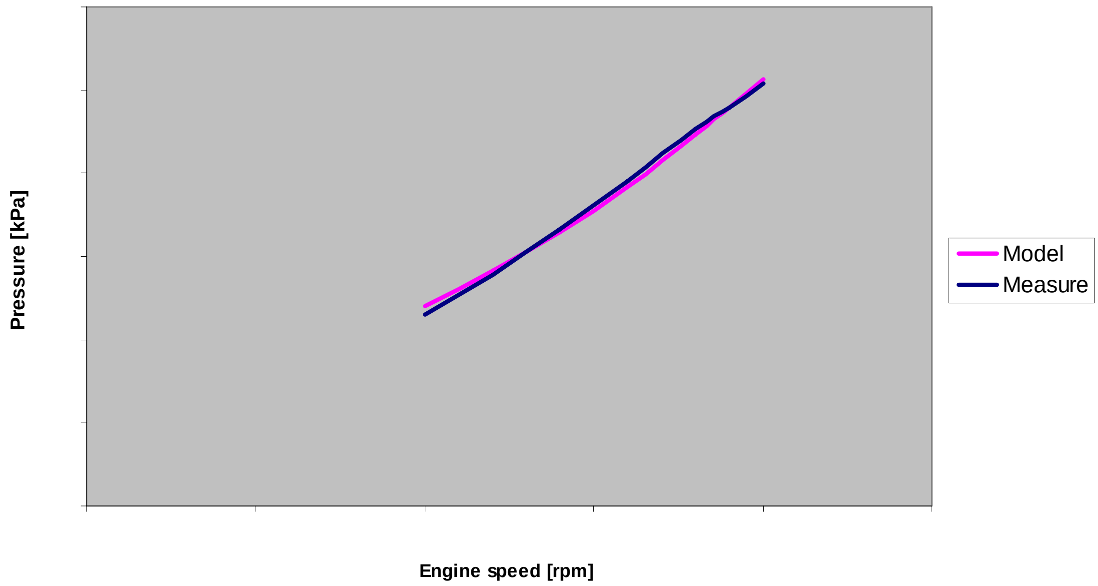

Figure 51. Pressure after coolant pump, Test 15.

6 Design Modification

An interesting case is when the design of the cooling system is changed. In this thesis project one modification is done to illustrate how the system design influences the flow distribution in the cooling system.

One modification that has been considered at Volvo 3P is to move the servo oil cooler inlet. Before the modification the inlet was at the oil cooler duct but it have been moved to after the cylinder block and head. Also the cab heater branch inlet has been change from the oil cooler duct to a position between the duct and the coolant pump and this may also affect the urea heater. In order to see how the flow through the servo oil cooler, the cab heater and the urea heater changes, this design modification is interesting. The new design is show in Figure 52.

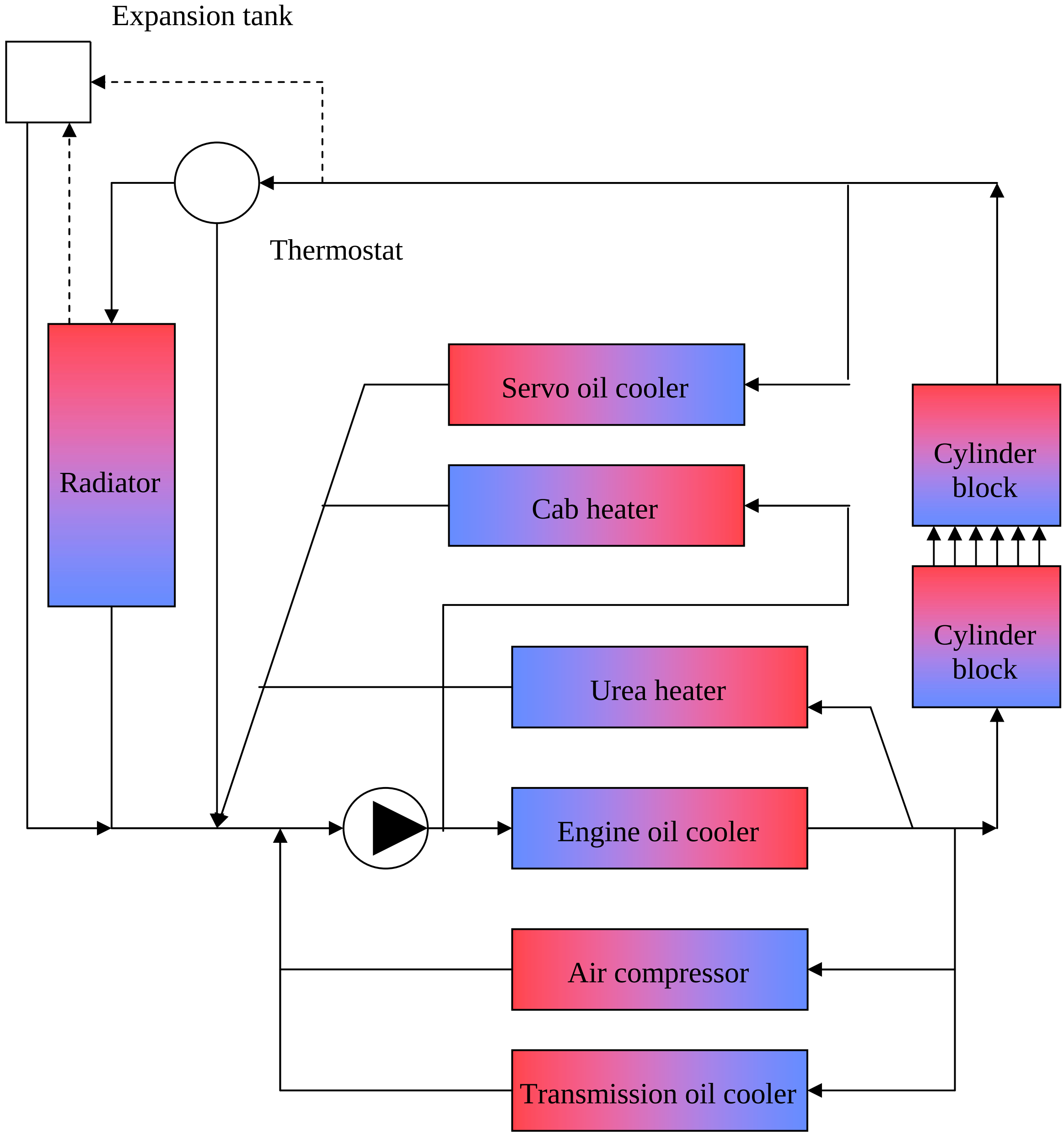

Figure 52. The modified circuit.

When comparing the flow in the coolant circuit before and after the design modification it shows that the flow in the cab heater has increased and the flow through the servo oil cooler has decreased. In the other component directed involve in the modification, the urea heater, the change has nearly no effect on the flow through this. The increase in % is showed in Table 5. This showed that to get a larger flow to the cab heater this design modification could be used without affecting the flow through the urea heater.

	2000	1900	1800	1600	1400	1200	1000
Urea	0,7%	0,7%	0,8%	0,9%	1,0%	1,0%	1,3%
Cab heater	24,0%	24,1%	24,3%	25,1%	27,0%	29,5%	30,2%
Servo	-33,9%	-34,0%	-34,1%	-32,6%	-----------	-----------	-----------

Table 5. Flow differences in % due to the design modification.

7 Result

In this chapter the result from the validation chapter is presented. First for the volume flow rate and then for the pressure and at the end transient.

7.1 Flow

The previous chapter shows that the model built in AMESim gives a good approximation of the experimental data in both validation tests. Especially the coolant volume flow rate differs very little between the measured data and the values given by the model. There it is just the volume flow rate through the servo oil cooler that gives big differences at all engine speeds and in both tests. It can also be worthwhile to notice that the flow through the servo oil cooler is the smallest of all the flows. This will not influence the temperature in the coolant system much because it is a small contributor to the system. It is also just half the flow compared the urea heater, which has the second smallest flow of all the components. The flow through the urea heater however has just a 5,8 % maximum difference between the model and the measured data. The maximum differences are found at 1000 rpm for all tests.

The model built in AMESim corresponds well for all volume flow rates except the servo oil cooler, which has a very low flow. This could, as stated before, depend on a wrong chosen sensor because the sensor used for the experimental data was designed for higher flows.

For the air compressor cooler the largest differences are found in the middle of the analysed interval and not as stated before at the lowest rpm. The flow through the coolant pump, which is the same as the maximum flow in the circuit, corresponds well with the cooling system. The differences are around 6% and the model can therefore be used for it purpose because these differences are lower than the stated goals from Volvo 3P. A summary of the largest flow differences is showed in the Table 6. Here is also the result from Test 11 after the parameter settings presented.

	Test 11	Test 13	Test 15
Urea Heater	3,2%	5,8%	2,4%
Cab Heater	5,5%	3,3%	14,0%
Servo oil cooler	-----	24,6%	61,2%
Radiator	7,9%	7,2%	8,7%
Transmission oil cooler	6,9%	5,5%	5,9%
Air compressor cooler	7,5%	3,4%	5,0%
Coolant pump	6,4%	6,0%	6,7%

Table 6. Maximum flow rate differences.

Table 6 shows that the maximum differences are not large except for the servo oil cooler and for the cab heater in Test 15. It is worthwhile to mention that the table showed the maximum value and not a mean value. If one value from a component is much larger than the others the maximum flow rate will be large. One value can therefore give a big influence in Table 6. A more complete table with engine speed rate can be found in Appendix 1.

7.2 Pressure

The result in terms of pressure is more complex than the volume flow rate. This can be seen in the figures found in Chapter 5 and Appendix 1. Some point of measurement can be very difficult to validate and compare due to the uncertainty in the measurement methods, approximation in the model, wrong chosen sensors and the placing of sensors. This can result in big differences between the model and experimental data. The model will be a harsh approximation because of this and the judgement of how well the model correspond with the cooling system can only be based on if the differences are not to big. This can of course be a question about what is ok and if the model can fill its purpose. Under the circumstances due to the uncertainty of the experimental data the differences in this thesis work are not perfect but not too bad according to Volvo 3P.

The result of the validation chapter and the additional figures in Appendix 1 shows that the model approximated from the experimental data of the coolant circuit is quite good. This result holds for every point measured in the system. The maximum value of the differences is showed in Table 7. Here is also the result from Test 11 after the parameter settings presented.

	Test 11	Test 13	Test 15
Pressure before Coolant pump	19,7%	20,1%	32,7%
Pressure before Radiator	12,4%	14,0%	6,1%
Pressure after Coolant pump	4,6%	7,3%	1,6%
Pressure on oil duct	14,8%	16,1%	10,9%
Pressure before Transmission oil cooler	17,7%	14,8%	20,7%
Pressure after Transmission oil cooler	12,3%	16,5%	6,3%

Table 7. Maximum pressure differences.

In Table 6 there is the smallest value for the pressure after the coolant pump. There the model is very good compared to the experimental value for all three tests. The pressure curves can be seen in Appendix 1 and Chapter 5. If Table 5 and Table 6 are compared it becomes obvious that the volume flow rate is better approximated than pressure.

8 Conclusions

The model described in Chapter 4 gives a good picture of the truck cooling system for a FH31 MD13 EURO5 520 hp. This has been showed by validation in the previous chapter. Because of several uncertainties the model is only valid for engine speeds between 1000 and 2000 rpm. Outside this interval the result from the model can not say anything about the cooling system and it behaviours. In order to be able to simulate the truck cooling system at engine speeds lower and higher than this interval, further studies are needed. One other way is to have more than one model where each one is valid in different intervals. The conclusion from the pressure validations is that the validation is far from perfect due to several uncertainties. Therefore further work is desired in order to have a better and more accurate pressure validation. On the other hand the differences between the model and the cooling system is small for volume flow rates. This implies that the model works well if the purpose is to study the flow distribution in a coolant circuit.

The model built during this thesis work is an example of what is possible to do and simulate in AMESim. The model can therefore be a starting point for further modelling because of its accurateness for volume flow rate but also for pressure.

This thesis work shows that AMESim work very well as a 1D analysing tool, which meets the requirement set by Volvo 3P in Gothenburg. It can handle flow, pressure, heat transfer and temperature, which is desire for future product development.

9 Future

The reasons for building the AMESim model are to exam and develop a new 1D analysis method for cooling system. With new legislation for example emission, the need for more analysis increases, both 3D and 1D. The aim for this thesis work was to get an understanding for AMESim as a new 1D tool that could help Volvo 3P in Gothenburg in future product development. The conclusion from this thesis work shows that it is possible to use AMESim for this type of analysis. In the future it is most probably that this tool will play a part in the development of the Volvo truck of tomorrow.

References

[1]. Ekroth el al, *Tillämpad termodynamik*, Stockholm, 1994

[2]. Glad och Ljung, *Reglerteori*, Lund, 2003

[3]. Idel'cik I.E, *Handbook of hydraulic resistance Third Edition*, Boca Raton, 1994

[4]. LMS, *AMESim Rev7 Tutorial*, Lyon, 2007

[5]. Miller D.S, Internal Flow System Second Edition, Watford, 1996

[6]. Stork et al, *Formelsamling i termo och fluiddynamik*, Linköping, 2003

[7]. Young et al, *A Brief Introduction to Fluid Mechanics*, 2004

Appendix 1

In this appendix the comparison curves from the parameter settings and validation will be presented for every point of measurement.

