ISSN 0280-5316
ISRN LUTFD2/TFRT--5767--SE

Engine Cooling & Model Verification

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May 2006

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Document name
MASTER THESIS

Date of issue
May 2006

Document Number
ISRNLUTFD2/TFRT--5767--SE

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Sponsoring organization

Title and subtitle
Engine Cooling & Model Verification (Motorkylning och modellverifiering)

Abstract
The automotive cooling system and thermal control strategy are important in order to reduce the pollution emission, extend the life of the components and reduce the fuel consumption. The scope of this thesis is to achieve a minimum model that works properly and a small library with the necessary components that the model could be made with. This will be the base for a future Engine Cooling library that covers all the components demanded by Daimler Chrysler.
This thesis is the description of the development of a basic Engine Cooling library and some test models. A simple engine cooling system is built and produces reasonable results. The given data to test the Engine Cooling model is for a complete model, for this reason the test of this library is made checking that the results are reasonable. Some simple components were tested with real data separately, but not in the complete Engine Cooling system. The thesis work is a good base for a complete and useful Engine Cooling library. The next steps to follow are to build a more complex and realistic engine and radiator models. Some components will have to be added like holes, pipes and different kinds of bends.

Keywords

Classification system and/or index terms (if any)

Supplementary bibliographical information

ISSN and key title
0280-5316

ISBN

Language
English

Number of pages
73

Recipient's notes

Security classification

Acknowledgements

I would like to specially thank the supervisor of this thesis Hubertus Tummescheit and also Jonas Eborn, for making this thesis possible with their deep knowledge in this field. I would also like to thank all the Modelon AB team for their help, specially Kristian Tuszynski and Faruk Selimovic.

Contents

1 Introduction

1.1 Background

Simulation is a useful tool to reduce time and money efforts in the competitive vehicle industry. More and more, the simulation is used during component and system design, achieving better vehicles. What is more, around 80% of the cost of a product is decided in the design.

The automotive cooling system and thermal control strategy are important in order to reduce the pollution emission, extend the life of the components and reduce the fuel consumption.

Daimler Chrysler vehicle department wants to improve their engine cooling design and develop tools to do it.

1.2 Purpose

The object of this thesis is to build a model of an already existing engine cooling system, and to create a library with different components, so they could change components and parameters in the model and see what happens in the system. The library will also be used to build new engine cooling models with different system layout. Thus, the components have to be reusable.

The modelling language used is called Modelica. This is an object-oriented equation-based programming language. The program used is Dymola, a simulation tool for modelling and simulation of integrated and complex systems, based Modelica language.

1.3 Scope

The scope of this thesis is to achieve a minimum model that works properly and a small library with the necessary components that the model could be made with. This will be the base for a future Engine Cooling library that covers all the components demanded by Daimler Chrysler.

2 Engine Cooling structure and operation

The conventional internal combustion engine cooling system consist of a thermostat, water pump, radiator, and engine block water jacket (refer to Figure 2.1). A copper impregnated wax enclosed thermostatic valve, or thermostat, self-regulates the coolant temperature by controlling flow through the radiator. A belt driven centrifugal pump traditionally circulates coolant through the system based on the engine speed. The engine cooling should be designed to maintain the engine's component temperatures within prescribed operating ranges during both steady-state and transient driving. The thermostat valve decides if the coolant goes through the radiator or not to control the temperature in the engine.

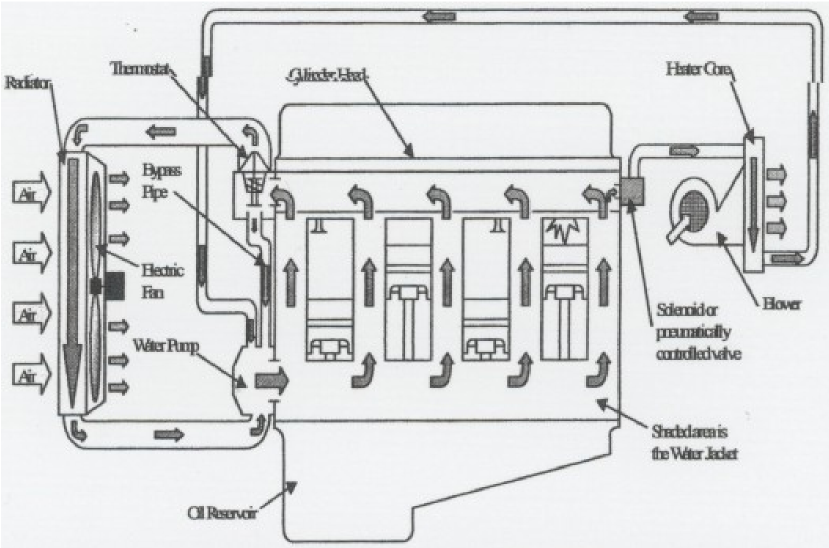

Figure 2.1: Spark ignition engine liquid cooling system configuration

A servo-motor valve and pump may be inserted into the vehicle's heating/cooling system to regulate the coolant flow with the engine control unit (Figure 2.2).

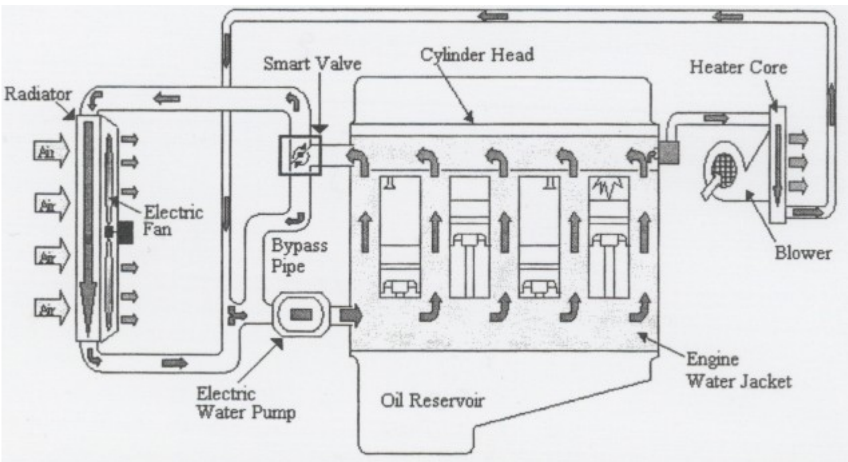

Figure 2.2: Advanced spark ignition engine thermal management system architecture

Some of the coolant heated in the engine block can be used to heat the cabin air, and then reintroduced to the circuit in the pump or somewhere else.

The spark ignition engine and the diesel engine have the same engine cooling system, but with minor modifications due to their differences.

In the diesel engines, the air is usually compressed and cooled before introduced into the cylinder. The exhaust gas is used to run a turbine that has the same shaft as the compressor, and some of this exhaust gas is introduced in to the combustion air to reduce the NOx emissions. This is done through an exhaust gas recirculation (EGR) valve.

This yields some structural differences like an air to air intercooler layer in the radiator, and has consequences in the air temperature of the radiator (Figure 2.3).

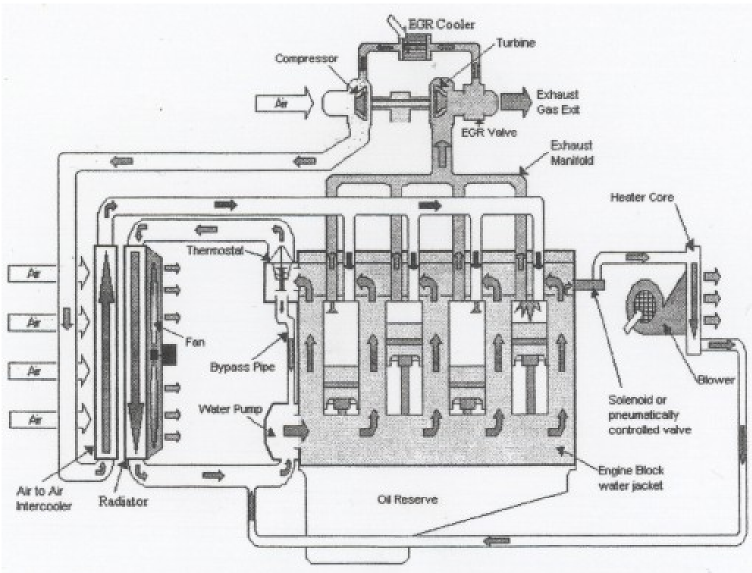

Figure 2.3: Diesel engine liquid cooling system configuration

Ideally the intercooler and the EGR cooler may use coolant provided that the fluid's inlet temperature is sufficiently low to permit the desired secondary fluid outlet temperature. Hence, multiple radiators are required for the diesel engine thermal management system. The ideal engine temperature may be controlled to a higher set point than required for these two coolers (EGR and Intercooler) as shown in figure 2.4. Thus, multiple radiators are required with individual cooling circuits to allow customization of the heat transfer rates in these circuits.

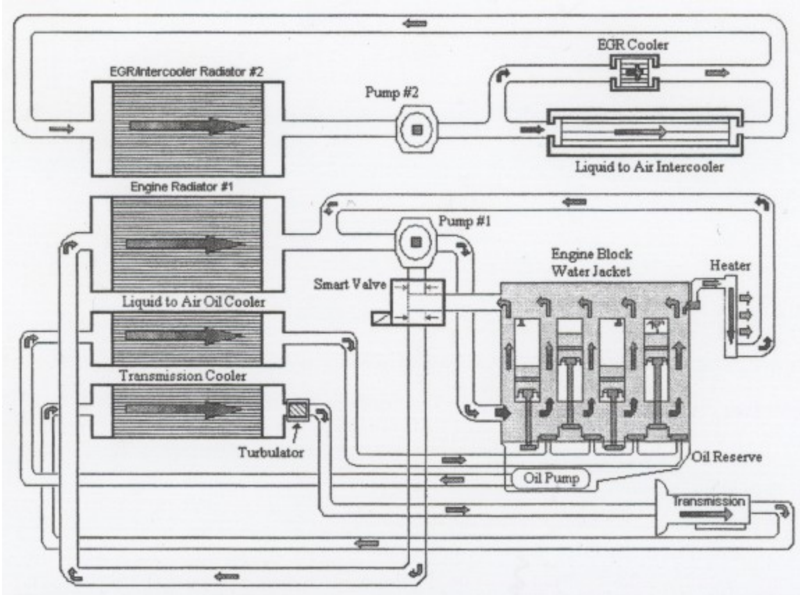

Figure 2.4: Advanced diesel engine thermal management system architecture

Advanced thermal system technologies can provide 1%-3% fuel economy improvements through lower parasitic losses, higher operating temperatures, reduced component temperature fluctuations, and alternative engine block monitoring temperatures.

NOx emission reductions are achieved by the EGR when exhaust gases are burned with the fresh air, and the addition of cooled EGR results in a 38% reduction of weighted brake specific NOx versus 32% for uncooled EGR in diesel engines. The introduction of a liquid-to-air heat exchanger should benefit the cooled EGR technology.

The automotive cooling system components and thermal control strategy have not been significantly redesigned to realize greater engine thermal efficiency.

(Gursaran D. Mathur (2004))

3 Library

3.1 Introduction

There are different ways to build a library. Design decisions have to be taken at an early stage. They determine the way to build its components and their usage.

This library is going to follow the structure of similar libraries like the CombiPlant library, used to model combined cycle power plants. This is a good example for its similarities to the engine cooling system. The main difference is that in the power plant there is two-phase media (steam and liquid), and this makes the models and the library more complicated.

This library has components that are models for the real components in the engine cooling system. The connector concept and the way how different fluid media are handled are taken from the Modelica-Fluid library. All the components have to have the same connectors and the same media in order to function together properly. It is important to decide the structure of the connectors before building the library because the models can be very different depending on the kind of connections they have.

It is also important to decide how tho build the components of the library. For example the library can be built thinking of the reuse of the component, models, functions of the library, so another future library can take some of them and use them with small modifications if it is necessary. The Engine Cooling library took some components of other libraries and it was easy because it uses the same structure that some other libraries use. Some models in the library are using functions instead of building everything inside the model because this functions could be used in a future modification of the library, or in other libraries.

3.2 Media and Connectors

The thermodynamic state of the fluid at any point is represented by two variables, e.g. pressure p and specific enthalpy h. Other thermodynamic quantities could be calculated from these two thermodynamic state variables.

It's important that a model for a device can use different media models. This can be achieved using the existing media replaceable package in the model, replacing the medium whenever it is needed. The device model is not influenced by the fact that the medium model is compressible or incompressible. This is possible because a tool will perform the necessary equation changes by index reduction when, e.g., an incompressible medium model is replaced by a compressible one in a device model.

It's very important to choose appropriate connectors to connect the different devices. The connector variables have to be selected in a way that connections between components fulfil the following balance equations:

- mass balance
- substance mass balance (for a medium with several substances)
- energy balance in the form of the "internal energy balance"

It's also important that the variables used in the connectors are non-redundant, so there are no restrictions in how components can be connected together. This leads to an unique selection of variables in the connectors:

- pressure **p**
- specific (mixing) enthalpy **h**
- independent (mixing) mass fractions **X** (empty vector for pure fluids)
- mass flow rate **m_dot**
- enthalpy flow rate **H_dot**
- independent substance mass flow rates **mX_dot**

In this design of the connectors, the mass and energy balance are fulfilled at connection points. The momentum balance is not taken into account in the connectors, so device couplings with a considerable amount of losses have to be modelled with a dedicated loss model (e. g. connection between pipes with different diameters).

The connections are defined in a way that some boundary conditions are needed. For example if we have a connection of two devices R and S (Figure 3.2.1).

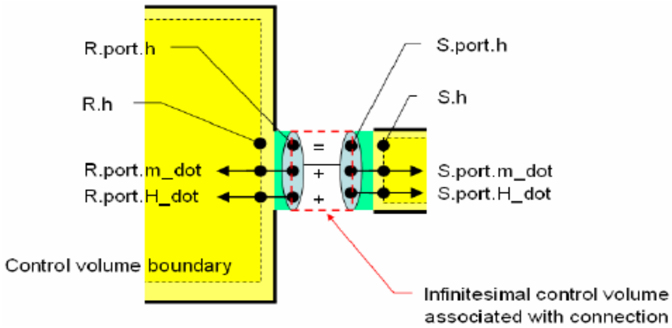

Figure 3.2.1: Details of device connection

R has the variable R.h but also the boundary condition R.port.h. The same for the variables m_dot and H_dot.