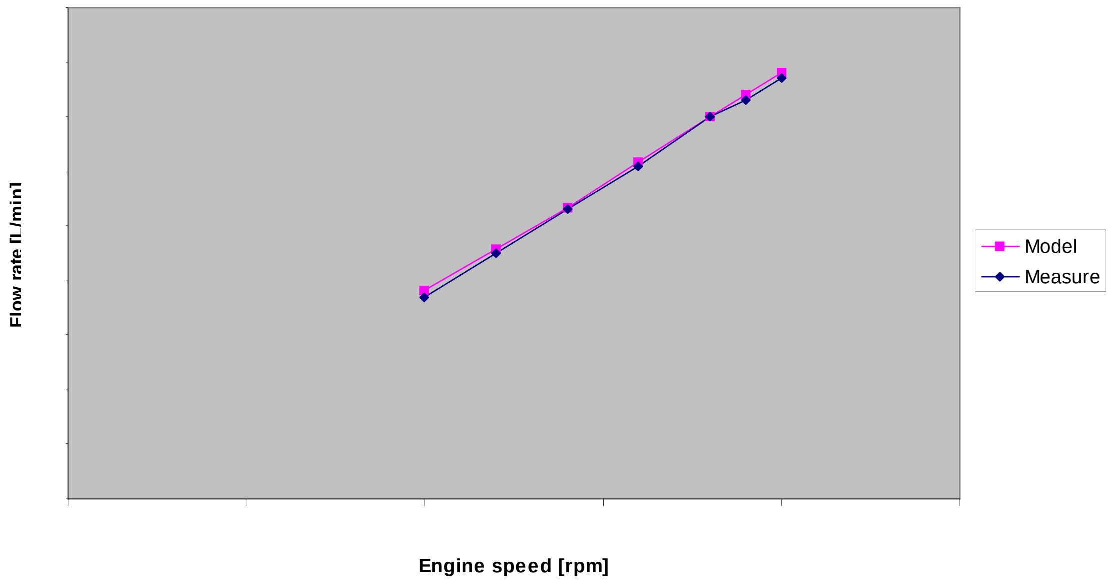

Figure 53. Flow through urea heater, Test 11.

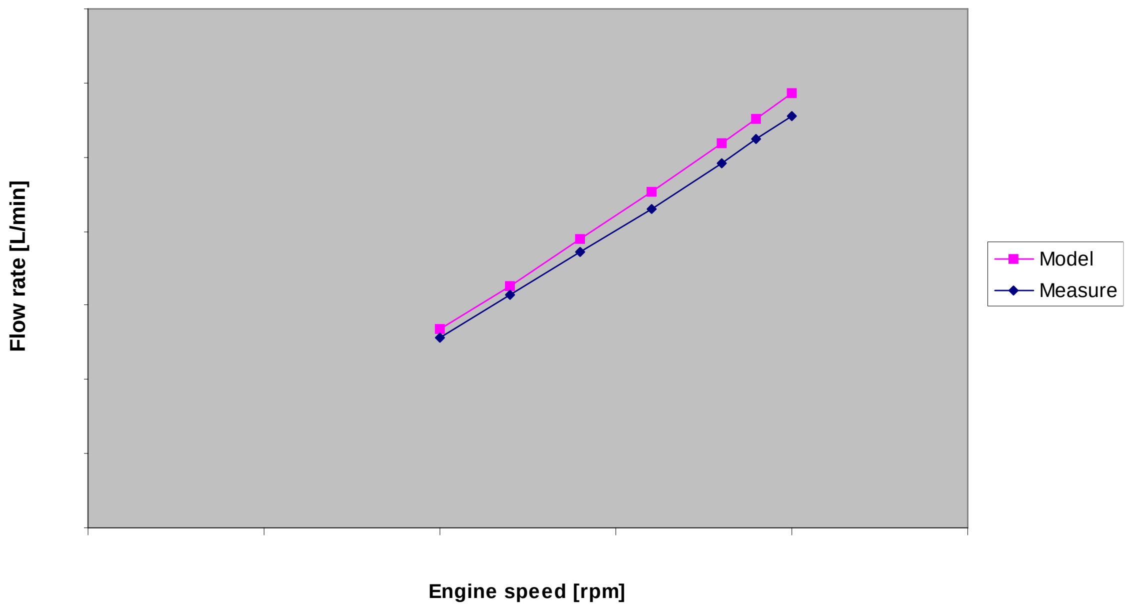

Figure 54. Flow through cab heater, Test 11.

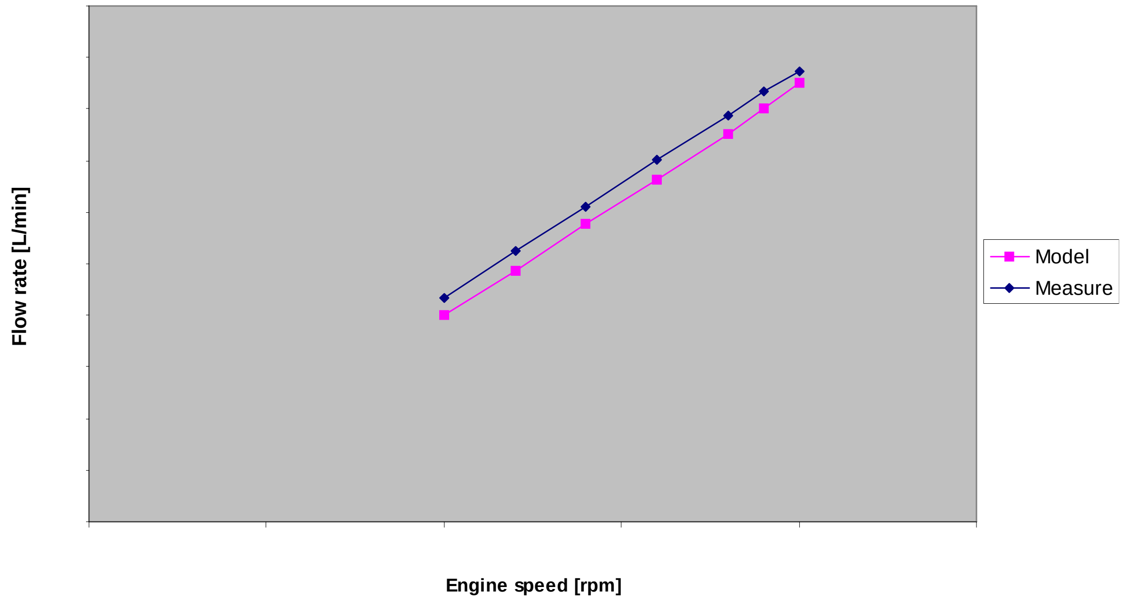

Figure 55. Flow through radiator, Test 11.

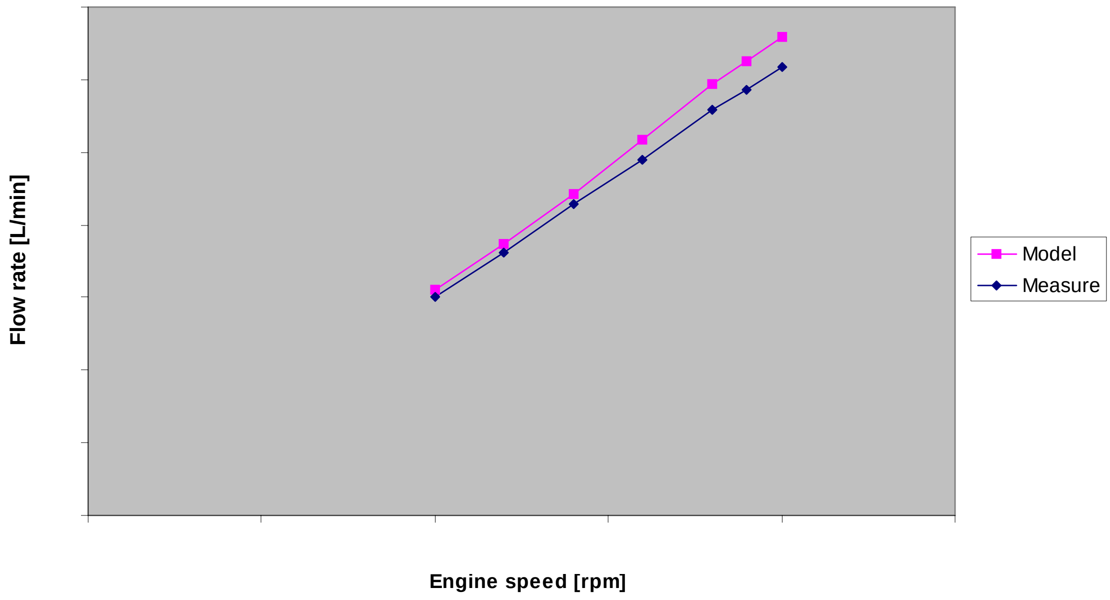

Figure 56. Flow through transmission oil cooler, Test 11.

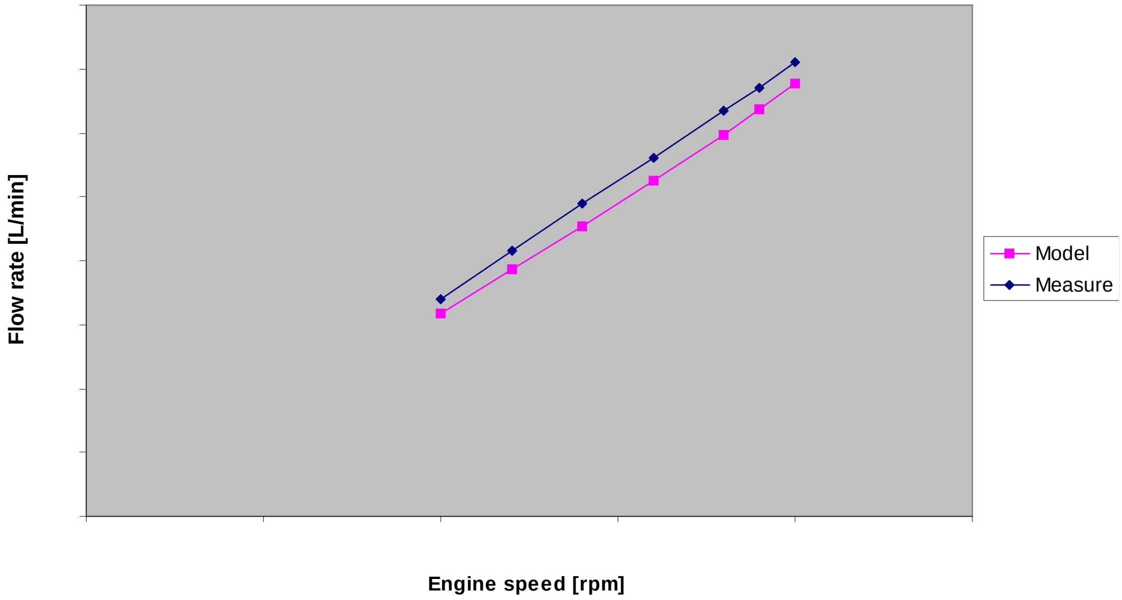

Figure 57. Flow through air compressor cooler, Test 11.

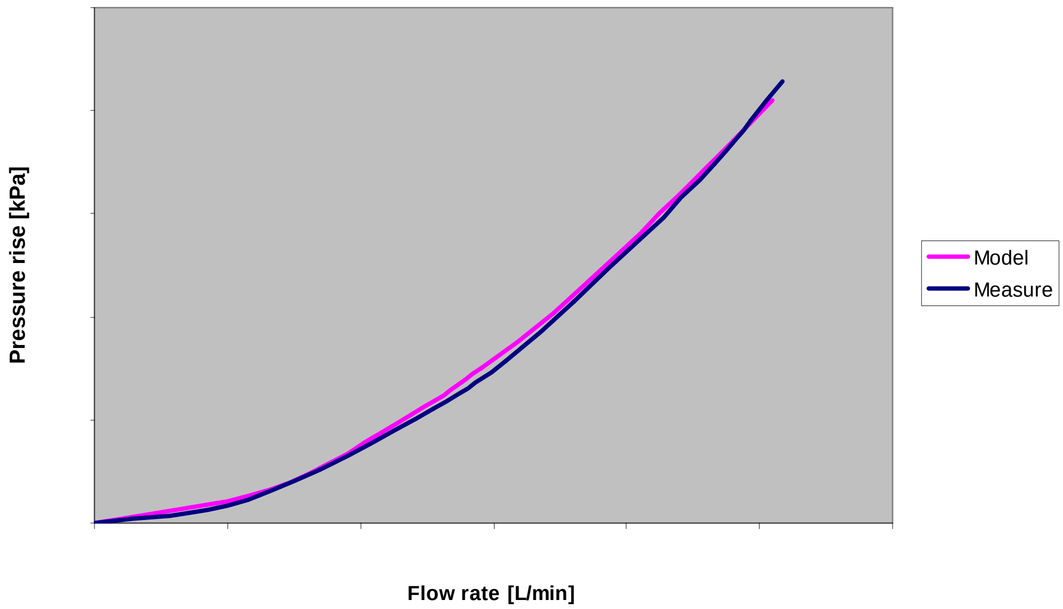

Figure 58. System curve, Test 11.

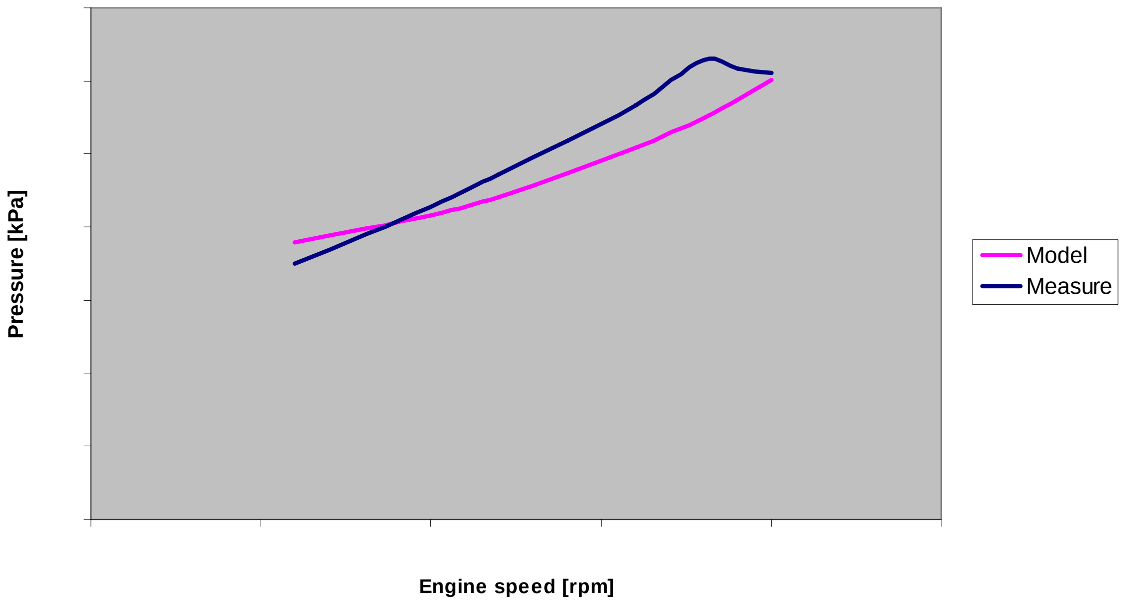

Figure 59. Pressure before radiator, Test 11.

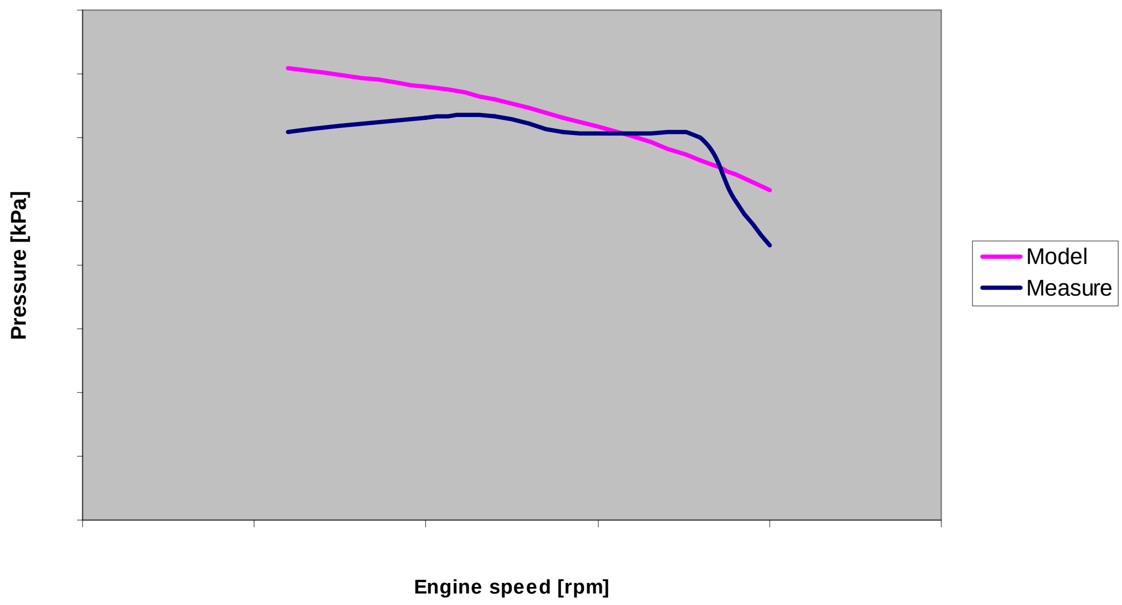

Figure 60. Pressure before coolant pump, Test 11.

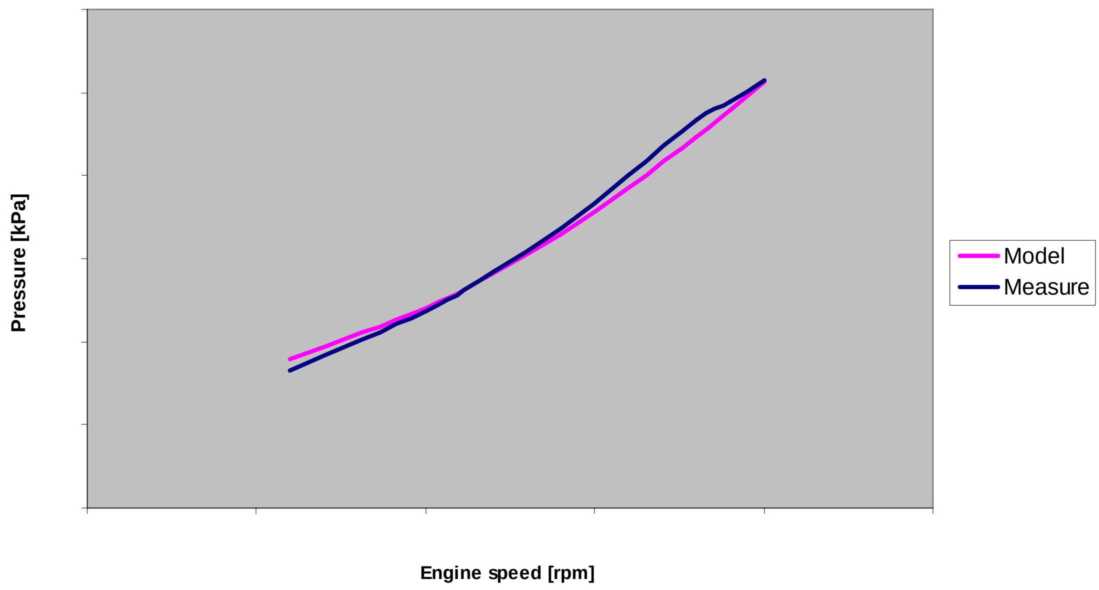

Figure 61. Pressure after coolant pump, Test 11.

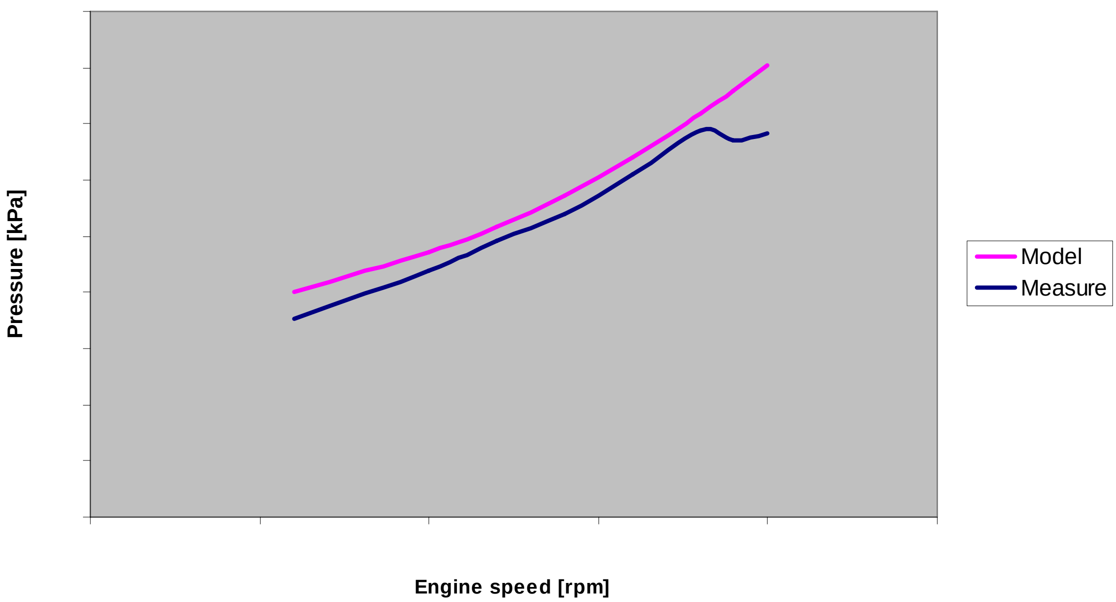

Figure 62. Pressure before transmission, Test 11.

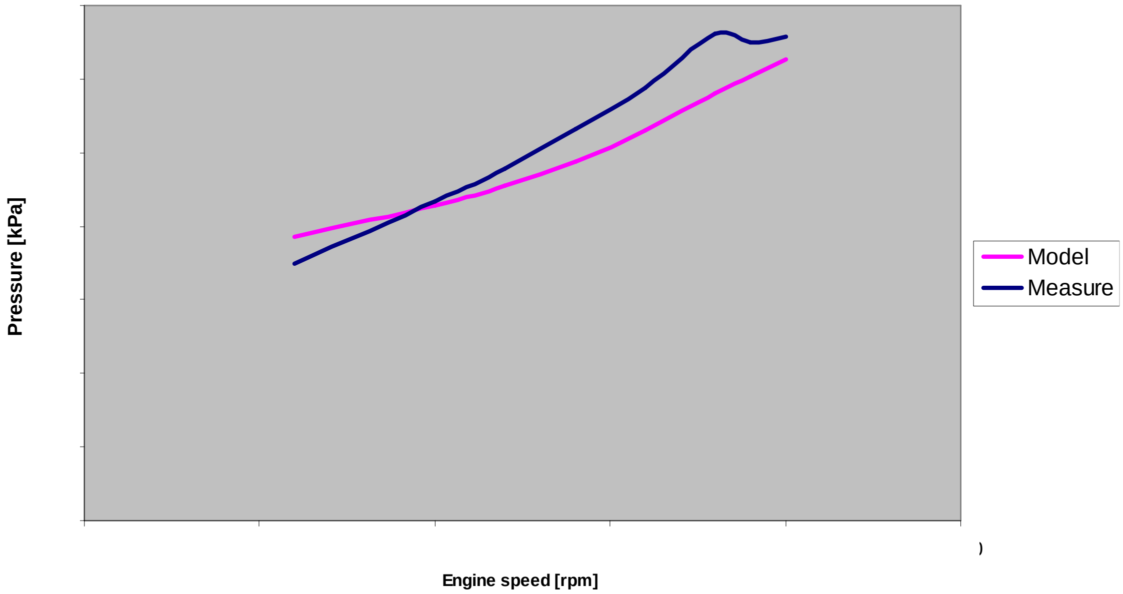

Figure 63. Pressure after transmission, Test 11.

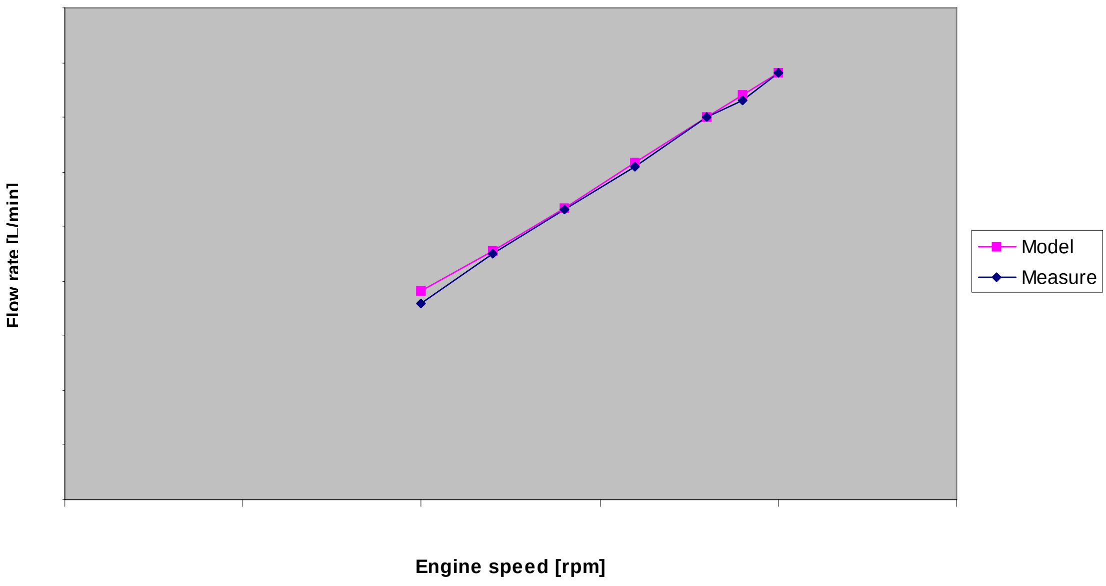

Figure 64. Flow through urea heater, Test 13.