The variable h_{port} = R.port.h = S.port.h is introduced to simplify the notation. This can be done because in the connection R.port.h = S.port.h.

To allow reverse flow, the enthalpy flow rate into device R, H_dot_R is then dependent on the mass flow rate, m_dot_R as follows:

$H\_dot_R = m\_dot_R * h_{port}$ for $m\_dot_R > 0$

$H\_dot_R = m\_dot_R * h_R$ otherwise

This equation is in the model of the device R, and could be written in an if clause, but Modelica has its own operator to simplify this: semiLinear(...).

The function is used like this:

port.H_dot = semiLinear(port.m_dot, port.h, h);

The other side of the connection is written in the same way:

$H\_dot_S = m\_dot_S * h_{port}$ for $m\_dot_S > 0$

$H\_dot_S = m\_dot_S * h_S$ otherwise

And the same function will be used in the device code for component S.

In the connectors an infinitesimal control volume is assumed. The mass and energy balances in the connections are the following zero sum equations:

$0 = m\_dot_R + m\_dot_S$

$0 = H\_dot_R + H\_dot_S$

From these equations h_{port} can be found:

$h_{port} = h_S$	for $m\_dot_R > 0$
$h_{port} = h_R$	for $m\_dot_R < 0$
h_{port} = undefined	for $m\_dot_R = 0$

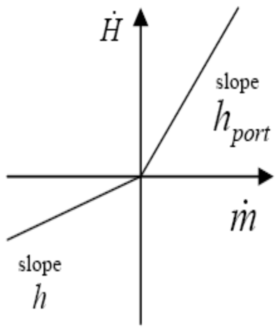

Figure 3.2.2: The semiLinear(...) function

It doesn't really mater which value h_{port} takes for $m\_dot_R = 0$, because H_dot_R and H_dot_S are well defined at zero, and then the dynamics of the system are independent of what value is chosen for h_{port} .

For more information about media and connectors Elmqvist et al. (2003) .

3.3 Models Usage

The components can be divided into two types: *Control volume* type and *flow model* type. The components that are modelled by storage of mass and energy in a volume are the components classified as control volume types. There is another type of component that has a small volume but high power densities like pumps, turbines, valves, short pipes and orifices. These components that have large changes in pressure and small energy storage are the flow model type. The dynamic mass and energy balances are modelled in control volume models and the quasi steady state or dynamic momentum balance is modelled in flow models.

The models have to be used in an alternating sequence of the two types for several reasons:

- Combining two volume models without a flow model in between leads to an index two DAE problem (for more information about high index DAE see Tummescheit, H. (2002)).
- Combining two flow models can lead to a non-linear system of equations. This is not desired because it reduces robustness unnecessarily.

If it is necessary two flow models can be connected directly by including a zero volume control volume between them.

(Tummescheit, H. (2002))

3.4 Library Structure

The library is a main package called EngineCooling. Inside there are different packages that determine the structure of the library (Figure 3.4.1).

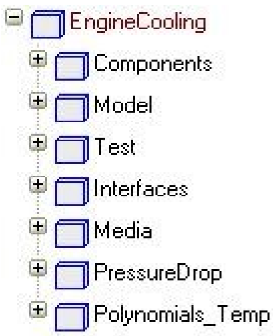

Figure 3.4.1: Library packages

The **Components package** contains all components intended for drag and drop usage of the library (Figure 3.4.2).

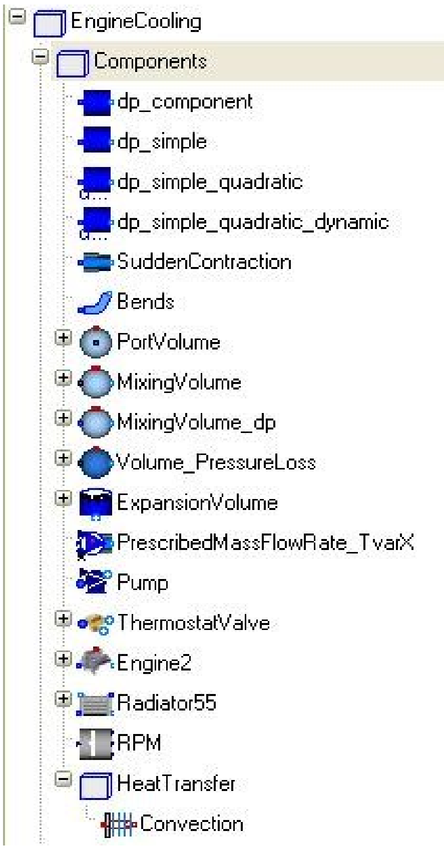

Figure 3.4.2: Components package

This is the main package used to build system models. The user can just drag and drop the components in the engine cooling models from here. There is another package inside called HeatTransfer. In this package there are the components used for the heat transfer.

In the **Model package** the engine cooling models are stored. The user of the library will find some engine cooling system examples. These examples can be used for the user as a base to build variations models of the engine cooling system.

The **Test package** contains functional tests of the components. These are simple models to test the components work. It is useful for the user to have some simple example models with reasonable parameter settings.

The connectors for the coolant side and the thermal side are stored in the **Interfaces package**. This package contains the bases of the connectors and the base classes for standard connector configurations of connections, for example with only one port, two ports, heat transfer port, etc. The component models extend and modify these base classes.

The **Media package** is where the media model is stored. The engine cooling uses one special cooling fluid (water with glycol) but we can use any fluids from the Modelica.Media package, where there are different models for fluids.

All the functions and models needed only for pressure drop correlations that the components use are stored in the **PressureDrop package**. Some of the components have their base.

The **Polynomials_Temp package** is stored in the EngineCooling library. This package contains functions to operate on polynomials. One example is the data fitting used in the pressure loss component where a function in this package is used to calculate the coefficients of the polynomial that fits optimal in a least squares sense with respect to given data points.

4 Component Models

4.1 Introduction

With the data provided by Daimler Chrysler we have to decide how the models should be built and how to structure them.

The model that will be built is a basic cooling cycle. This basic cycle needs a pump to pump the refrigerant through the circuit. Pressure drop components are needed to create a differential pressure in the pump and obtain a stable working point. Two components that add and remove heat from the cycle are needed, these are the radiator and the engine models. When there is a closed cycle, an expansion tank is needed to compensate for the volume change of the cooling fluid. An important component is the thermostat valve that regulates the coolant flow that flows to the radiator and the coolant that flows through the bypass. There is also a pressure drop that is dynamic to solve numerical problems when the loop is closed.

This are the basic components to build an engine cooling system. This basic components and the basic cycle can be the bases for a more advanced cycle with more components.

4.2 Bend

The Bend is the component that models the pressure loss in a smoothly bended pipe with circular section.

The component could be easily adapted to a non circular section.

The source for the model is from Idelchik, I.E. (1996) the component is modelled after this reference.

Splines are used to interpolate the data given in various tables in Idelchik,I.E. (1996).The splines can be constructed in different ways and a good one for this data had to be chosen.

The data is given in a table with two inputs and one output:

input - Radius of the pipe's curvature divided by the diameter of the pipe (R_0/D_0)

input - Reynolds number (Re)

output - λ coefficient, called friction coefficient (the pressure loss is computed from this coefficient)

$$\frac{\Delta p}{\rho w_0^2/2} = 0.0175\,\lambda_{el}\,\delta\,\frac{R_0}{D_h}$$

δ is the curve angle in degrees and 0.0175 is a factor to convert the angle from degrees to radians. The density of the fluid is represented by ρ and its velocity by w.

For each R_0/D_0 a spline was built with the different Re and λ values. In between the R_0/D_0 numbers, an interpolation of the λ value is done in the model using the two closest splines.

For the spline calculation the λ_2 transformation was applied to the data. This transformation consists in multipling the λ axe by the square of the Reynolds number.

$$\lambda_2 = \lambda \cdot Re \cdot |Re|$$

This is a good transformation because the infinite asymptote at Re = 0 (Figure 4.2.1) disappears and it converges to zero (Figure 4.2.2).

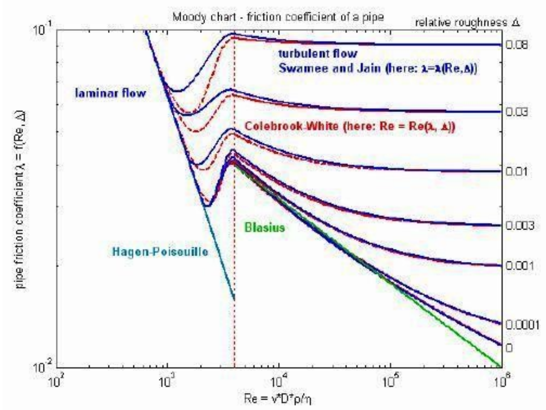

Figure 4.2.1: Moody Chart: $\lg(\lambda) = f(\lg(Re), \Delta)$

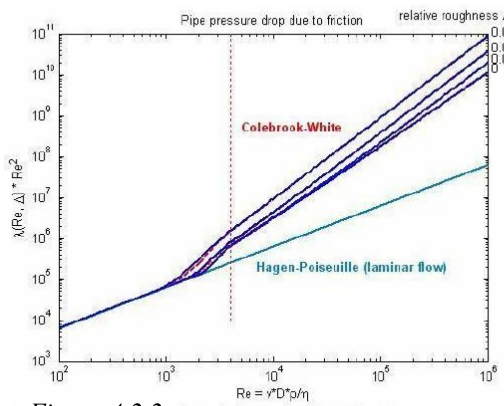

Figure 4.2.2: $\lambda_2 = \lambda_2(Re, \Delta) = \lambda \cdot Re \cdot |Re|$.
(x-axis: lg(Re), y-axis: $\lg(\lambda_2)$)

For more information about the λ_2 transformation see Elmqvuist, H. et al. (2003).

The spline coefficients are calculated and stored in a record. A function to evaluate the spline accesses the data to calculate the values of λ.

The bend component uses the CoolantTransport partial model and the function that calculates the λ value.

The different kinds of parametrizations to obtain the splines are:

- ChordLength: the parameters are chosen proportional to the distances of the points.
- Centripetal: This is often useful to minimise artefacts if there are irregularities in the data. The idea of the centripetal parametrization is to minimize the centripetal force of an object moving on the curve described by the spline.
- Equidistant: This is normally not recommended. The parameters are chosen equidistant.
- Foley: for computing the parameter the distances and the angle formed by the points are needed.
- Angular: this parametrization is like the Foley parametrization (distances and the angle are needed).
- AreaBased: the parameters are chosen proportional to the area of the parallelograms formed by the points.

For the low Reynolds zone, a straight line can be assumed in the logarithmic space that corresponds to:

$$\lambda = \frac{64}{Re}$$

This zone is needed for low mass flows in our model and we want it to work also in this region. Some points were added at the low Reynolds region to make the splines go to zero. This was done in the λ_2 transformation because it was easier to work here.

This modification of the splines is also useful if we would like to work in reverse flow conditions.

In order to improve smoothness of the final spline some modifications in the data were needed, in particular in the laminar-turbulent transition region.

Final splines plotted against λ_2:

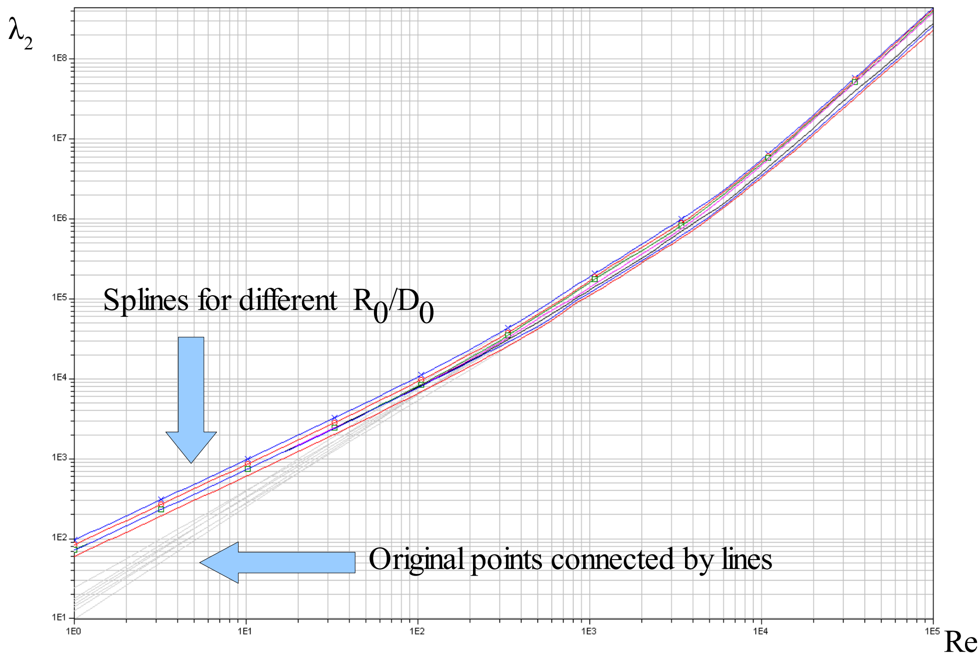

Figure 4.2.1: Plot of the modified splines and the original data

Detail of the low region but without the logarithmic scale:

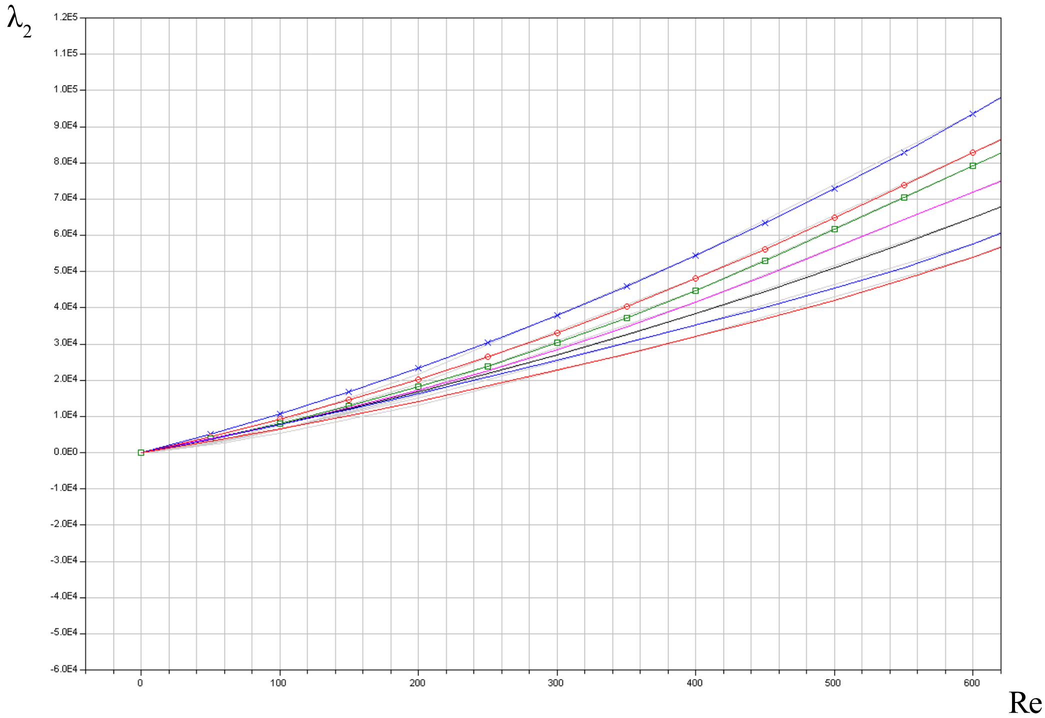

Figure 4.2.2: Detail of the splines in the low Reynolds region without the logarithmic scale

This component was done before having the data from the system that has to be modelled and it has to be expanded for $R_0/D_0 < 3$, because most of the components are $R_0/D_0 < 3$. The data for model $R_0/D_0 < 3$ ratios is given in Idelchik, I.E.

4.3 Sudden Contraction

The sudden contraction is a component that models the pressure loss in a sudden change of section for a circular section pipe, from a bigger section to a smaller one.

The formulation from Idelchik, I.E. (1996), is used to model the sudden contraction component.

There are different equations for the regions with different Reynolds number. The problem with this is that they are not continuous and this is a problem for the model because this can give simulation problems.

The basic formula is:

$$\zeta \equiv \frac{\Delta p}{\rho w_0^2 / 2}$$

Where ζ is the coefficient of fluid resistance (pressure loss coefficient).

This coefficient can be split into two different ones, the coefficient of local fluid resistance (ζloc) and the coefficient of friction resistance of the segment (ζfr):

$$\zeta = \zeta_{loc} + \zeta_{fr}$$

ζ_{loc} is the friction generated due to the flow through the duct. This is continuous and depends on λ (the friction coefficient of the fluid).

ζ_{fr} is the added friction due to the contraction and its consequences (e.g. reduction of the diameter due to phenomena called 'vena constricta'). This is discontinuous for each region.

These discontinuities can be seen in this plot of the ζ_{fr} as a function of the Reynolds number (the different lines are for the different values of the two section area relation F_0/F_1):

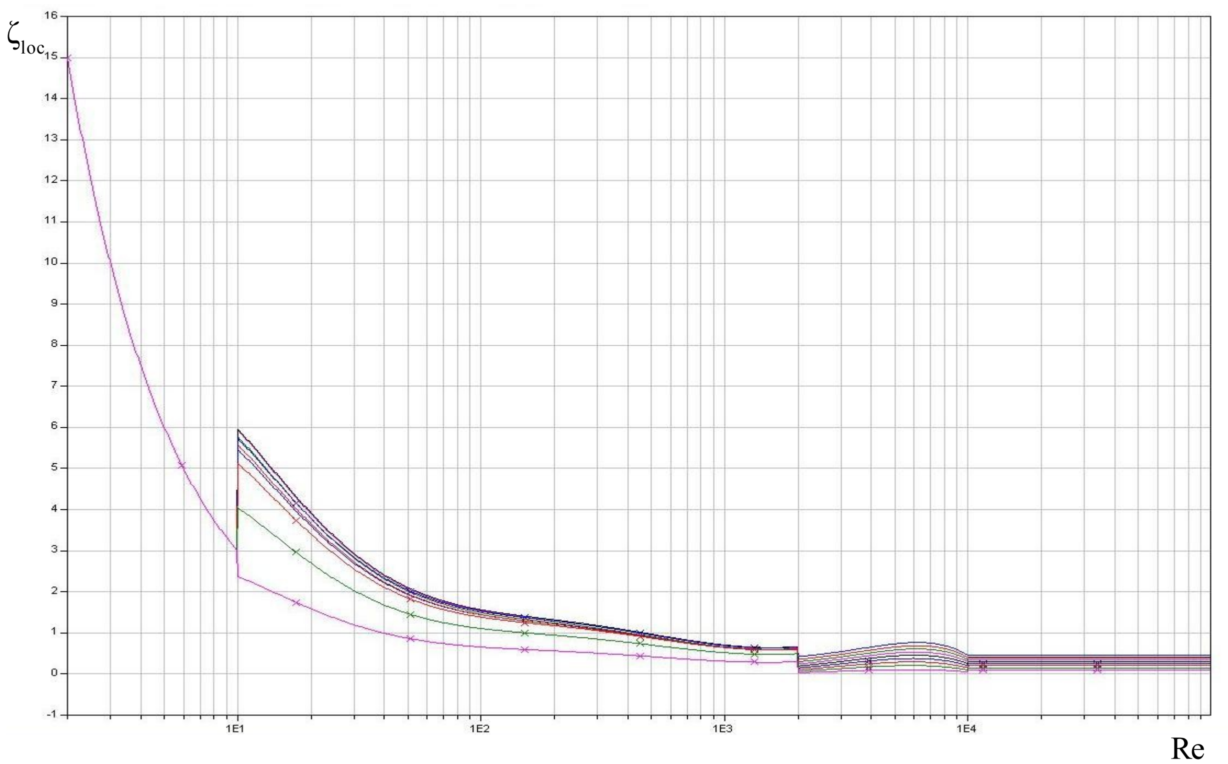

Figure 4.3.1: Discontinuities in the ζ_{loc} splines

For this reason the data from the table in page 218 of Idelchik, I.E. (1996) is used to model the ζ_{loc}

continuously.

The same procedure as in the Bend component had to be done and build the splines from the data.

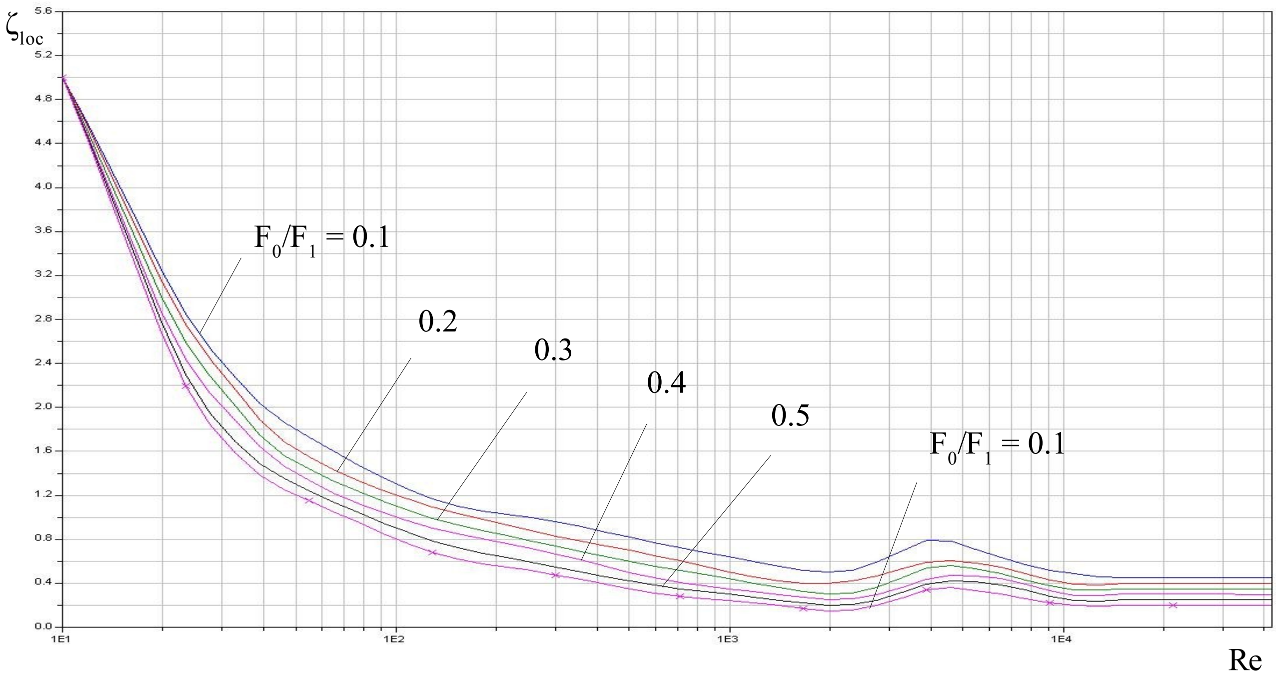

Figure 4.3.2: Continuous ζ_{loc} splines

To obtain a continuously varying pressure loss the λ coefficient has to go to 0 at low Reynolds numbers. Because of this a spline was constructed at low Re < 10 that make the splines go to zero in a smooth way. We added some points that follows the slope of the original data, taking high values at low Reynolds numbers and then going progressively to zero when the Reynolds number is close enough to zero.

To recalculate the splines easily when data was added, a function to calculate the spline coefficients was built.

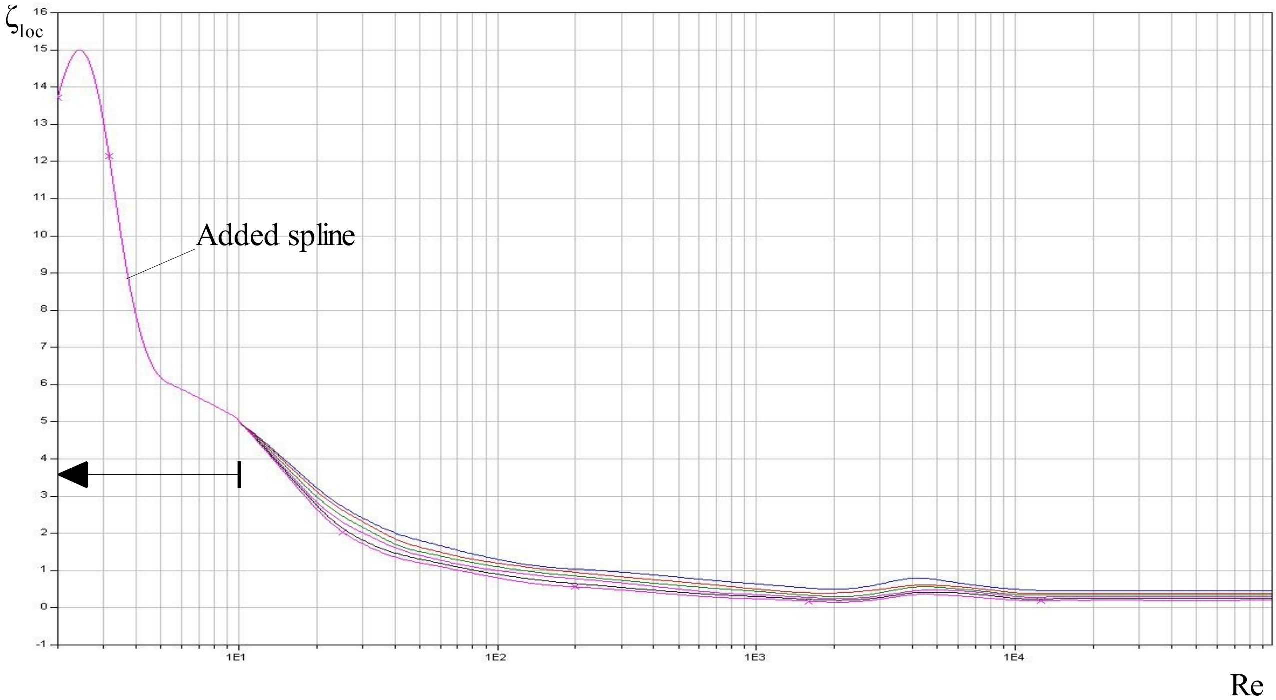

Figure 4.3.3: Smooth extension of the ζ_{loc} splines for the low Reynolds number region

The low Reynolds region in more detail:

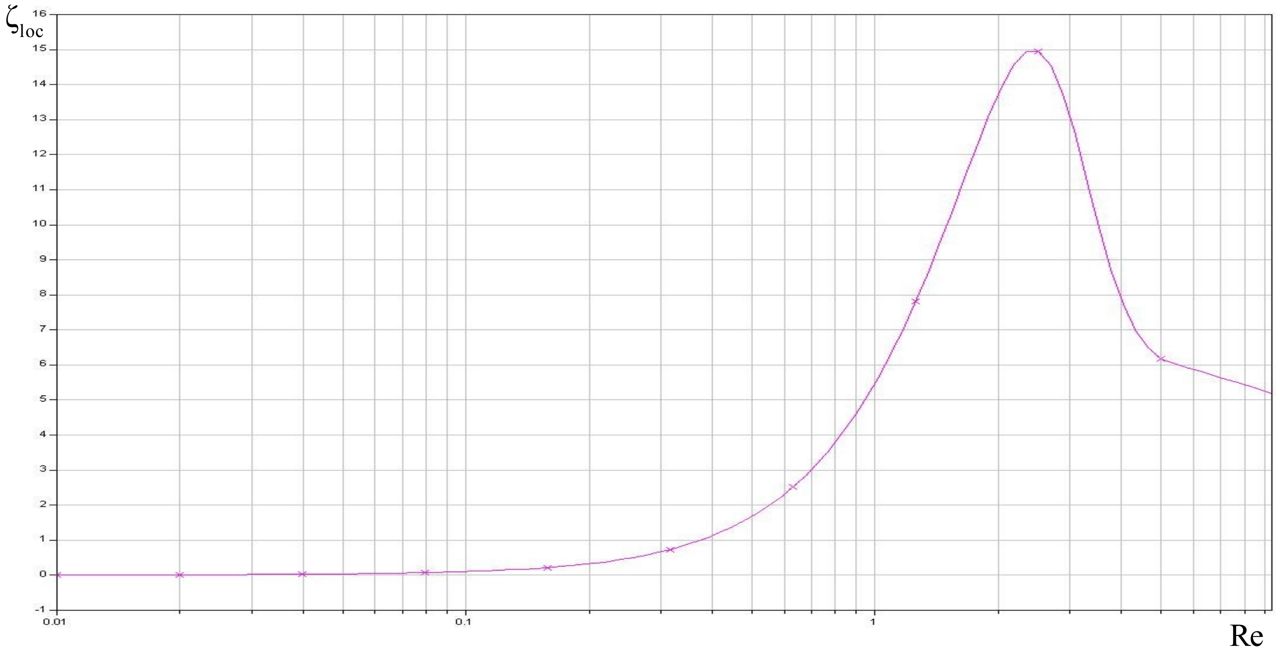

Figure 4.3.4: Detail of the smooth extension of the ζ_{loc} splines for the low Reynolds number region

The ζ_{loc} depends on λ. The λ (the friction coefficient of the fluid) calculation of the fluid will be taken from another function already done. Because of this it is not used yet, waiting for the function that will be taken from another library. Now the sudden contraction only calculates the losses generated for the contraction, not for the pipe.