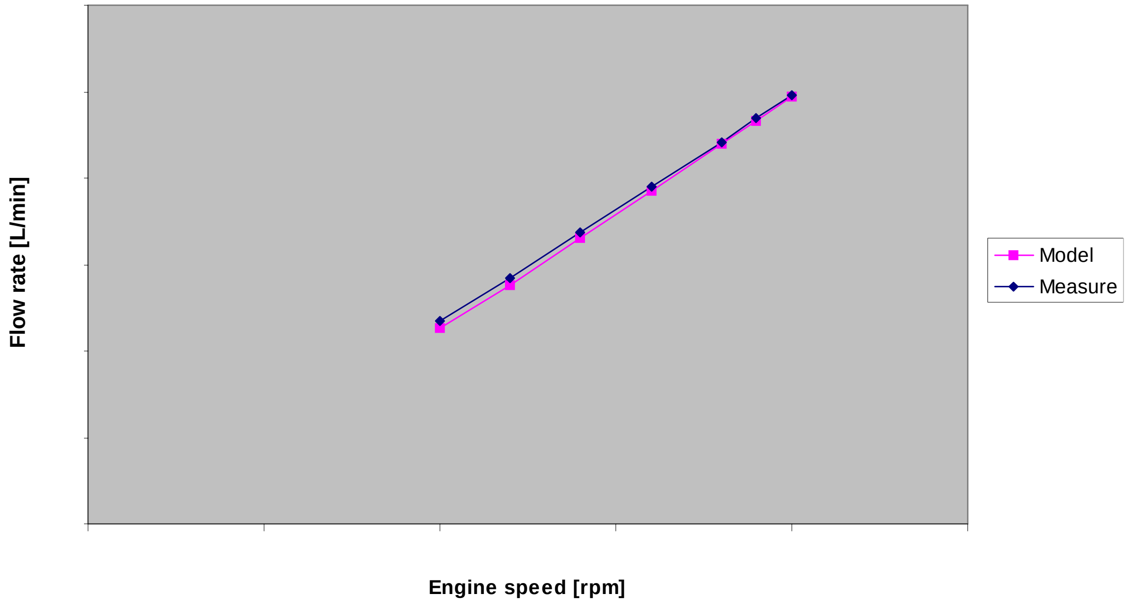

Figure 65. Flow through cab heater, Test 13.

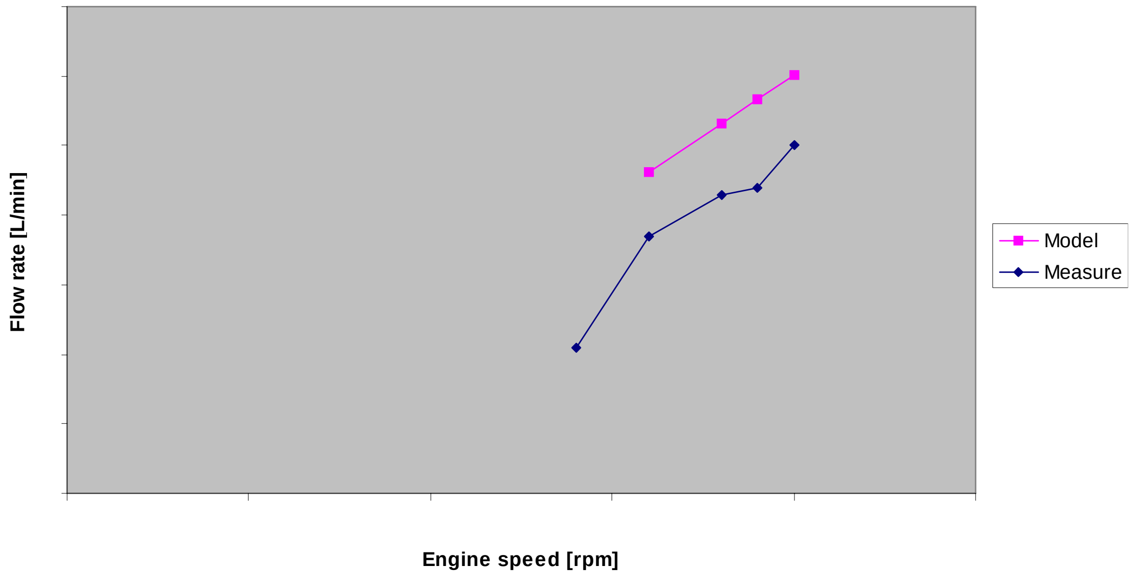

Figure 66. Flow through servo oil cooler, Test 13.

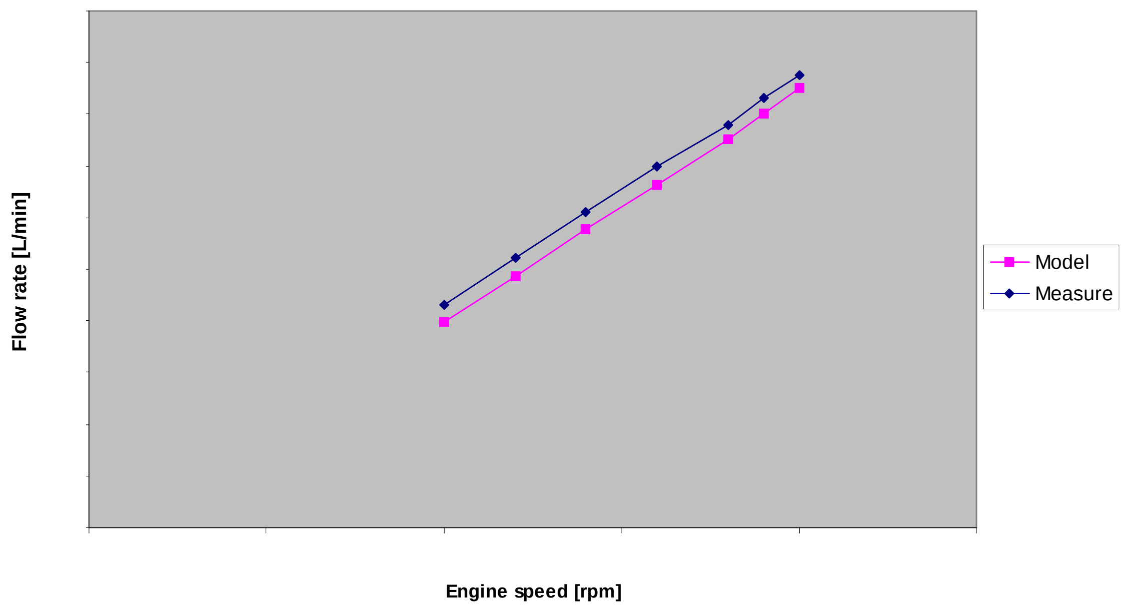

Figure 67. Flow through radiator, Test 13.

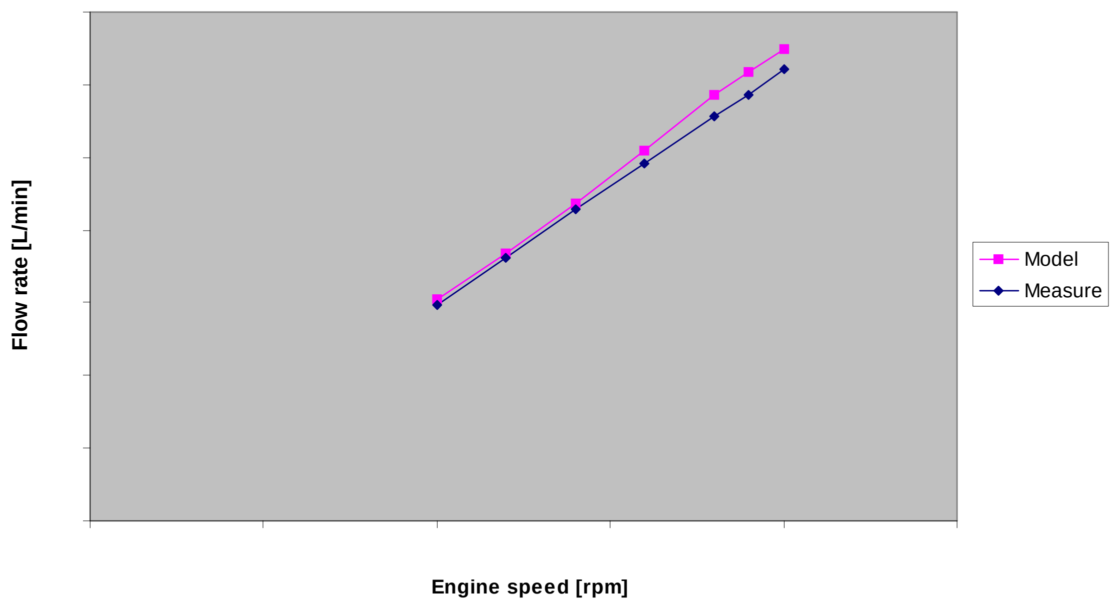

Figure 68. Flow through transmission oil cooler, Test 13.

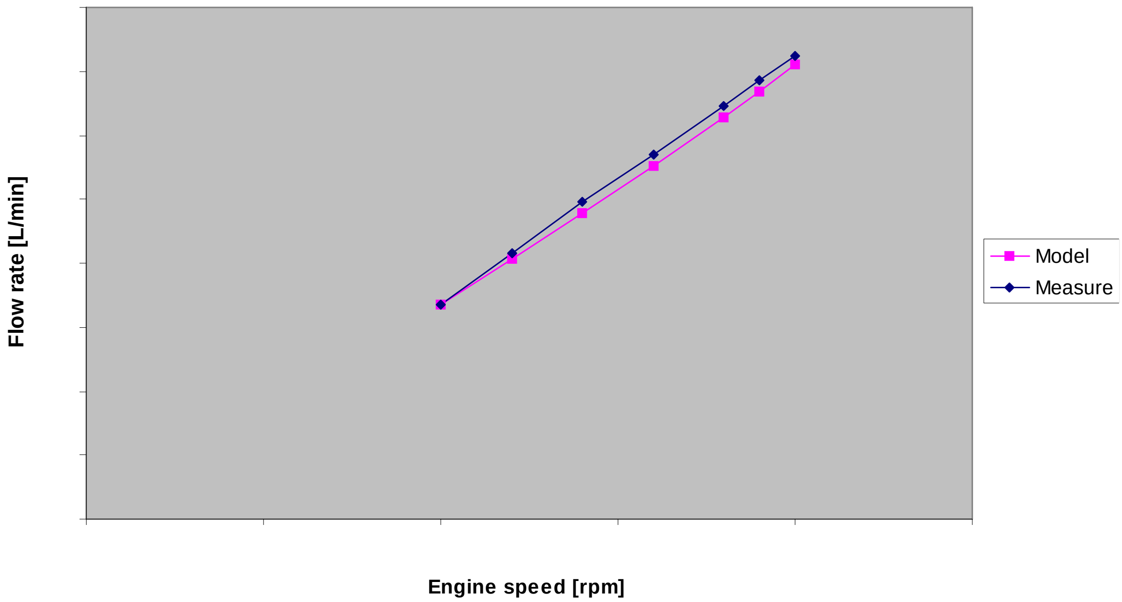

Figure 69. Flow through air comperessor cooler, Test 13.

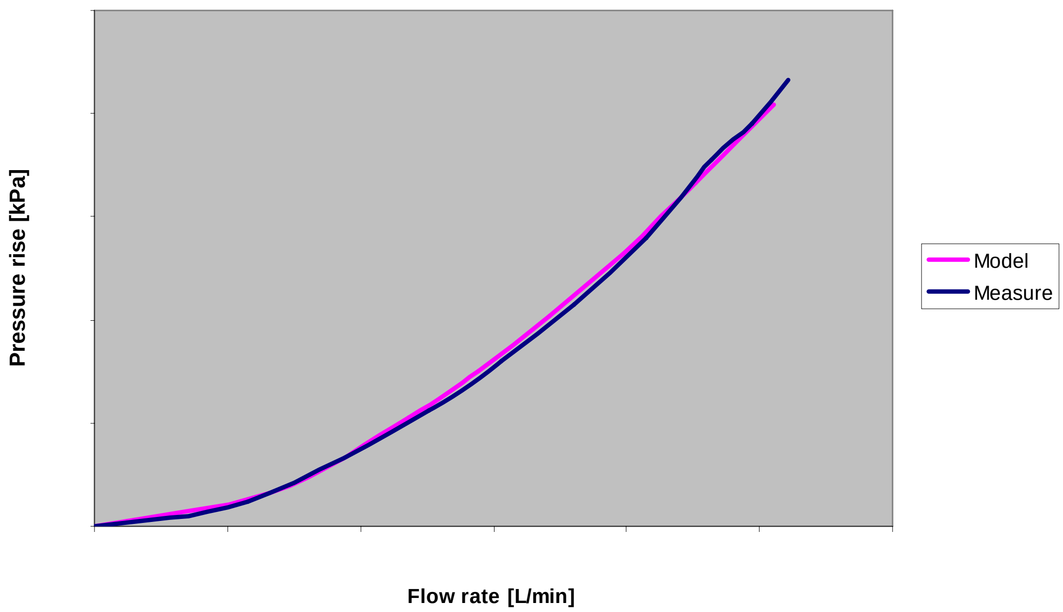

Figure 70. System curve, Test 13.