4.4 Thermostat Valve

This component has one input and two outputs. Depending on the temperature of the coolant it will open or close making the fluid pass through the radiator. This is the control of the temperature of the engine. It's a very important component. When the engine starts, it is cool and needs to increase its temperature fast. The thermostat valve is closed so the coolant does not go into the radiator and keeps all the heat coming from the engine.

The thermostat valve is a component that could be modeled in a different ways. Since data that describes the behaviour of the thermostat valve is provided, the component will be modeled as proceeds.

The data provided is described in three graphs:

a) Figure 4.4.1 shows the data that describes the position of the valve as a function of the temperature. The test is done by first increasing the temperature from where the valve is completely closed until it is entirely opened, and then decreasing the temperature until the valve is closed again.

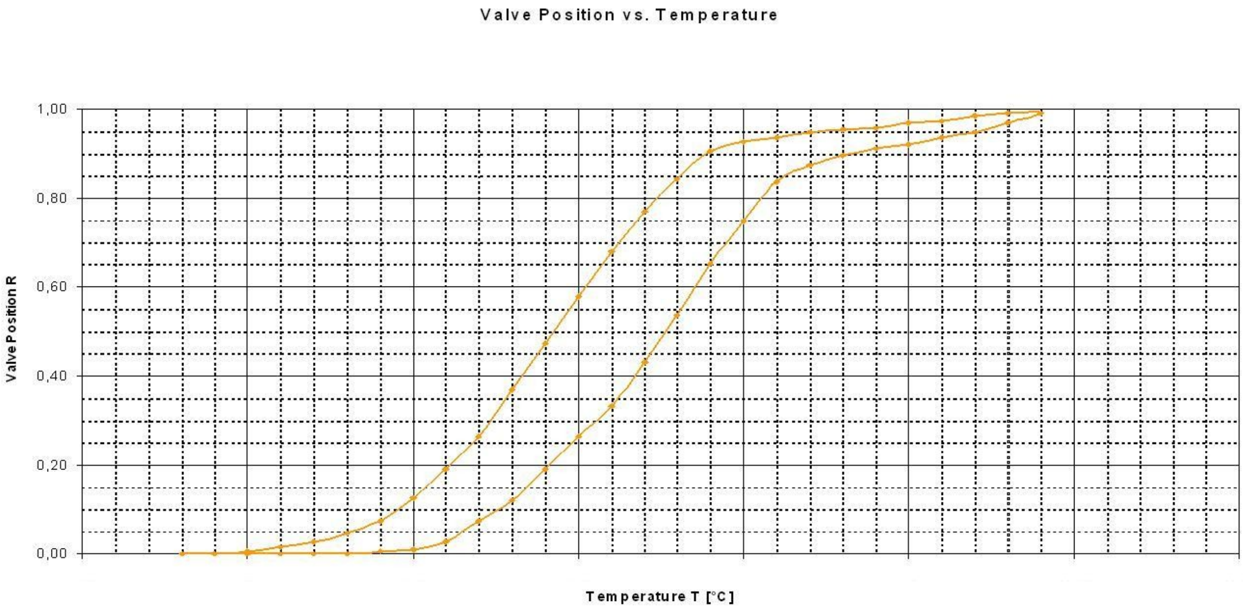

Figure 4.4.1: Thermostat valve position as a function of the temperature

The curves show that there is hysteresis. This component has wax inside that melts when the temperature is high and solidifies at low temperature. This produces the change of volume that makes the valve open and close. The hysteresis happens because of the nucleation in the solidification process and due to mechanical and fluidic friction.

b) Figure 4.4.2 shows the pressure loss of the radiator depending on the flow to the radiator and the position of the valve.

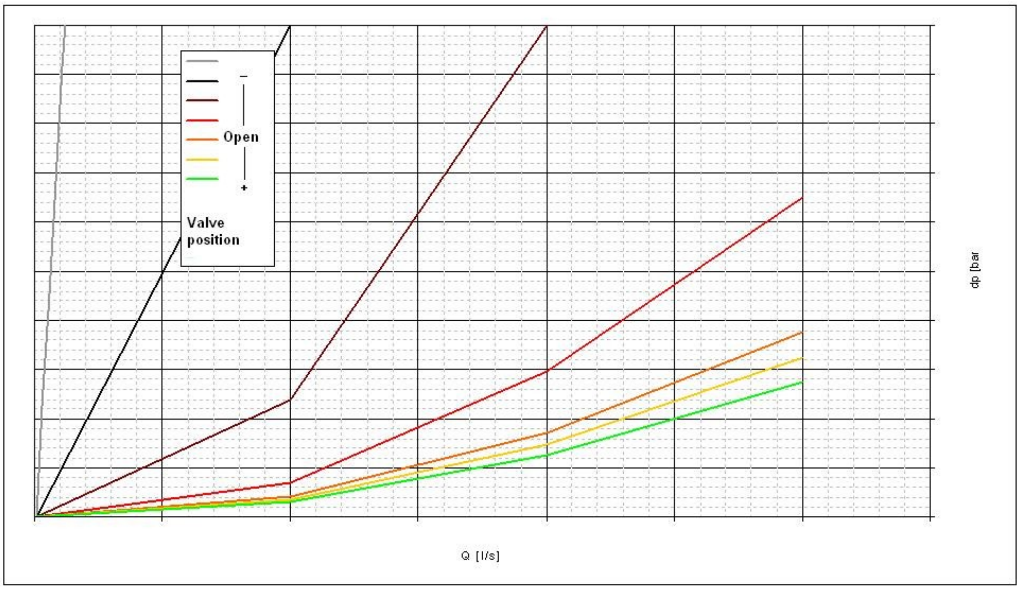

Figure 4.4.2: Pressure drop in the radiator branch of the thermostat, as a function of the flow and the valve position.

c) Figure 4.4.3 shows the pressure loss of the bypass depending on the flow to the bypass and the position of the valve.

   The bypass drives the coolant to the engine without going to the fan where gets cold.

Figure 4.4.3: Pressure drop in the bypass branch of the thermostat, as a function of the flow and the valve position.

This will be a component that can be used for different thermostat valves. Then the hysteresis points will be provided as an input, but the model will contain the points of this one as a default values.

These data points that describe the position as a function of the temperature are going to be converted to a continuous function with splines. This is important because some simulation problems occur if we do not have the continuous derivative in every point.

The splines construction has to be done carefully because the splines can have some strange waves like in figure 4.4.4.

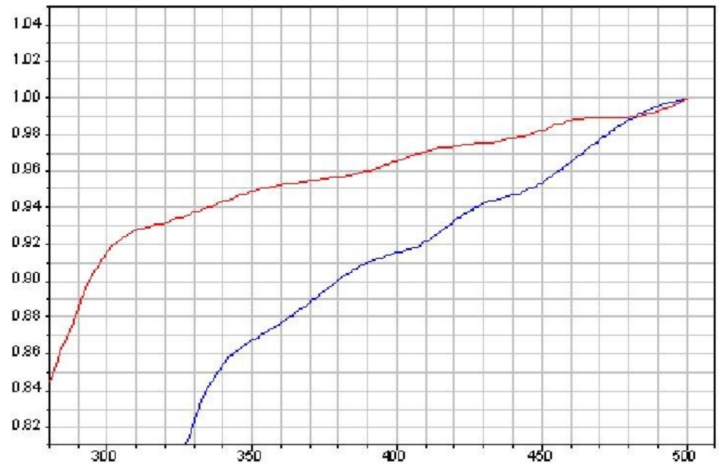

Figure 4.4.4: Spline waves and lines cross in the thermostat valve position

The spline curves also intersect and this can lead to some errors and simulation problems. This is fixed by using appropriate interpolation points. The users have to make sure that the spline looks reasonable, and smooth to get good simulation results. For this reason there is a model to plot the splines built with the data and check the shape.

One of the problems was that at the ends of the splines some points after the limit of the spline were needed for Dymola to compute the derivative at every point. But it can't have a jump, so when the spline finishes they can't just take a constant value. This discontinuity gives some problems with the solver. The solution is that the function calculates automatically the slope automatically at the end and builds a line with this slope, then there are no discontinuities.

The thermostat valve uses a function where position is calculated from the temperature. As an input we have to give the temperature, the two extreme temperatures of the spline where they finish, and the data that defines the spline. The output of the function will be a real between 0 and 1 that defines the valve position. There are two functions, one for the upward branch, when the valve is opening, and another one for the downward branch, where the valve is closing.

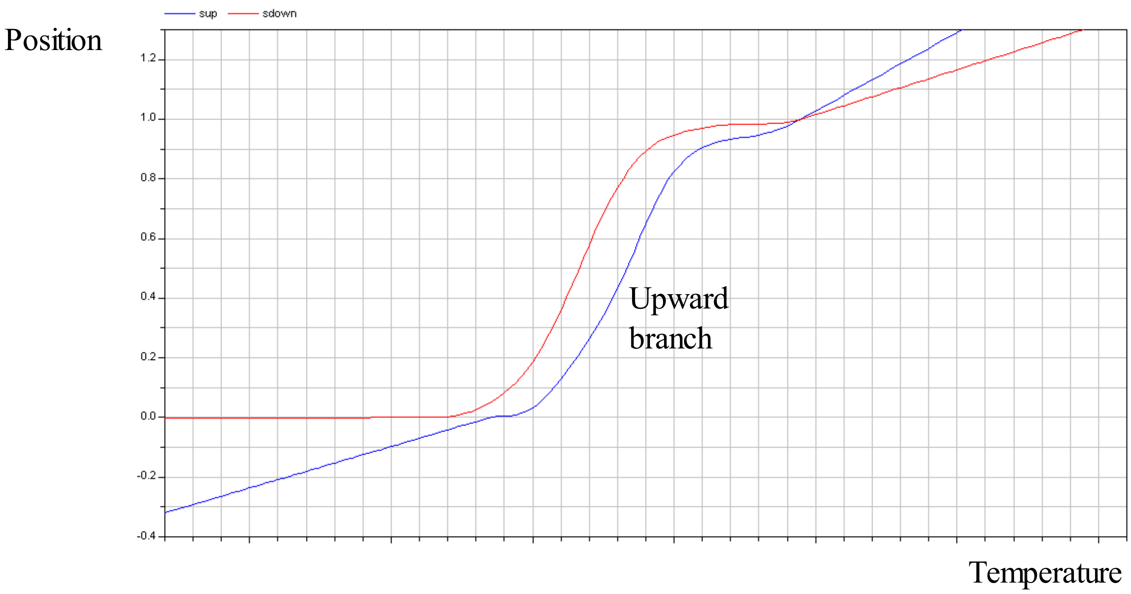

Figure 4.4.5: Valve position splines, and their smooth elongations.

A model to draw the splines and its extensions in an easy way was built. It's called

Check_Draw_spline_shape.

For the construction of the valve position model, hybrid modelling is used. Models combining Discrete Event Dynamic Systems (DEDS) and Continuous Variable Dynamic Systems (CVDS) are called hybrid models.

Two different models for this component were built. The first one uses the *finite state machine*, and it worked but due to its complicated structure we had to be very careful when we joined it with the other components. There was a second idea to build the component and this was more robust due to its simplicity.

- First model: *Finite state machine:*

A good way to build a model is to try to find an existing model that works properly, from where to adapt some ideas. The clutch model in the Modelica library is a good example because it has some similarities in behaviour and is a hybrid model with a similar structure.

The model has the *finite state machine* to know in each moment the situation and the last instant. A graph with the states and the conditions to change state is done to make the model clearer. See figure 4.4.6.

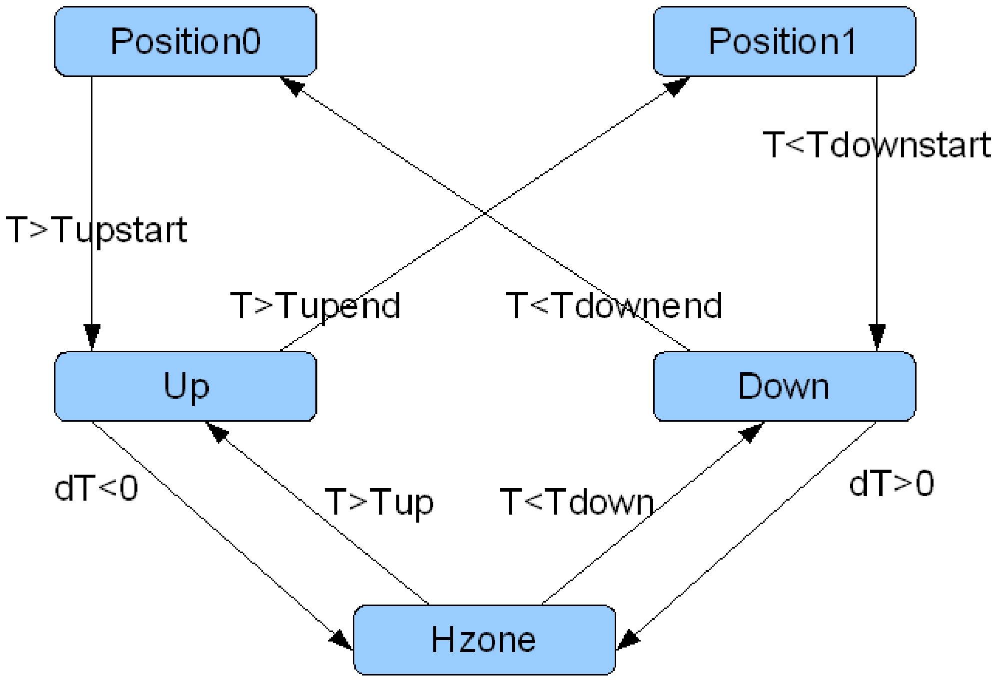

Figure 4.4.6: Finite state machine diagram for the thermostat valve

States:

Position0: Valve complete closed

Position1: Valve complete opened

Up: Valve opening

Down: Valve closing

Hzone: Valve in hysteresis zone between Up and Down states

State transitions:

T: Temperature

Tupstart: Temperature when the valve begins to open

Tupend: Temperature when the valve is complete opened

Tdownstart: Temperature when the valve begins to close

Tdownend: Temperature when the valve is complete closed

Tup: Up temperature to get out of the Hzone

Tdown: Lower temperature to get out of the Hzone

dT: Derivative of the temperature

The idea of this model is to have a finite state machine that decides in which mode the system is in every moment depending on the last mode and some conditions to go from one mode to another.

– Second model:

This second model also uses hybrid modelling. The difference is that it does not use a *finite state machine,* although there are some similarities. There are three main parts of the model:

-This model uses a boolean flag to know if the valve is opening or closing. The condition is that when the derivative of the temperature is positive the flag is true which means that the valve is opening. When the derivative of the temperature is negative the flag is false and the valve is closing.

-There is a “when clause” that keeps the value of the valve position at a discrete value after each change of the flag.

-The third part of the model is the assignment of the position variable. There is an “if clause” that determines which equation computes the position of the valve. In this “if clause”, when the temperature of the fluid is higher or lower that certain values, the position is set to 0 or 1 (closed or opened). If the temperature is in between those values the flag is checked. If the flag is true, that means that the valve is opening, the model checks if the position of the valve in the upwards spline is higher that the value computed in the “when clause”, when the flag changed. The valve position will be the maximum of this two. The same, but using the minimum and the downwards spline, occurs when the flag is false. The example is illustrated in figure 4.4.7.

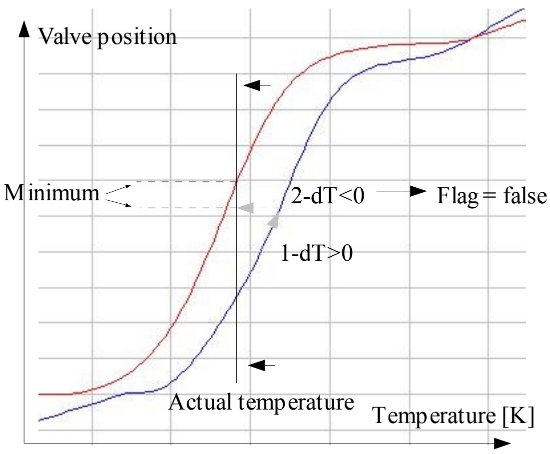

Figure 4.4.7: Example of the model behaviour

The simplicity of the equations of the model are an advantage compared to the finite state machine built before. The simpler way of modelling the thermostat valve is advantageous in particular for users who want to understand the model, for this reason this is the model used in the library.

The thermostat model does not only calculates the valve position. In each of the two branches a pressure loss is computed depending on the position of the thermostat valve. This model is called: *ThermostatValve_branches.* The model uses a variation of the CombiTable2D in Modelica.Blocks.Tables. This model interpolates the values between the data table in data obtained for a realistic component. The model also calculates a continuous first derivative..

For the connections, a three pin connector is needed with one input and two outputs. The final model extends a partial model with only connectors. It also uses the valve position model to compute the flow through each branch, and the pressure loss model that gives the pressure loss in each branch (Figure 4.4.8).

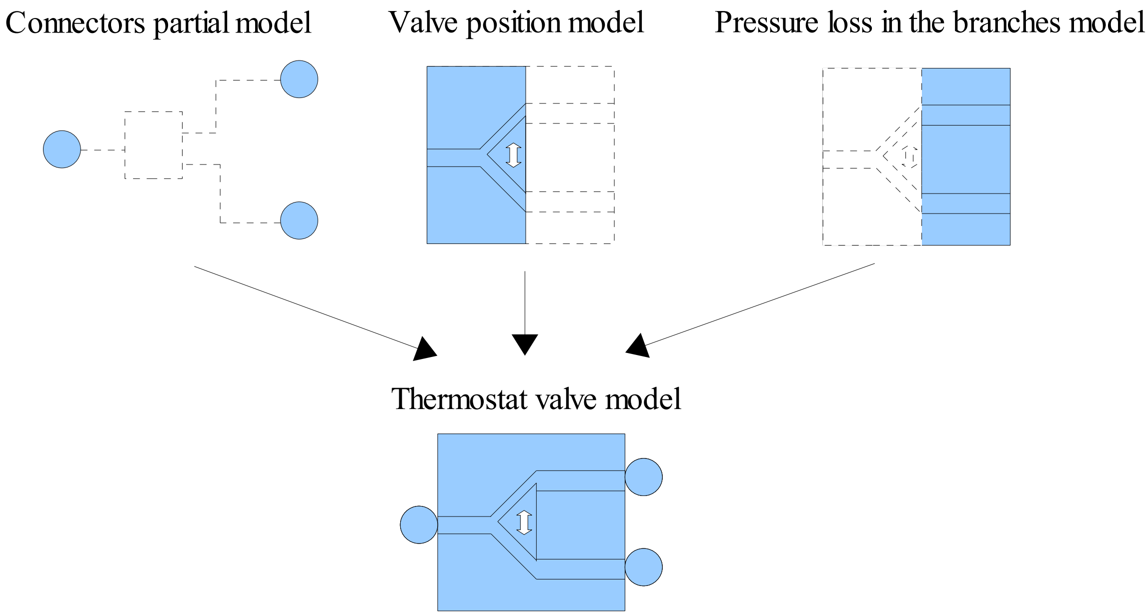

Figure 4.4.8: Themrostat valve composition

4.5 Pressure Loss Components

In the engine cooling circuit there are some components to be modelled that have a pressure loss function. This components are different but since they concern a pressure loss, they can be handled in the same way. Data is provided from experimental essays. This data is the pressure loss in the component at different volumetric flows.

At laminar flow the pressure loss has a linear relation with the flow, but at a turbulent flow the relation is quadratic. These models are in between the two zones, for this reason they can be approximated by a function like:

$$\Delta p = R \cdot q^n$$

where the *n* can be a number between 1 and 2 depending on the fluid turbulences into the component.

In *Modelica.Media.Incompressible.TableBased.Polynomials_Temp* there is a tool that finds the best coefficients to approximate a group of points to a polynomial. This uses the least square method.

A transformation can be done to this formula to have a polynomial structure. If we apply logarithms to both sides of the equation, the result is:

$$\log(\Delta p) = n \cdot \log(q) + \log(R)$$

This is a polynomial of grade 1:

$$p' = n \cdot q' + R'$$

with:

$$p' = \log(\Delta p)$$

$$q' = \log(q)$$

$$R' = \log(R)$$

Then the tool that gives the minimum coefficients can be used, and only a logarithmic transformation in the coefficients and in the data has to be done.

The data is stored in records, and a function does the logarithmic transformations, and the back exponential transformations also, and returns the coefficients for each data we give as an input.

The 0 point of the data has to be removed because we have to apply logarithm to the points. The data points have to be appropriately chosen, so we have a good fitting. We could find some problems in the fitting due to the logarithmic transformation. The data has a logarithmic distribution, and we have to be careful with points that are far from the other ones. For example the logarithmic transformation gives too much weight to points close to 0.

We are going to give priority to the points that have a high pressure loss, because this is where the component is going to work and we want more precision at this points. Because it's difficult to obtain precise data from the low flow points, then the high points may have less error.

The component is going to be a unique model that has a parameter where the user can decide which pressure loss he wants to use. This is done in this way because the components are very similar, and it will be better for the user to not have a lot of models.

The data of the real component is stored in different records. These records are replaced in the model by the user when the pressure loss component is chosen.

Pressure loss components:

- Oil Heat Exchanger

  The high relative movements in the engine, and the high velocities generates big amounts of heat due to the friction. Excessive temperatures reduce the oil's viscosity and shorten the engine's life.

  The coolant goes through a heat exchanger that cools down the engine oil. When the coolant goes through the heat exchanger it has a pressure loss that is between 0 and 1 bar for flow values that go from 0 to 1 l/s (Figure 4.5.1).

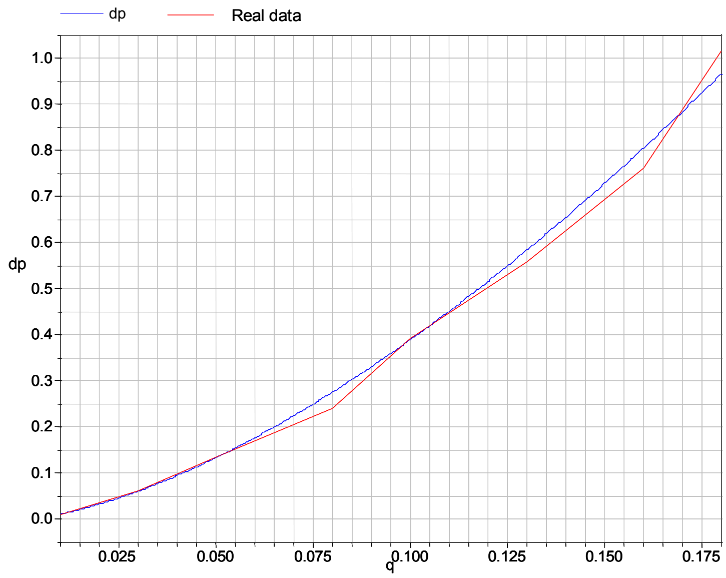

Figure 4.5.1: Coolant pressure loss for the oil heat exchanger. Continuous line for the fitting curve and points with connection lines for the real data.

- Crank case

  The crank rotation at high r.p.m. produces an increase of the temperature of the crank case. It's good to keep it in a controlled range of temperature to avoid dilatation problems along the life span of the materials.

  For this reasons the crank case has also cooling due to the friction production in this zone. The coolant has to go through some conducts inside the crank and has a pressure loss. This pressure loss goes from 0 to 0.2 bar for flow that is from 0 to 2.7 l/s (Figure 4.5.2).

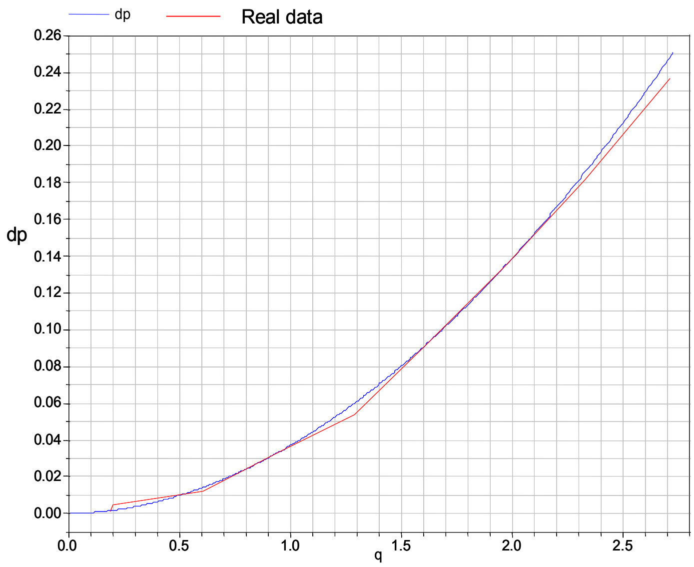

Figure 4.5.2: Coolant pressure loss for the oil crank case. Continuous line for the fitting curve and points with connection lines for the real data.

- Cylinder bank

  The cylinder bank is a high friction zone. The temperature of the cylinder bank has to be at good levels to not burn the oil that avoids higher friction between the piston and the cylinder bank. It's also good to keep its temperature steady because too high temperatures could accelerate the combustion inside the cylinder, or burn at too high temperature fuel and increase the NO_x production.

  The cylinder bank has some ducts where the coolant flows to cool it down. The pressure loss produced here could go from 0 to 1 bar, for flows from 0 to 2.5 l/s (Figure 4.5.3).

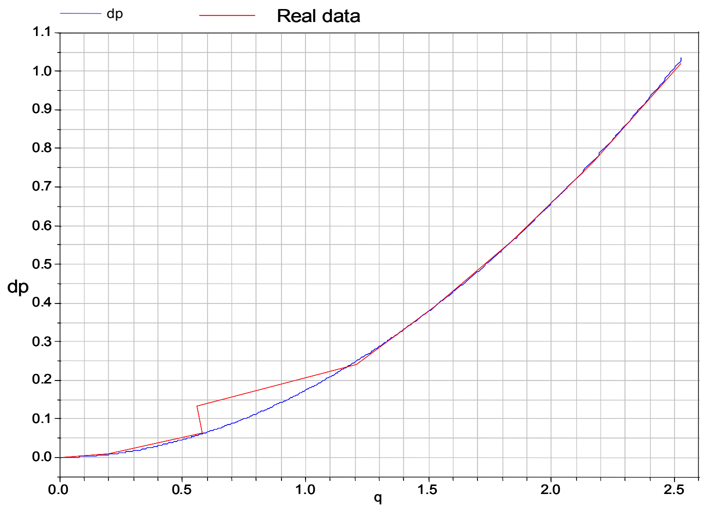

Figure 4.5.3: Coolant pressure loss for the cylinder bank. Continuous line for the fitting curve and points with connection lines for the real data.