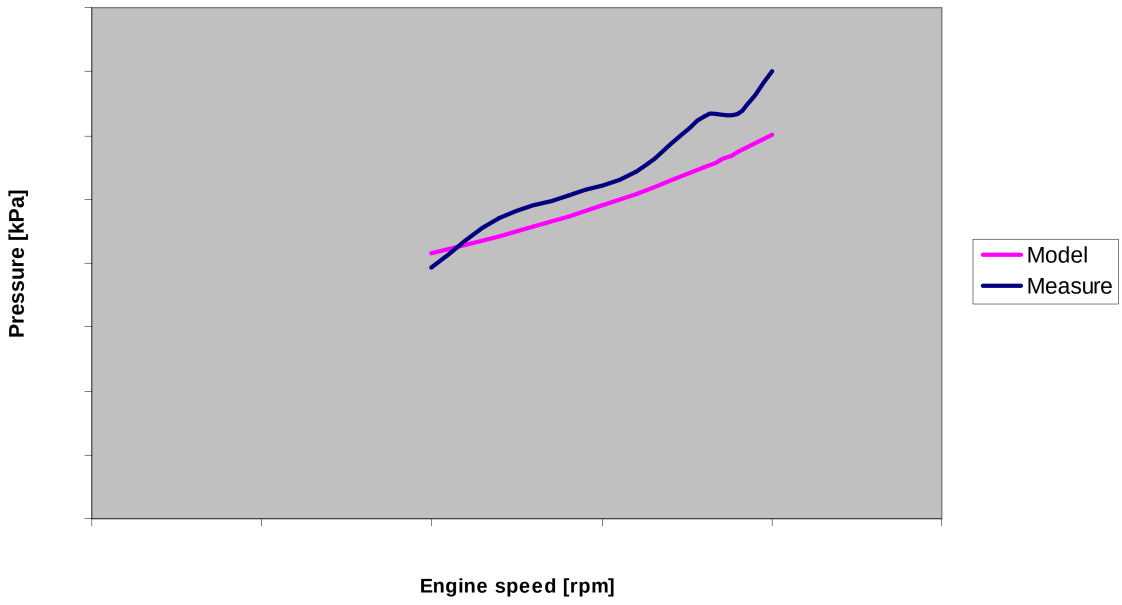

Figure 71. Pressure before radiator, Test 13.

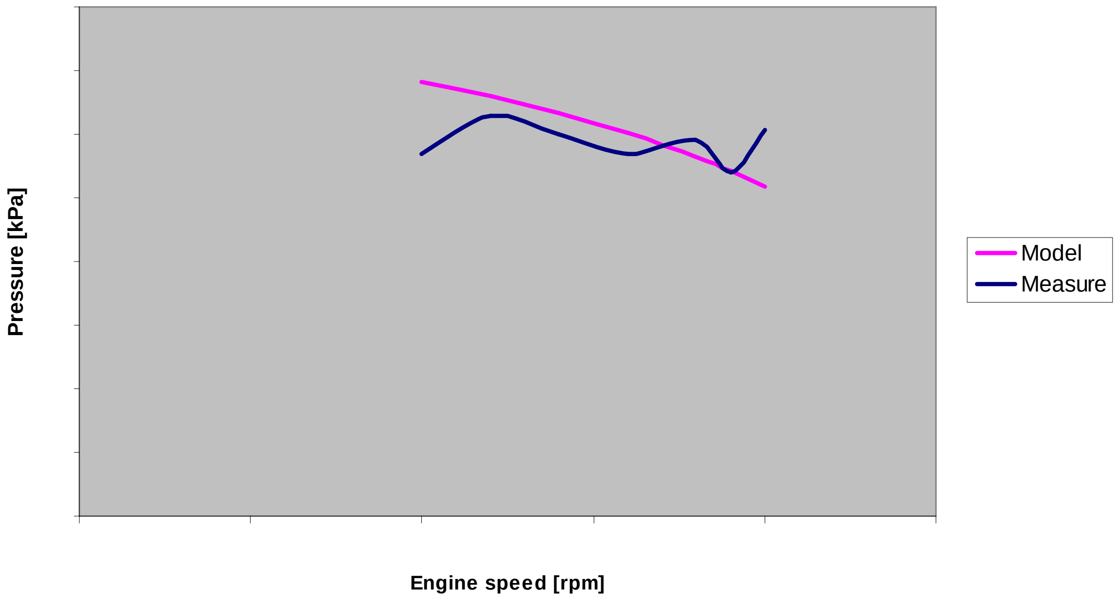

Figure 72. Pressure before coolant pump, Test 13.

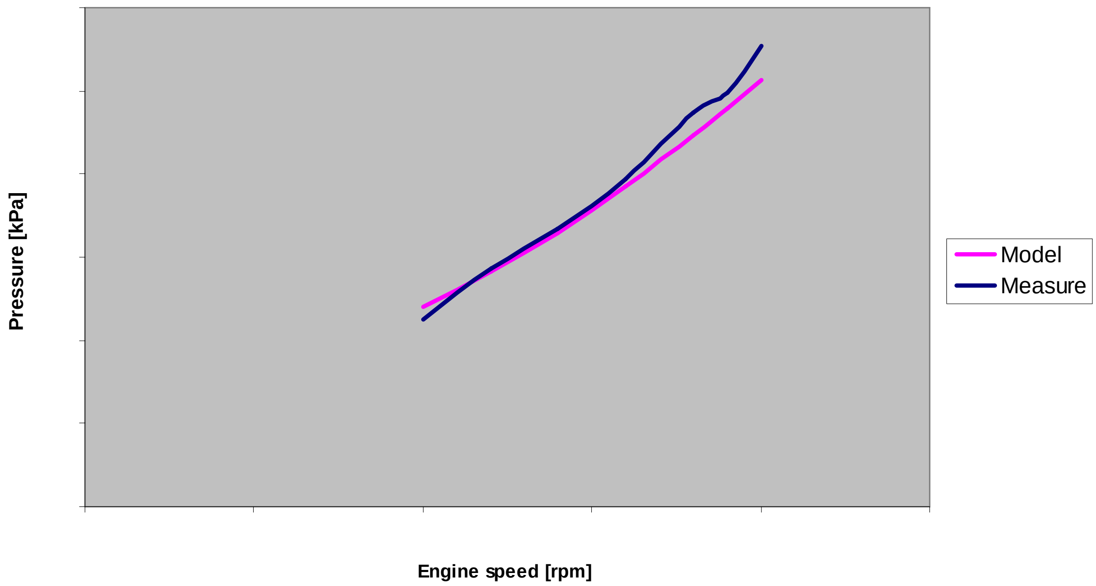

Figure 73. Pressure after coolant pump, Test 13.

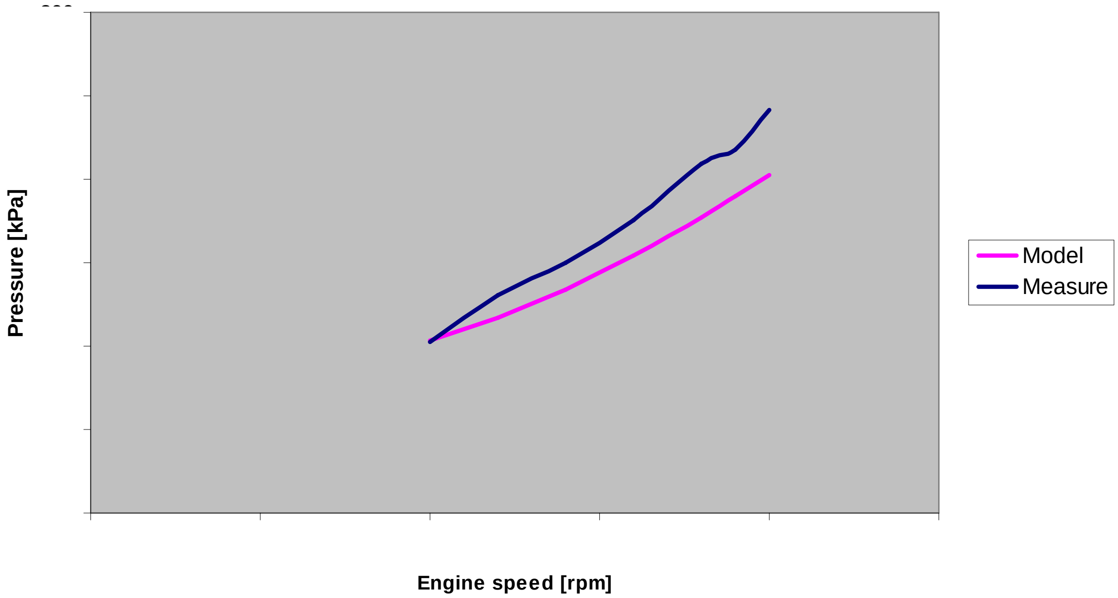

Figure 74. Pressure after oil cooler, Test 13.

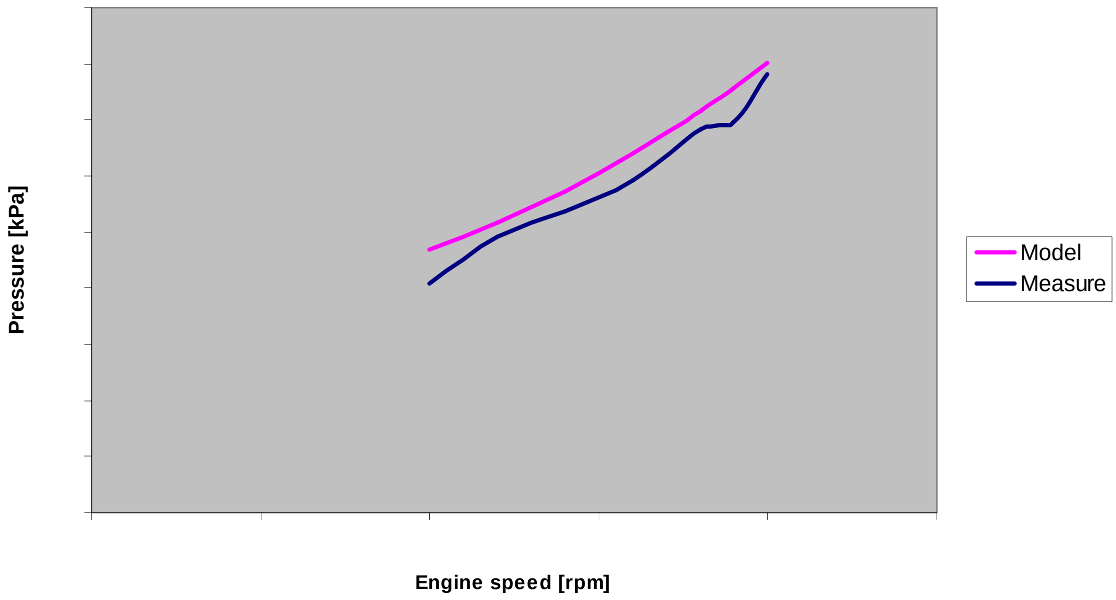

Figure 75. Pressure before transmission, Test 13.

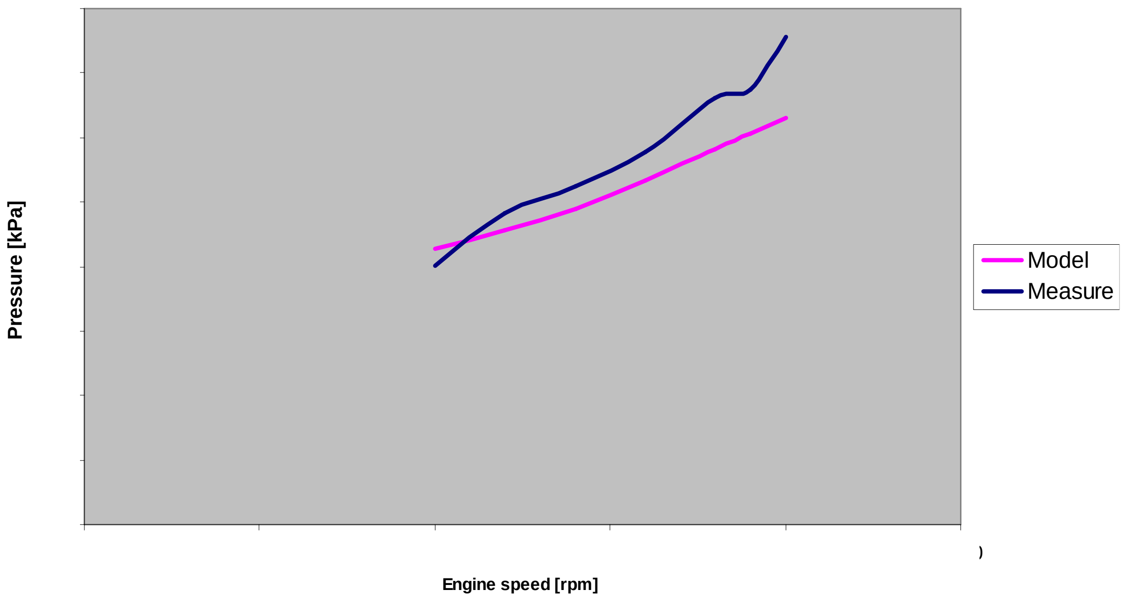

Figure 76. Pressure after transmission, Test 13.

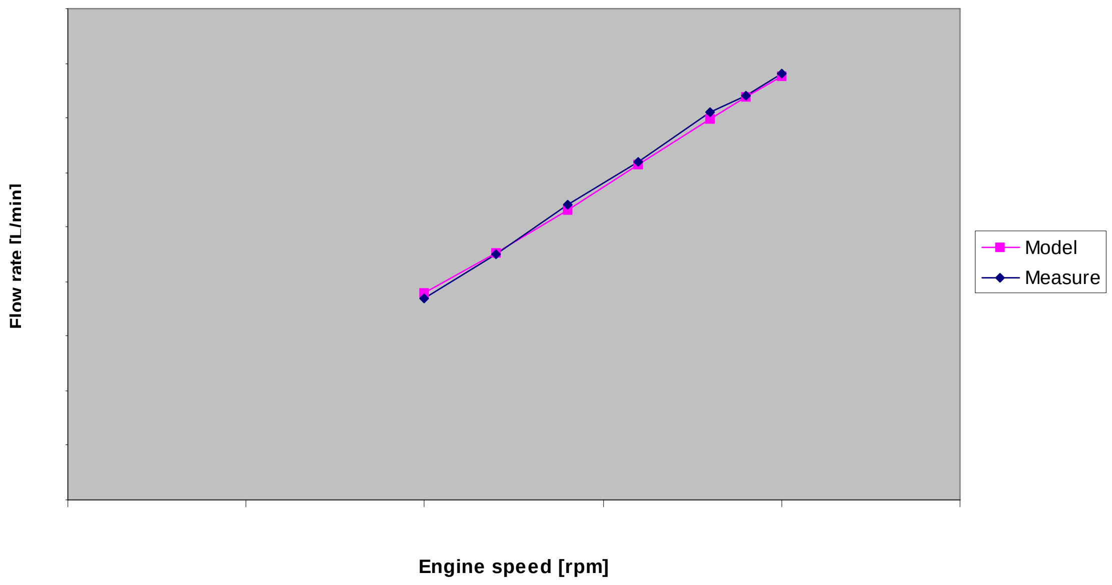

Figure 77. Flow through urea heater, Test 15.