- Cylinder Head Outlet Heating

  The cylinder head has some ducts where the coolant goes through to cool or heat it when the engine is turned on.

  The pressure loss produced by this ducts can go from 0 to almost 0.3 bar for flow smaller than 0.6 l/s (Figure 4.5.4).

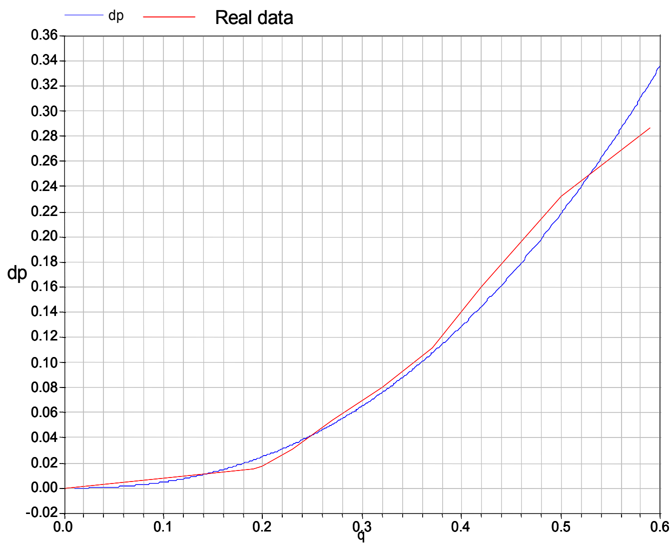

Figure 4.5.4: Coolant pressure loss for the Cylinder Head Outlet Heating . Continuous line for the fitting curve and points with connection lines for the real data.

– Cylinder Head Outlet Thermostat

This is the component located just before the thermostat valve.

The main importance of these components for our circuit is the pressure loss they produce to the coolant, and these positions.

The pressure loss for flow between 0 and 0.2 are from 0 to 0.5 bar (Figure 4.5.5).

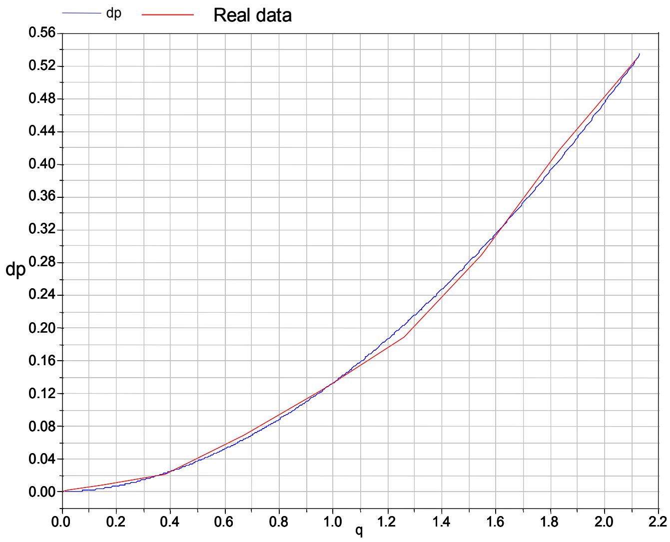

Figure 4.5.5: Coolant pressure loss for the Cylinder Head Outlet Thermostat . Continuous line for the fitting curve and points with connection lines for the real data.

– Cabin Heat Exchanger

When hot air is demanded in the cabin, a heat exchanger heats the air into the cabin. The heat of this air comes from the engine. The coolant gets heated in the engine and before being cooled again by the radiator, some of this coolant is conduced to the heat exchanger of the cabin air to increase this air temperature.

The pressure loss in this component can go from 0 to 0.8 bar in flow from 0 to 1 l/s (Figure 4.5.6).

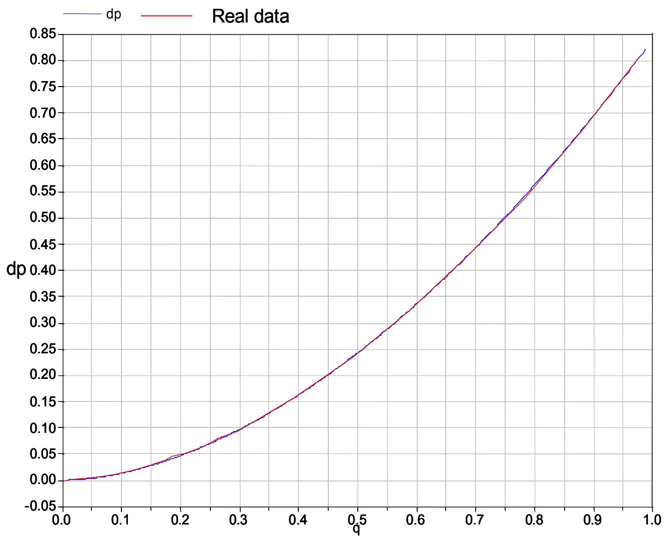

Figure 4.5.6: Coolant pressure loss for the Cabin Heat Exchanger . Continuous line for the fitting curve and points with connection lines for the real data.

- Washing Water Heating

  The water that is pumped to clean the windscreen is also heated. To heat this water the engine temperature is used through the coolant circuit of the engine. There is not a lot of coolant needed to heat the washing water, hence the test of this component uses flow between 0 and 0.1 l/s, and produces pressure loss between 0 and 0.6 bar (Figure 4.5.7).

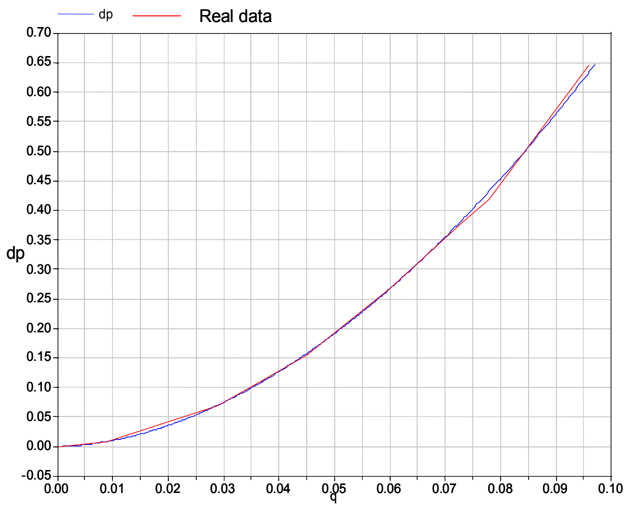

Figure 4.5.7: Coolant pressure loss for the Washing Water Heating . Continuous line for the fitting curve and points with connection lines for the real data.

- Expansion Tank

  The fluid in the circuit has an increase or a decrease of the volume due to the changes of temperature. Since we have a closed circuit, an Expansion Volume where fluid can be stored is needed. In this tank fluid is stored when the volume of the fluid increases. This expansion volume will be modelled, but its pressure loss has to be modelled too, and it is modelled separately. There is a pressure loss component that will be connected to an Expansion Volume model.

  The pressure loss in this component test was from 0 to below 0.7 for flow from 0 to 0.1 bar (Figure 4.5.8).

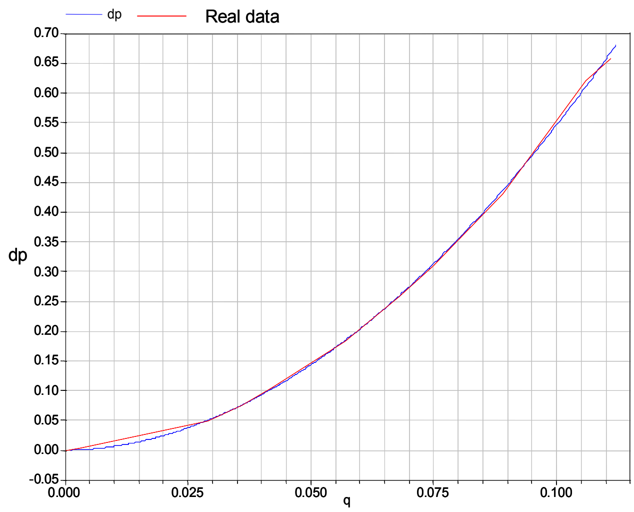

Figure 4.5.8: Coolant pressure loss for the Expansion Tank . Continuous line for the fitting curve and points with connection lines for the real data.

- Radiator

  In the radiator the coolant travels a long way through small pipes to have the maximum heat exchange surface. This leads to pressure losses that have to be counted. Data from a test with flow from 0 to almost 3 l/s was provided. The resultant pressure losses go from 0 to near 1 bar (Figure 4.5.9).

  The radiator heat exchange will be modelled separately. The pressure losses are in a separate component that can be connected to the heat exchange model.

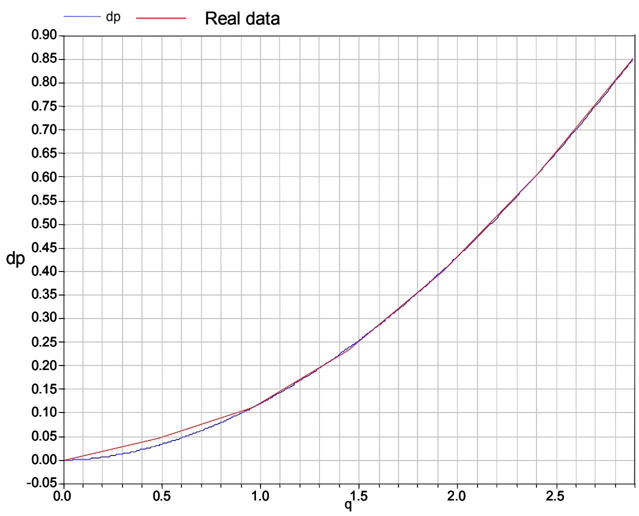

Figure 4.5.9: Coolant pressure loss for the Radiator . Continuous line for the fitting curve and points with connection lines for the real data.

4.6 Expansion Volume

When coolant is heated, it expands. If the cooling system was completely closed, the forces generated by expansion of the coolant could damage the coolant piping. The expansion tank provides space for coolant expansion and added coolant capacity to make up for small losses. It also keeps the pressure of the system constant for small pressure variations. This is a way to keep the stability of the system.

This device is generally installed some distance above the engine to provide a static head to prevent coolant flashing to steam which might result in cavitation in coolant circulating pumps. It is usually made of glass fiber-reinforced polyamide.

The structure of the expansion volume (Figure 4.6.1) consists in a coolant storage volume, air chamber (or another gas), and a membrane that separates this two volumes. The membrane flexibility lets the two volumes have almost the same pressure. There is a thermal conduction through the membrane that tends to equilibrate the temperatures between the two chambers.

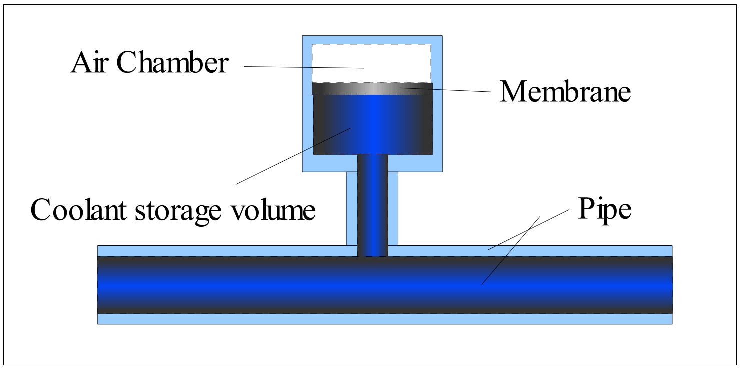

Figure 4.6.1: Expansion volume

To model the expansion tank, some simplifications are assumed. This makes the model simpler with small differences in the numerical results.

Assumptions:

- The pressure in the air chamber is the same as the pressure in the coolant volume. The coolant can freely flow into the volume, then the pressure of the air chamber is the same as the medium pressure.

$$P_{liquid} = P_{gas}$$

- The add of the coolant volume and the air chamber volume is a constant parameter. This is the total volume V.

$$V_{exp.vol.} = V_{gas} + V_{liquid}$$

- The air is an ideal gas. The ideal gas equation is:

$$p \cdot V = n \cdot R_u \cdot T$$

  Where the R_u is the universal constant of gas, and n is the number of moles of gas.

  This equation can be transformed in the following way:

$$m_{gas} = n \cdot M$$

Where m_{gas} is the mass of the gas, n is the number of mole in the volume and M is the molar mass.

$$R_u = R \cdot M$$

Where R is the air constant and R_u is the universal gas constant.

The final equation for ideal gas is:

$$p \cdot V = m_{gas} \cdot R \cdot T$$

- The temperature conduction between the coolant volume and the air chamber has a delay that depends on a constant heat transfer coefficient.

  The air is closed in a chamber that only has heat exchange with the liquid through the membrane. The increase or decrease of the energy in the gas can be written in the following way:

$$\Delta U = cp_{gas} \cdot m_{gas} \cdot \delta T_{gas}$$

  We know that the only incoming energy that this gas has is the contact with the liquid. Then we can write:

$$\Delta U = -W + Q$$

  Where the work is:

$$W = p_{fluid} \cdot \delta V_{gas}$$

  And the heat flow is:

$$Q = 1/k_{ht} \cdot (T_{fluid} - T_{gas})$$

  Where k_{ht} represents the heat transfer coefficient. This is the equation for the heat transfer through a wall. It is written in this way because we want to have a k_{ht} coefficient that is proportional to the delay of the heat transfer. This parameter will decide the time response of the component.

$$Cp_{gas} \cdot m_{gas} \cdot \delta T_{gas} = -p_{liquid} \cdot \delta V_{gas} + \frac{1}{k_{ht}} \cdot (T_{liquid} - T_{gas})$$

Balances

This model only has one connector. We can not use the two pin connections used before. This model stores mass and energy because the coolant chamber changes its volume.

The energy and mass balances change because of the mass storage variation.

- Mass balance:

  The mass (m_{liquid}) into the volume depends on the density of the liquid (d_{liquid}) and the volume of the liquid (V_{liquid}).

$$m_{liquid} = V_{liquid} \cdot d_{liquid}$$

  The mass flow (m_{flow}) of the port is the variation of the liquid mass in the component.

$$\delta m_{liquid} = port.m_{flow}$$

- Energy balance:

The energy balance is also different from the components that does not store mass. The energy not only depends on the input and the output of energy (enthalpy flow rate), it also depends on the variation of energy inside the volume.

The definition for the total Enthalpy (H) is:

$$H = U + P \cdot V$$

This is parallel to the first law of the thermodynamics since in this case Q = ΔH:

$$\Delta H = \Delta U + P \cdot \Delta V$$

Then this is the balance equation where the exchange of energy through the connector and the variation of the stored energy (P.ΔV) produces the internal energy variation.

The remaining variables are computed as follows:

The fluid property submodel calculates the specific internal energy u_{liquid}. This is the internal energy per unit of mass, so it has to be multiplied by the mass inside the volume (m_{liquid}) to have its internal energy (U_{liquid}).

$$U_{liquid} = m_{liquid} \cdot u_{liquid}$$

For the ΔH the port has the enthalpy flow rate that depends on the mass flow and the enthalpy of the fluid:

$$\Delta H = \dot{H} = \dot{m} \cdot h$$

Where $\dot{H}$ is the enthalpy flow rate, $\dot{m}$ is the mass flow and h is the liquid enthalpy.

The enthalpy flow rate is given by the port.

The pressure of the volume is given by the fluid property submodel.

The volume variation can be given in two ways, but with a difference in the signs:

$$\dot{V}_{liquid} = -\dot{V}_{gas}$$

Where V_{liquid} is the volume of the liquid and the V_{gas} is the volume on the gas side.

The balance equation of the model will be:

$$\delta U_{liquid} = \dot{H} + p_{liquid} \cdot \delta V_{gas}$$

4.7 Pump

The pump is an important component of the engine cooling system. Its function is to provide energy to the fluid so it can flow through the circuit. There are different pump configurations, but if look to the pump is seen as a black box they provide the same function. Every pump has a characteristic set of curves that describes its performance. This is data usually provided by the pump manufacturer.

The curves usual given are the pump head and the power consumption, as a function of the coolant flow.

The efficiency curve is usually also given, but it is not strictly necessary for the calculation. It is used to decide which pump gives the best efficiency in a circuit.

The head and the power consumption curves are input parameters for the pump model. The least square fitting is used to approximate a quadratic curve to the points given by the user. Theoretically the head curve of the pump follows a 3rd order curve (figure 4.7.1), but since the pump is only used in the positive region, it can be fitted with a 2nd order curve (figure 4.7.2).

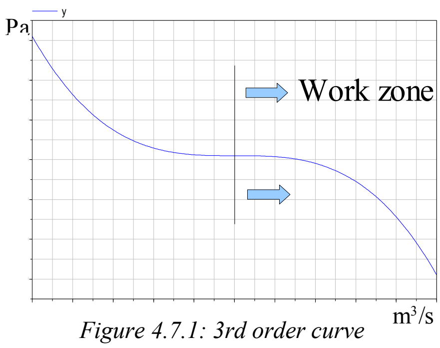

Figure 4.7.1: 3rd order curve

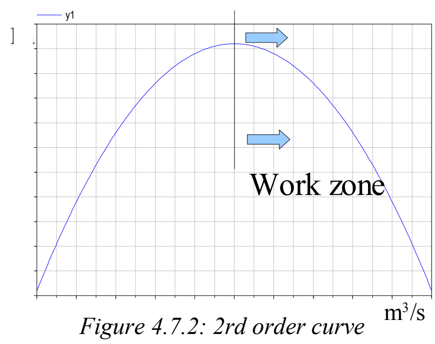

Figure 4.7.2: 2rd order curve

To obtain this 3rd order curve, a special test has to be done. Imagine a long vertical pipe, with a pump in the bottom (figure 4.7.3). The pump is turned on and the head produced by the pump will increase with the time when the fluid goes up into the vertical pipe. There is a point where the head is too big for the pump and the flow goes backwards. If the back flow wants to be rise, fluid has to be poured into the vertical pipe, so the head gets bigger. The more back flow wanted, the more head neede, and this is not a linear relation, this is a 3rd order relation as in the figure 4.7.1.

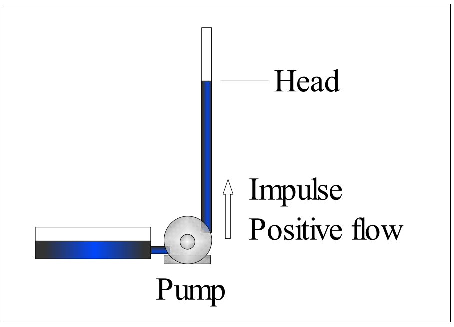

Figure 4.7.3: Pump test

From the points given by the user, the model computes the coefficients of the quadratic curve that fits best with the least square function.

Then, this coefficients are used to compute the head produced by the pump (Tummescheit, H. (1996)):

$$p_B=\rho\cdot(k_1\cdot n^2+2\cdot k_2\cdot n\cdot\dot{V}-k_3\cdot\dot{V}^2)$$

Where p_B is the pressure gain generated by the pump, ρ is the density, k_i are the coefficients of the curve, n the revolutions per second and $\dot{V}$ is the volumetric flow.

But it is wanted in mass flow, and also scaled with the nominal revolutions per second. This means that the pump will give different pressure rise in the system depending on the mass flow and the revolutions per second of the shaft.

$$p_B=\rho\cdot(k_1\cdot(n/n_{nom})^2+2\cdot k_2\cdot(n/n_{nom})\cdot\frac{\dot{m}}{\rho}+k_3\cdot\frac{\dot{m}^2}{\rho^2})$$

Where $\dot{m}$ is the mass flow, n_{nom} is the nominal revolutions per second of the introduced data, and k_i are the coefficients of the curve.

There is a change of the sign because the constants will provide the sign.

We have a similar formula for the power:

$$P=k_{P1}\cdot\frac{\dot{m}}{\rho}+2\cdot k_{P2}\cdot n\cdot\frac{\dot{m}^2}{\rho^2}+k_{P3}\cdot n^2$$

Where P is the power consumption and the k_P are the coefficients of the power curve.

When the pump is coupled to a pipe system, it is an advantage to use a dynamic model for the pump mass flow, based on the steady-state characteristic (Tummescheit, H. (1996)):

$$\frac{\delta\dot{m}}{\delta t}=(p_1-p_2+p_B)\cdot\frac{A}{l}$$

Where p_1 is the pressure in the pump inlet, p_2 is the pressure in the pump outlet, A is the area of the pipe and l is the length of the pipe.

To use this formula it has to be known the area of the pipe and the length, but there is different pipe area, and different branches. It is not a unique pipe. For this reason the equation is transformed.

This relation $\frac{A}{l}$ is take away (this is the inertia of the liquid in the pipes), and now $(p_1-p_1+p_B)$ is scaled with $k_{inertia}$ and divided by a time constant $\frac{1}{T_{inertia}}$.

$$k_{inertia}=\frac{\dot{m}_{nom}}{he_{nom}}$$

$\dot{m}_{nom}$ is the nominal mass flow and he_{nom} is the nominal head.

$T_{inertia}$ is the equivalent hydraulic time constant. This constant is the estimation of the time it takes for the pump to begin to move the coolant in the circuit.

Then the final equation is:

$$\frac{\delta \dot{m}}{\delta t} = (p_1 - p_2 + p_B) \cdot \frac{k_{inertia}}{T_{inertia}}$$

The pump model also needs the balance equations. The volume inside the pump does not change, so the mass flow is the same in the inlet and in the outlet.

$$\dot{m}_{inlet} + \dot{m}_{outlet} = 0$$

Since there is an input of energy, the power balance will be different than in the other models:

$$\rho \cdot V \cdot \delta h = \dot{H}_{inlet} + \dot{H}_{outlet} + P_{hyd}$$

Where:

ρ fluid density

V pump internal volume

δh derivative of the fluid specific enthalpy

P_{hyd} hydraulic power

$P_{hyd} = P \cdot \eta_{mech}$ where P is the power calculated before with the curve coefficients, and η_{mech} is the mechanical efficiency

$\dot{H} = \dot{m} \cdot h$ enthalpy flow

As in other models, the *semilinear* function is used to calculate the enthalpy flow. In the model the enthalpy (*h)* will be the enthalpy of the fluid in one side or the other side of the pump, depending on the direction of the mass flow. This is the way to handle reverse flow situations, although in the engine cooling is not necessary. For this reason we have a flag in the parameters that allows or disallows reversing flow. This flag is called *checkValve* because it is as if there were a valve that does not allow reverse flow.

It is interesting to know the global efficiency of the pump. This is the relation of the power generated in the fluid and the power given by the pump:

$$\eta = \frac{(p_{outlet} - p_{inlet}) \frac{\dot{m}}{\rho}}{P + P_{eps}}$$

Where:

η global efficiency

p_{outlet} and p_{inlet} are the pressures in both sides of the pump

P pump power consumption

$P_{eps} = 1e^{-8}$ small coefficient to avoid numerical singularities when the power is zero

The model also has initial equations. The initialization could give numerical problems that can be solved with proper initial values.

The model has two types of initialization *steadyState = true* and s*teadyState = false*. In the *steadyState = true* the energy balances are initialized simulating a steady-state situation, where the derivatives of all states are zero.

$$\delta h = 0$$
$$\delta \dot{m} = 0$$

In the *steadyState = false* option we just set this variables with start values.

$$h = h_{start}$$
$$\dot{m} = m_{flow\,nominal}$$

Where h_{start} and $m_{flow\ nominal}$ are parameters that can be changed by the user.

The model uses the medium package and the standard connectors of the library as well.

This model is a modification of the original pump model from the CombiPlant library.

4.8 Engine

The combustion and friction in the engine produce heat that has to be removed to avoid an excessive increase of the temperature.

The engine is cooled in three ways:

- Coolant: The cylinders are surrounded by a water-filled cavity, the water jacket or cooling jacket. An important engineering dimension is the water jacket depth, defined as the distance from the top plate plane to the lowest point in the water jacket. In earlier grey cast engine blocks, this dimension was as much as 95% of the length of the cylinder running surface. In modern cast iron engine block designs, the water jacket ends in the area swept by the lower piston ring. The water jacket is even shorter in modern aluminium engine blocks. The water jacket depth corresponds to about one third of the length of the cylinder running surface. This is made possible by the greater thermal conductivity of aluminium alloys in comparison with cast iron materials and by pistons with even shorter compression heights. A short water jacket reduces the coolant volume in the engine and, thus, the engine weight. The smaller coolant volume and thermal capacitance shortens the engine's warm-up phase, with positive effects in terms of unburned hydrocarbon emissions and the response time for the catalytic converter.
- Air: The air going through the radiator and the bypass of the radiator cools down the engine also. There are also some engines cooled only by air instead of the coolant. Only a very few manufacturers still use air-cooled cylinders in automotive engines today. Heat dissipation in air-cooled cylinders is dependent upon the thermal conductivity of the cylinder fins and on the cylinder materials, shape of the cooling fins and the way in which cooling air passes across the fins.
- Oil: The oil that lubricates the engine also cools down the engine. In some cases the oil is refrigerated by the engine cooling system to improve the efficiency of the engine, or to maintain the oil bellow the maximum permissible temperature. To cool down this oil we can have an engine oil cooler that uses coolant or directly the air in the cooling module. The ones that uses coolant are preferably located near the engine and can have a round disc shape, disc stack, or flat tube design. They are usually made of aluminium. Direct cooling with oil and air coolers is also common, with very high-pressure soldered flat tube designs in aluminium being arranged in the cooling module. In commercial vehicles, cooling is always performed with coolant, while the coolers are normally installed in an opening in the crankcase where they are exposed to the main flow of the coolant. The most widespread design is with plate-type coolers of stainless steel with turbulence inserts on the inside through which the oil flows. More recently, the use of aluminium coolers with higher capacities and comparable strengths but roughly half the weight has become possible. The oil used in transmission and the hydraulic oil used in power steering or other servo systems also has to be cooled in some cases.

The engine could be represented as a big thermal capacitance where there is an incoming of energy from the combustion and the friction. The coolant is in contact with the mass through a wall and there is convective heat transfer between wall and coolant.

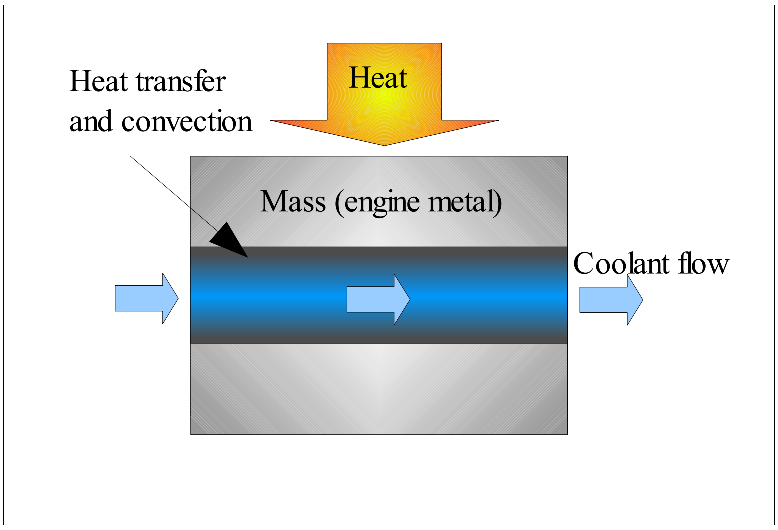

Figure 4.8.1: Engine simplification

We also discretizate the engine in to volumes and pressure drops. Every segment is equivalent to a small volume and the corresponding pressure drop.