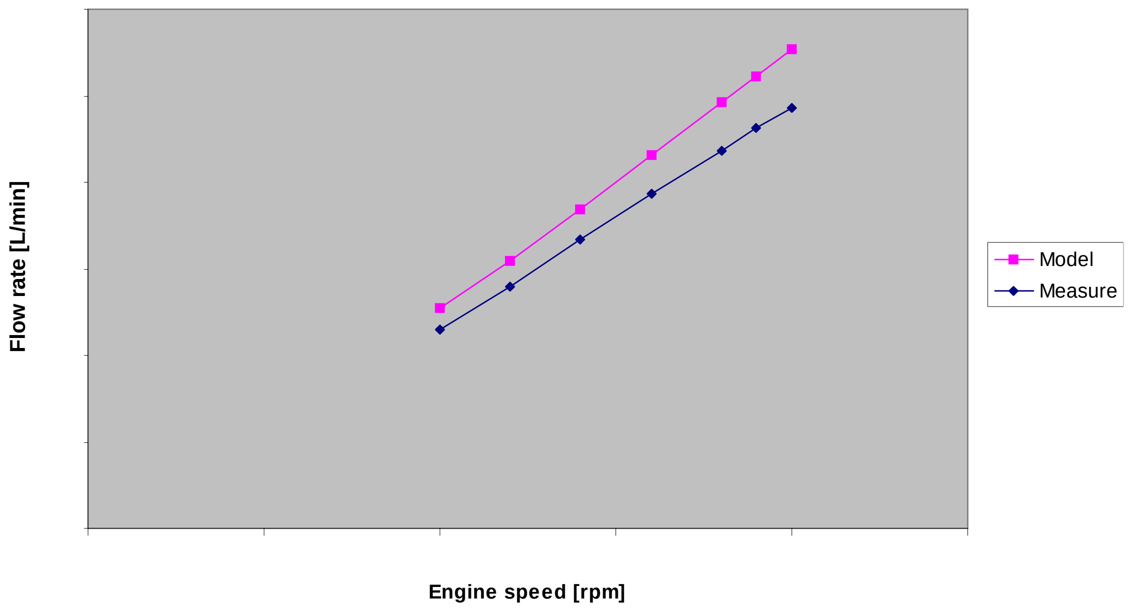

Figure 78. Flow through cab heater, Test 15.

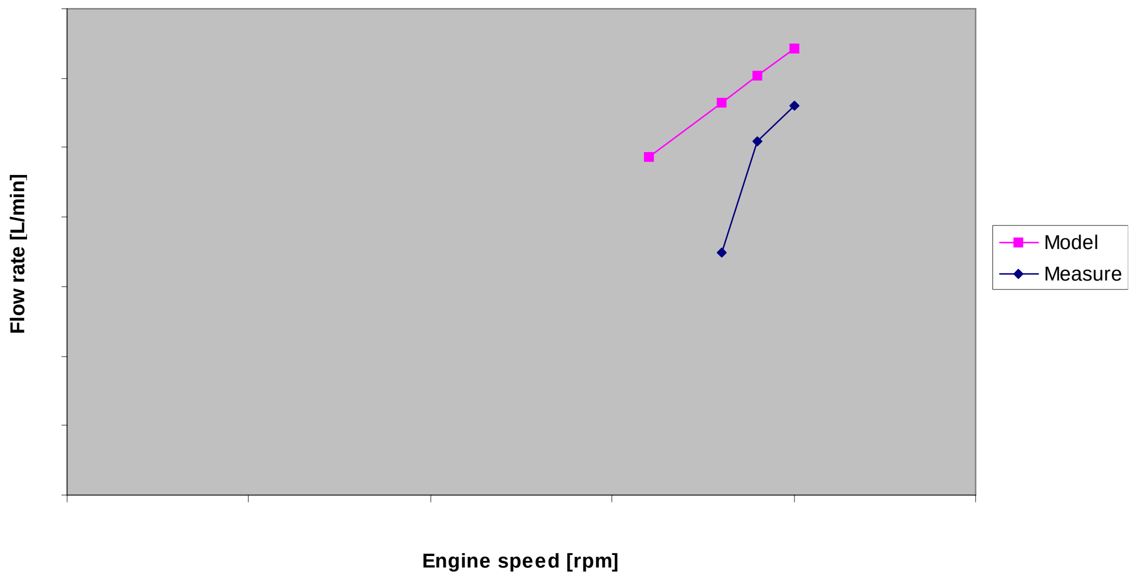

Figure 79. Flow through servo oil cooler, Test 15.

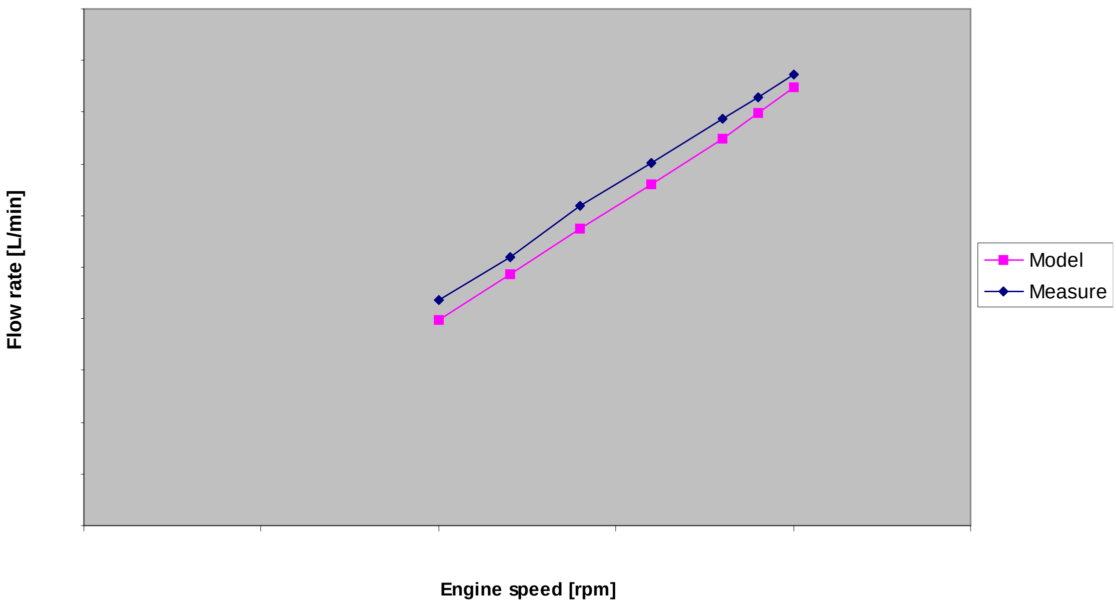

Figure 80. Flow through radiator, Test 15.

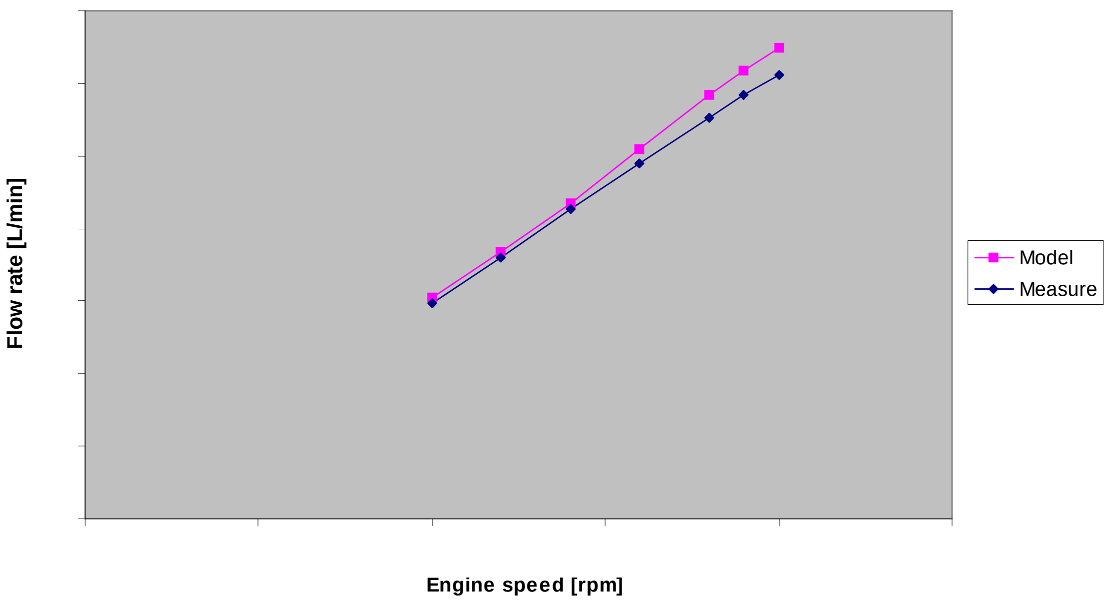

Figure 81. Flow through transmission, Test 15.

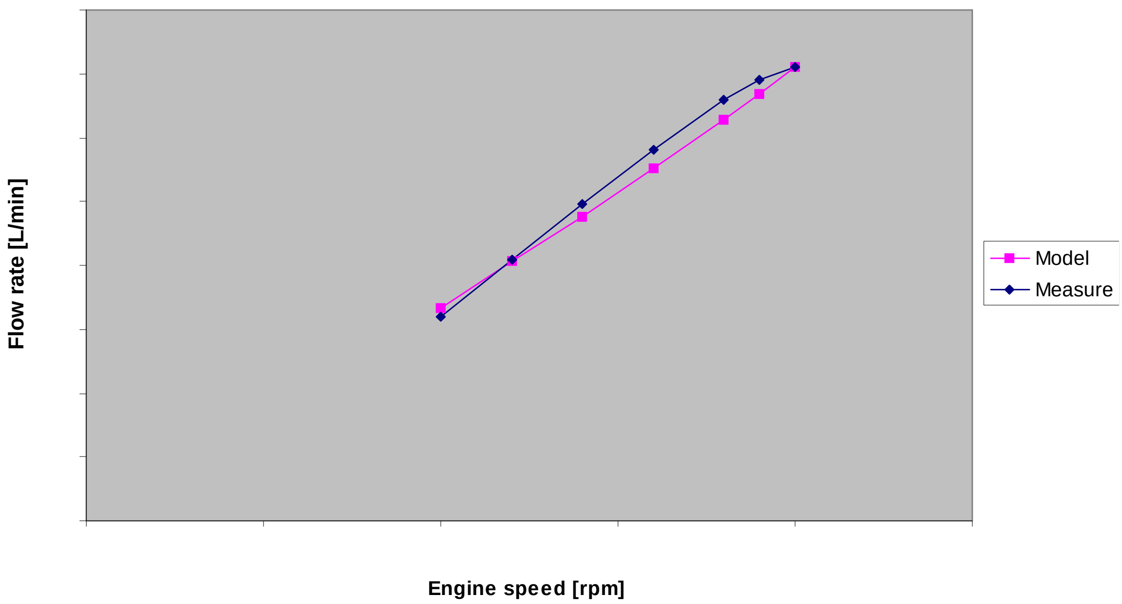

Figure 82. Flow through air compressor cooler, Test 15.

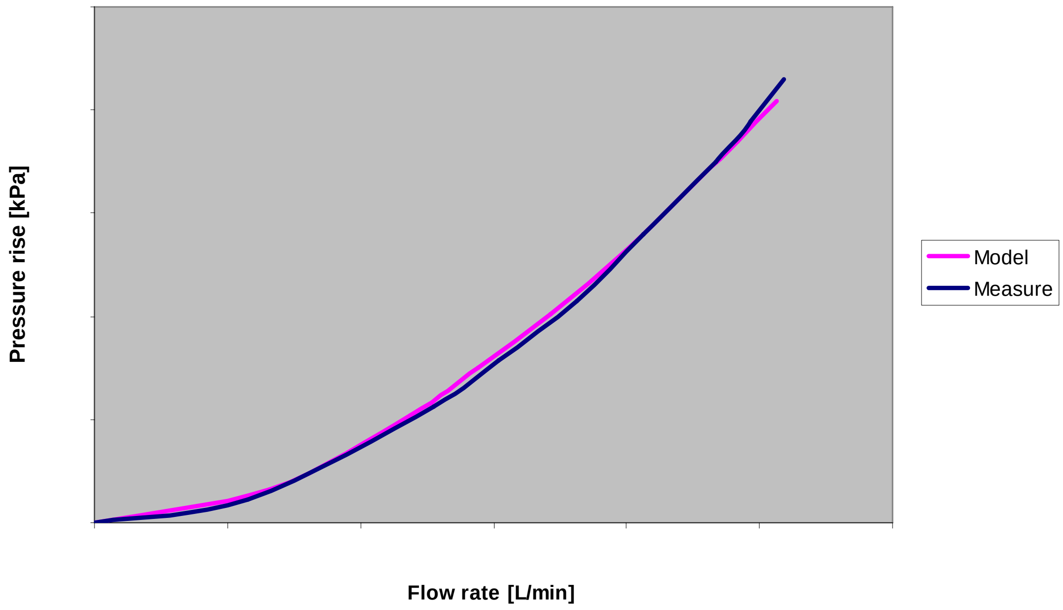

Figure 83. System curve, Test 15.

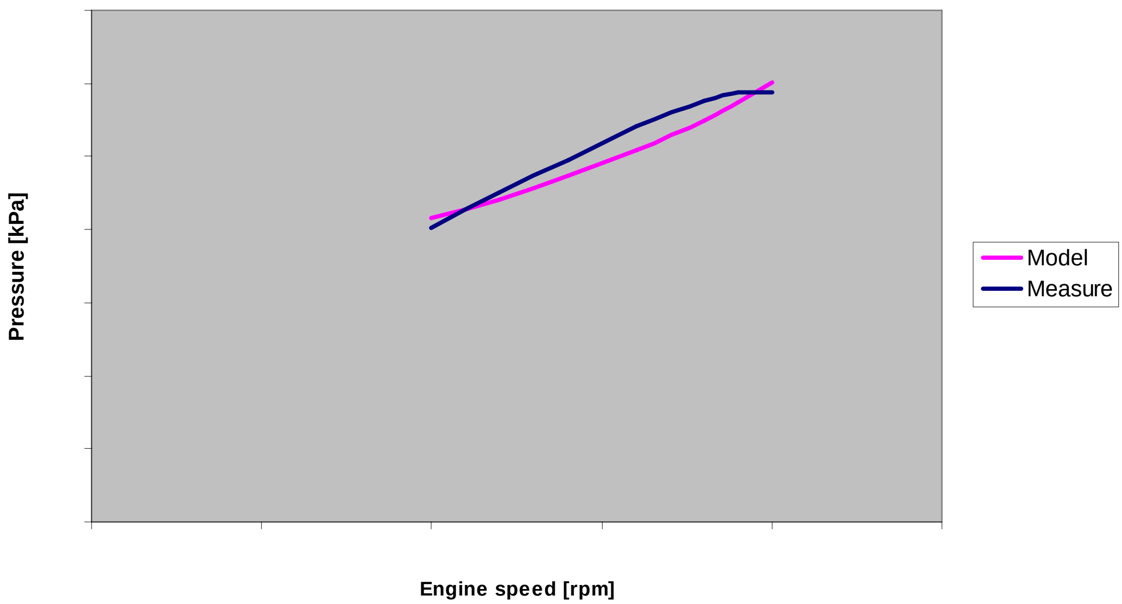

Figure 84. Pressure before radiator, Test 15.

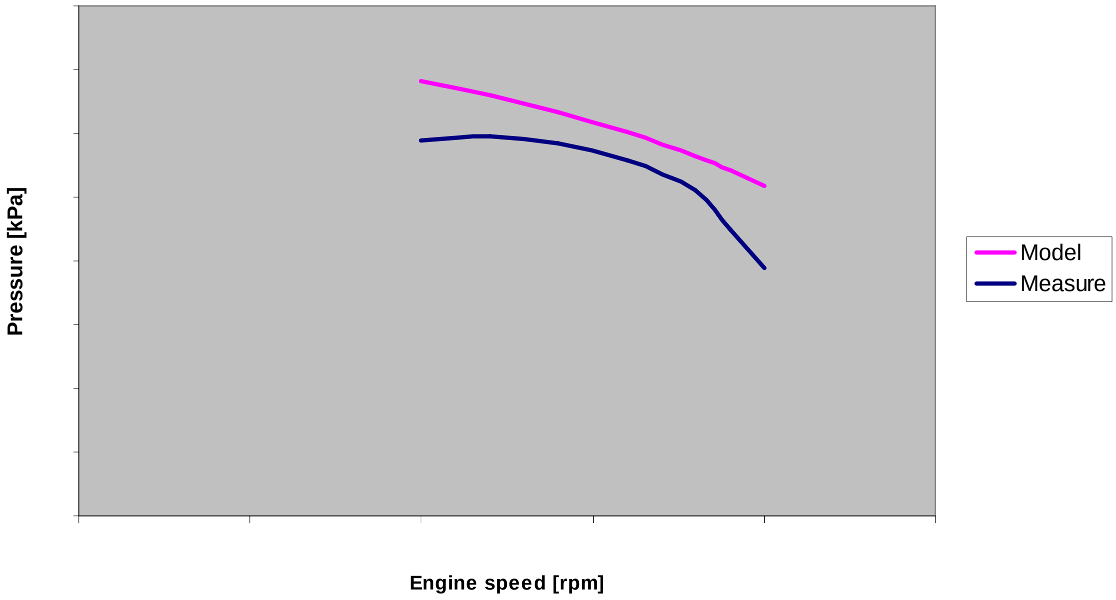

Figure 85. Pressure before coolant pump, Test 15.

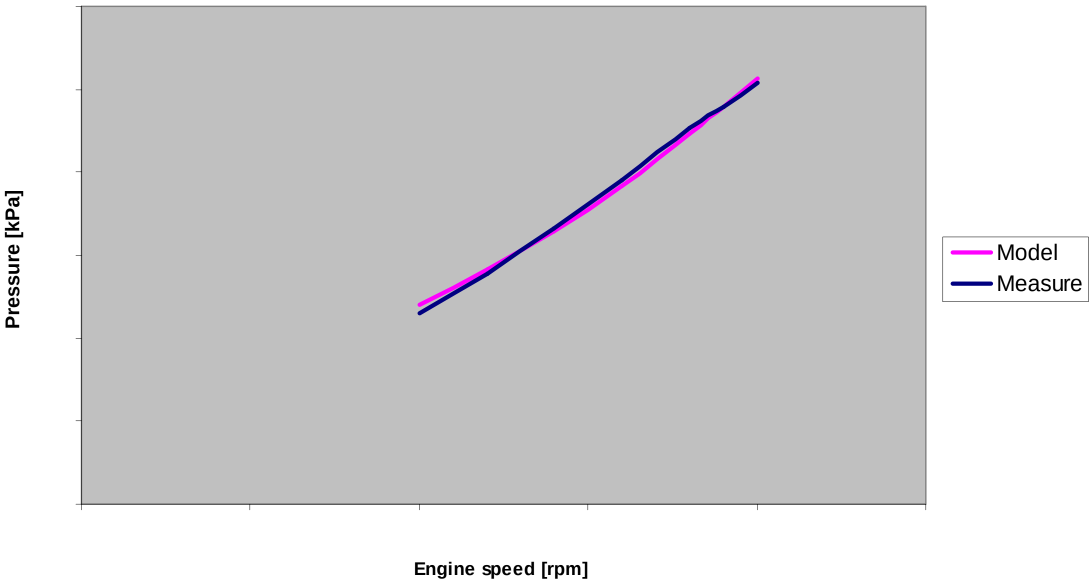

Figure 86. Pressure after coolant pump, Test 15.

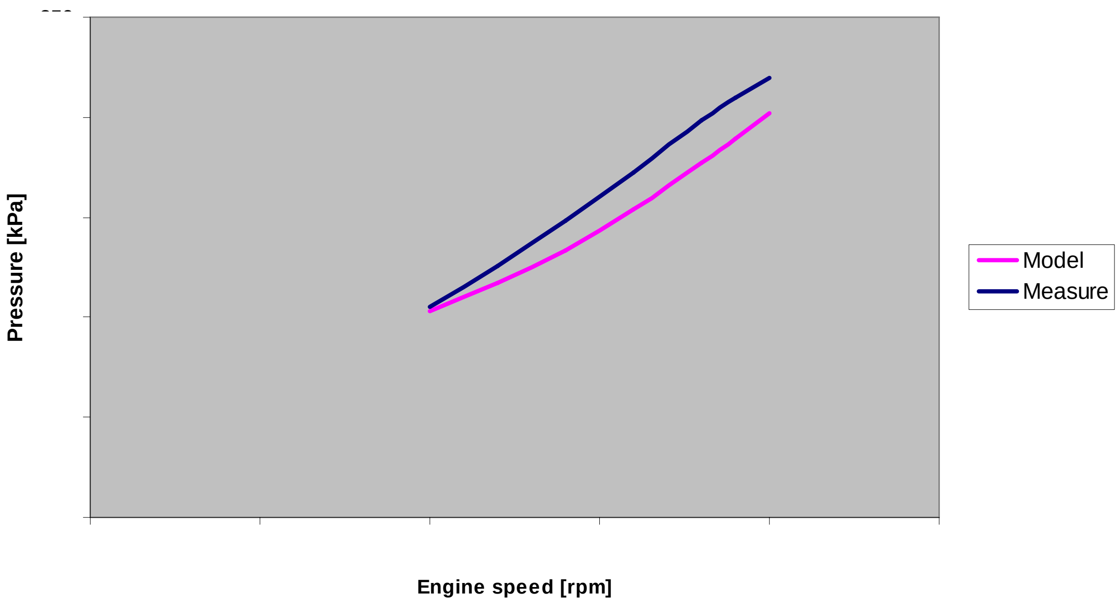

Figure 87. Pressure after oil cooler, Test 15.

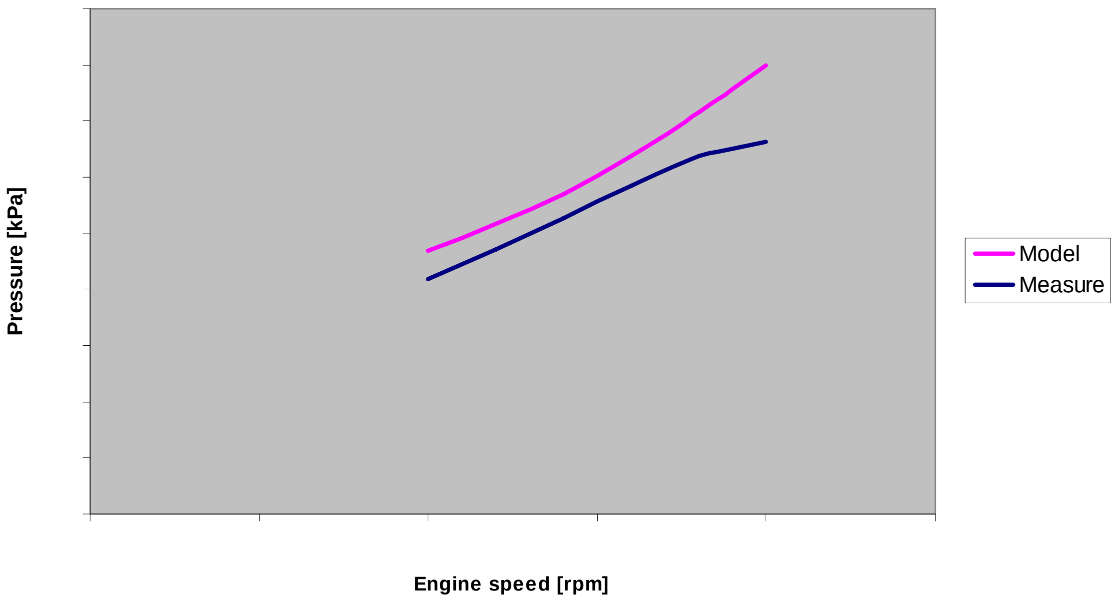

Figure 88. Pressure before transmission, Test 15.

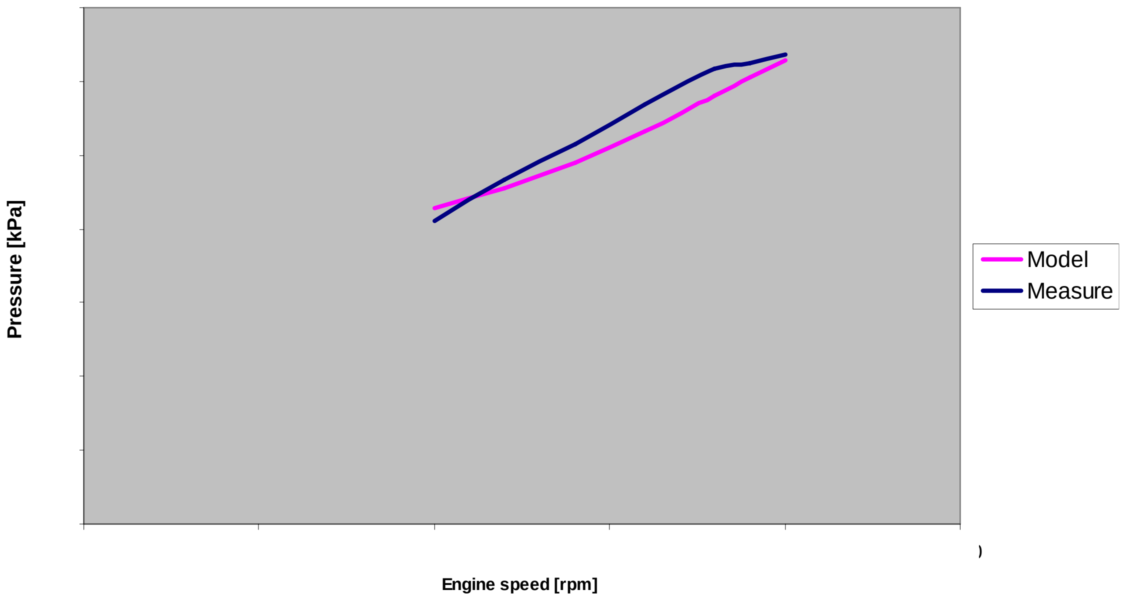

Figure 89. Pressure after transmission, Test 15.

Appendix 2

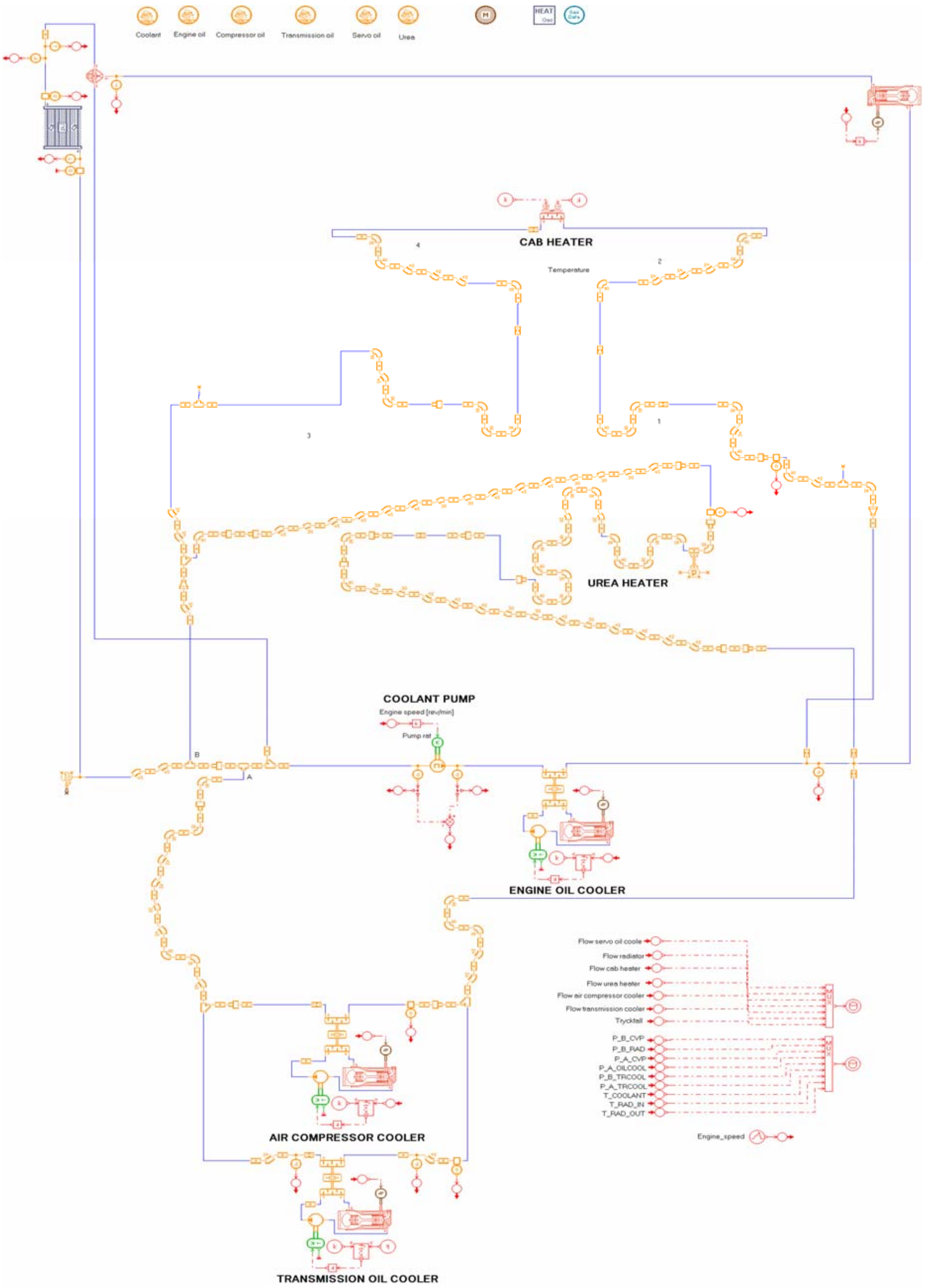

Figure 90. The AMESim model of test 11.

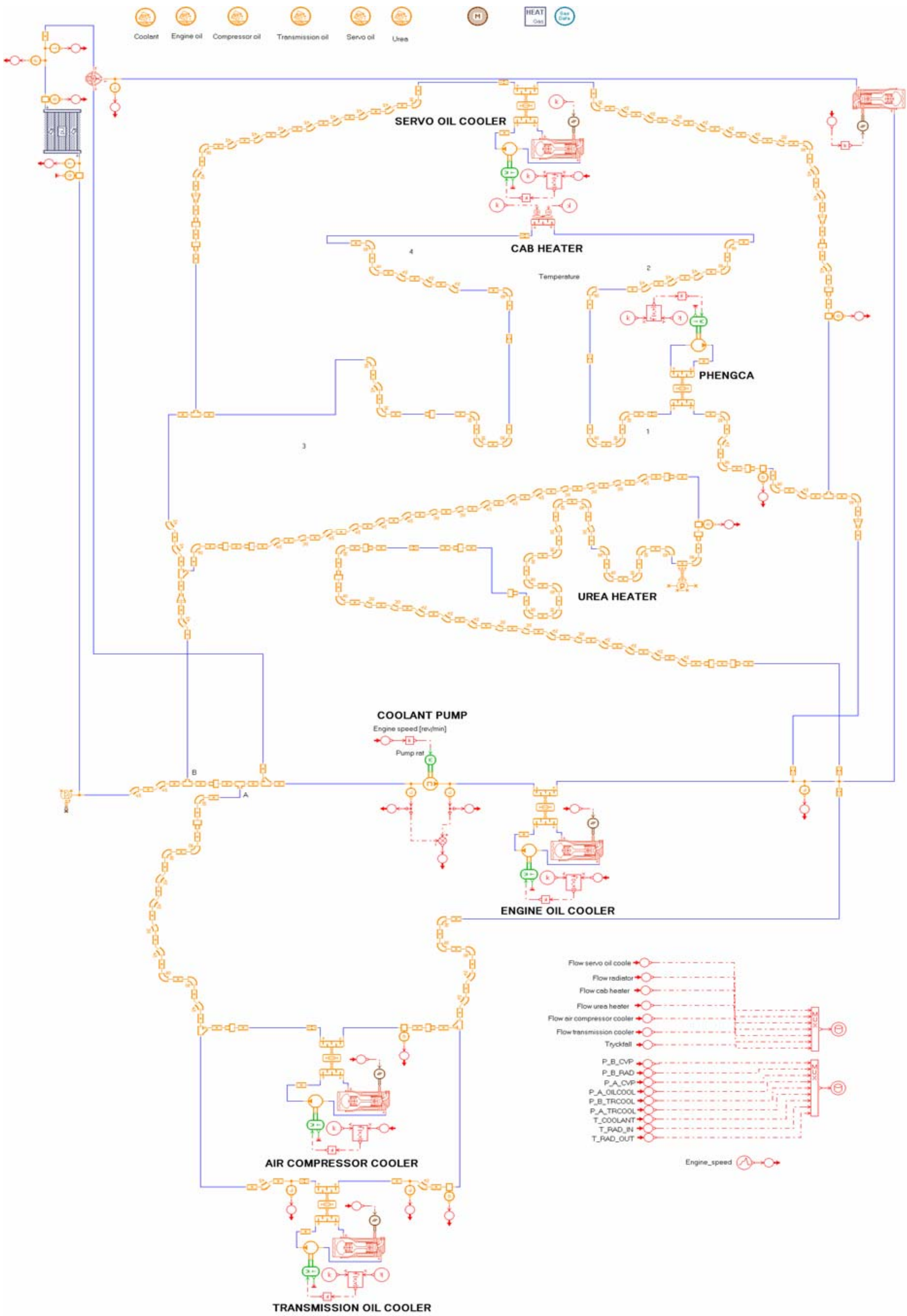

Figure 91. The AMESim model of test 13.

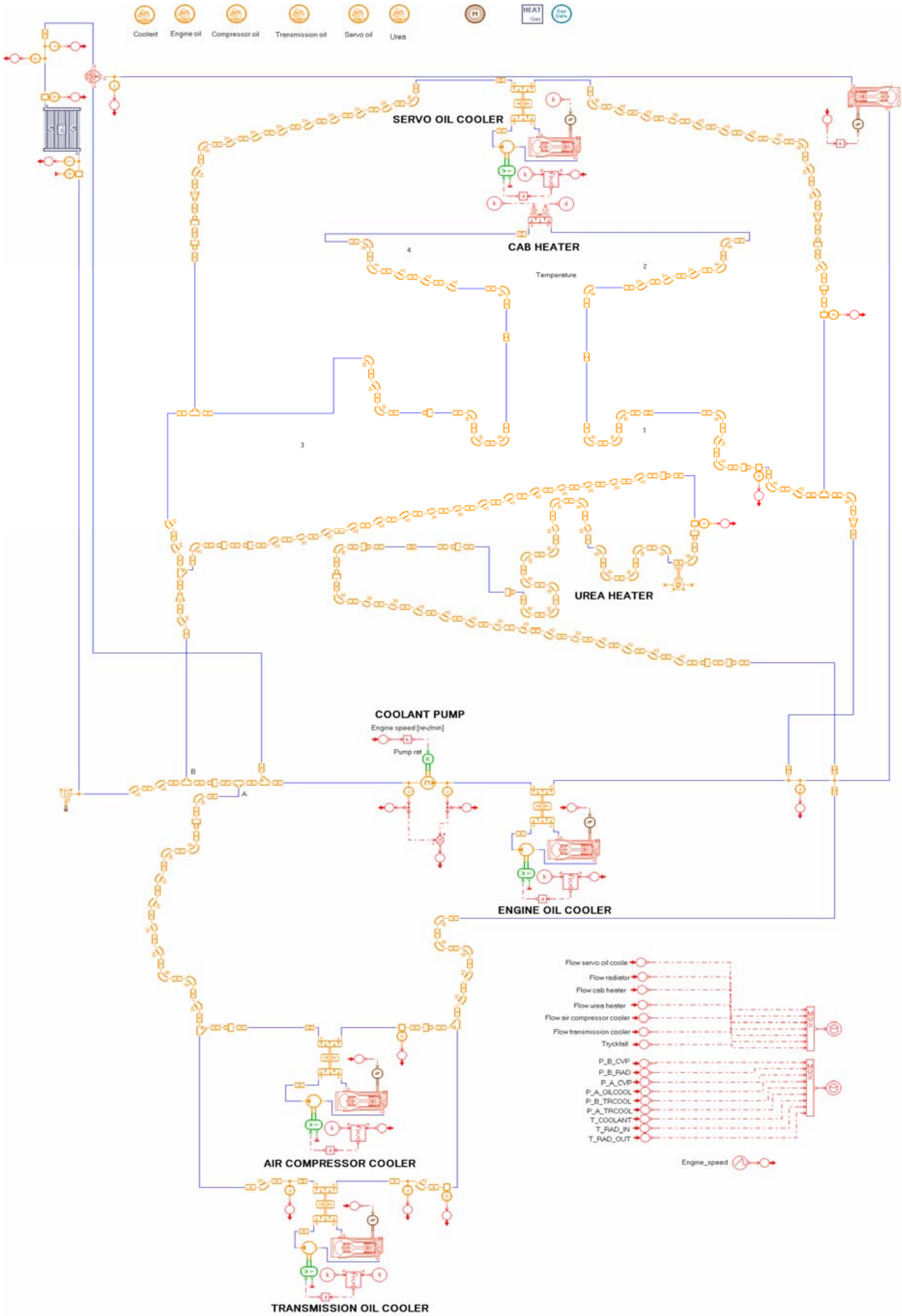

Figure 92. The AMESim model of test 15.

Appendix 3

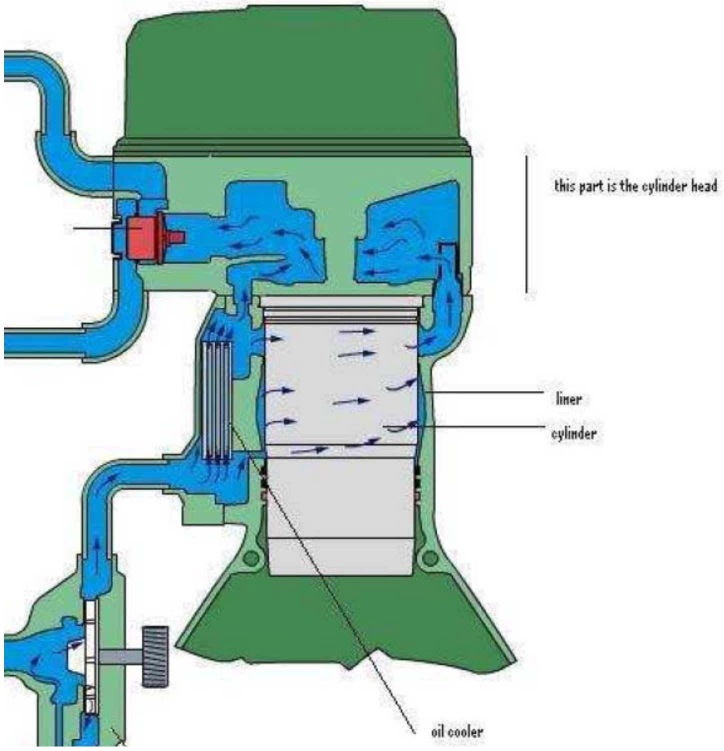

Figure 93. The flow through cylinder head and cylinder block.