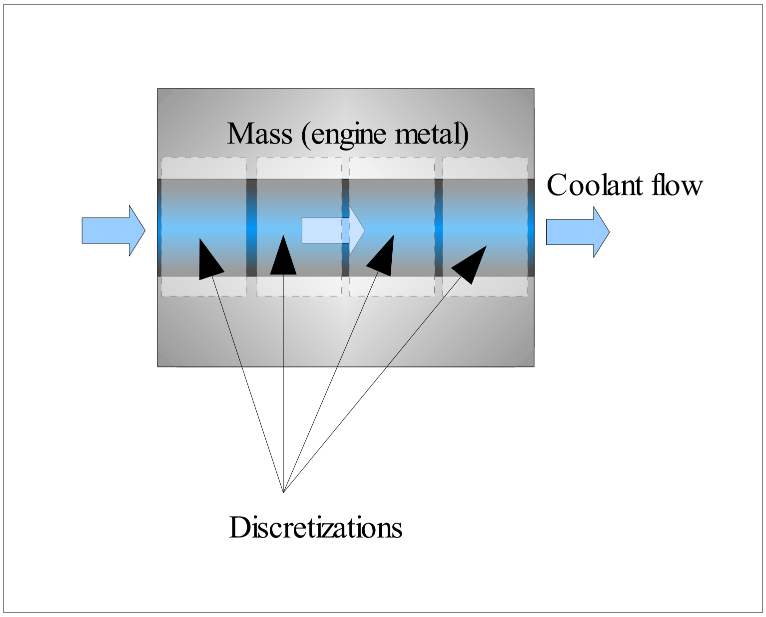

Figure 4.8.2: Engine discretization

Every volume has the wall with the convection and the corresponding piece of metal mass. The number of discretizations is a parameter that can be changed by the user. The model looks like in figure 4.8.3.

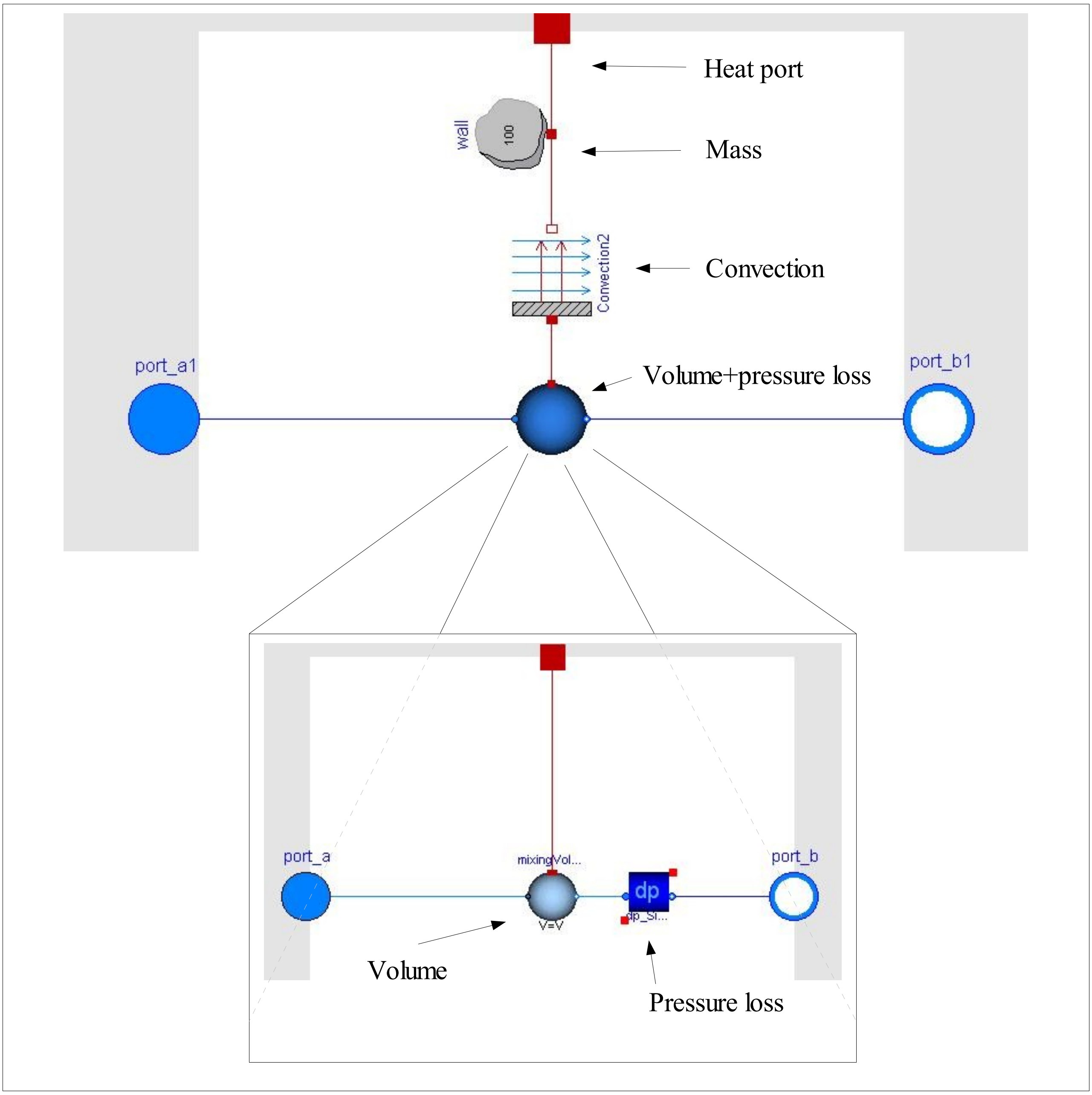

Figure 4.8.3: Dymola model schematic of engine model

The pressure loss is a very simple component that depends on only one coefficient. Is a simple quadratic pressure loss.

$$dp = dp_{coef} \cdot \dot{m}^2$$

Where dp is the pressure drop in Pascal, dp_{coef} is the coefficient of the pressure drop and the $\dot{m}$ is the mass flow through the component.

The mass is a component from the Modelica Standard Library. It is called HeatCapacitor. This represents a generic model for the heat capacity of a material. This model assumes that there is constant temperature in the entire volume, no shape assigned and a constant, heat capacity independent of the temperature.

The convection element is a simple modification of the element called Convection, from the Modelica Standard Library. This component is a model of linear heat convection, e.g. the heat transfer between a plate and the surrounding air. We can use it for a complicated solid geometries and fluid flow over the solid by determining the convective thermal conductance Gc by measurements. The modification makes the thermal conductance Gc constant, calculated from the convection area and the head transfer coefficient.

4.9 Radiator

The radiator is the component where heat is removed from the coolant through heat exchange with the air that is passing through it.

The traditional vehicle radiator was made by non-ferrous metal materials, with copper fins and brass pipes. In Europe, they have been suppressed in cars since 1975 and in commercial vehicles since 1988 by further-developed Al alloys offering a weight advantage of up to 30% with high-pressure resistance thanks to the brazing and a higher corrosion resistance (Richard van Basshuysen, et al. (2002)).

The radiator is part of a structure formed by different layers. On the front layer we have the air conditioning heat exchanger. The second layer is the engine cooling heat exchanger. The structure of this two heat exchangers is different. The air conditioning has two phase refrigerant, due to this its structure is a group of folded pipes that decrease its number in each fold due to the decrease of volume of the refrigerant. The engine cooling radiator is a simpler structure formed by two vertical main leaders that are connected through a group of smaller horizontal pipes (see Figure 4.9.1). The horizontal pipes can be round or oval.

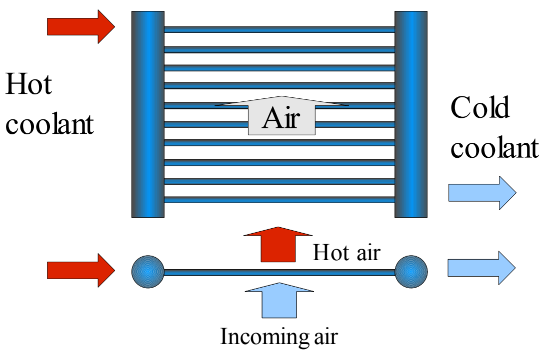

Figure 4.9.1: Radiator pipes structure

In between the horizontal pipes are fin structures to increase the heat exchanging surface.

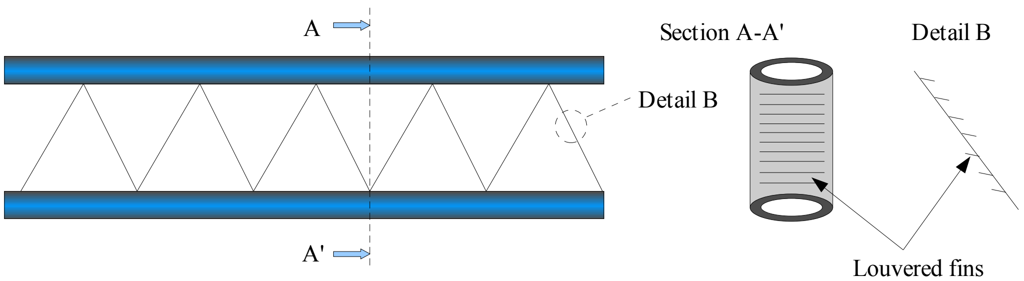

Figure 4.9.2: Fin structure

This fin structures can have different shapes. The fins have stamped smaller fins called louvred fins (Figure 4.9.2).

Some times the engine has a heat exchanger for the oil cooling. The engine cooling heat exchanger doesn't take all the radiator area. This area is used for the heat exchanger of the oil cooling. In some cases this heat exchanger is in the air conditioning layer.

Pipes and fins form the “radiator matrix”. A distinction is made between two kinds of radiator matrices:

- *Mechanically assembled rib and pipe systems* of round or oval pipes and slot-fitted punched ribs linked to one another by expanding the pipes. These systems typically cover the lower power segment, but thanks to improvement expansion techniques and ever narrowed oval pipes also achieve the power spectrum of soldered systems.

-*Soldered systems of flat pipes and rolled corrugated ribs*. Today, these are generally manufactured with only one tube in the system depth. They are some times ribbed to increase the strength.

The radiator, from the smaller vehicles to the largest commercial ones, has a depth between 14 to 55 mm, and it can be more than 80 mm for non-ferrous metal radiators. The cooling air surfaces range from 15 to 55 dm^2.

There are some differences between the vehicles used in Europe and the vehicles used in the United States and Japan. For the European cars the non-ferrous metal radiators are completely substituted by the Al alloy radiators, while in Japan and in the United States the non-ferrous metal radiators are still widespread.

There is another big difference in the pipes position of the radiator. The radiators for the European cars are mainly constructed in cross-flow design with the pipes running horizontally, whereas in the United States and Japan they are frequently also built in down-draft design.

Another important part of the radiator is the radiator fan. Most often it is a suction fan, and can be single fan or double (twin) fans. The maximum vane diameter is approximately 500 mm. The fan is driven by an electric motors or, most often, a belt driven connected to the crankshaft. For the upper power segment of cars and the full range of commercial vehicles, viscous couplings are used. The drive speed is dictated by the crankshaft or an engine-side gearing. This shaft velocity is coupled by oil friction to the fan. A variable oil filling of the coupling allows the fan speed to be varied from an idle speed to just bellow the drive speed.

The electrical motor have consumptions up to 600W. The power of viscous coupled fans can go up to 30kW, with maximum fan diameter used in commercial vehicles around 750 mm. (Richard van Basshuysen, et al. (2002))

The model built for this component is similar to the engine model. This is a simple component that has a coolant flow side and an air flow side, and there is heat exchange between both fluids. Normally the flow is in cross-flow arrangement. There is a heat capacitor component that represents the radiator mass (pipes and fins) between two fluids. This is the same component that was used in the engine component to represent the engine mass. Then there are two convective heat transfer components on both sides of the heat capacitor that represent the heat transfer through convection to the fluids.

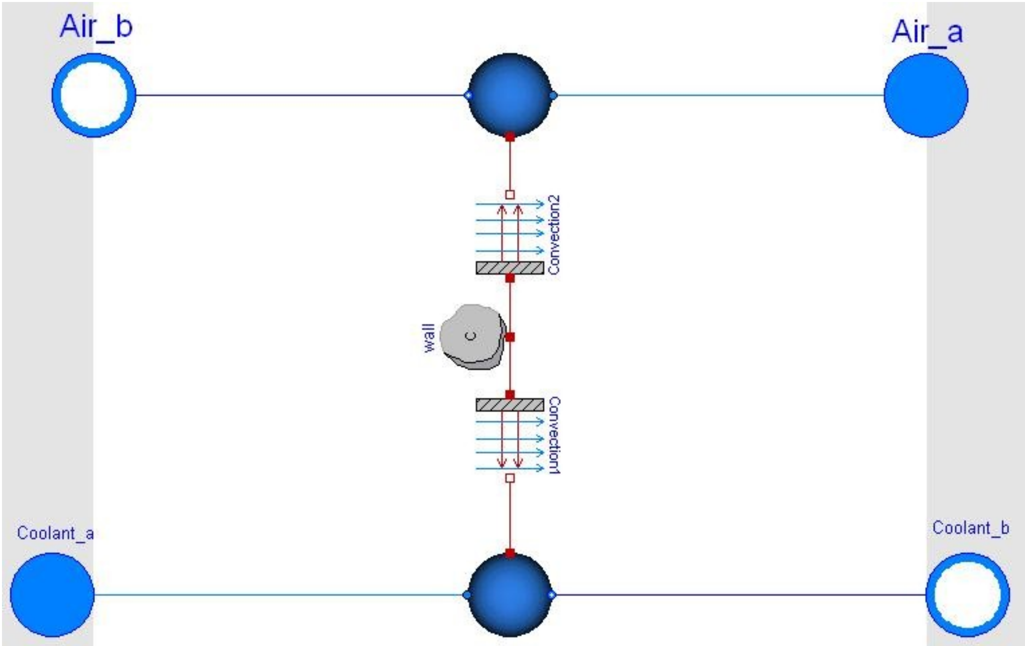

Figure 4.9.3: Radiator model

The structure of this model is similar to the engine model, but with two fluid sides. We have a discretization of the both sides with volumes and pressure drops. The user sets the number of discretizations in a parameter. Each volume in the coolant side is connected to a volume in the air side through a thermal port and some components in between. These components are two convective heat transfer components and a heat capacitor in between. The convective heat transfer components simulate the heat transfer between the fluid and the wall, and the heat capacitor simulates the heat capacitance of the walls and fins of the radiator. Figure 4.9.3 shows the Dymola model schematic, but this is for one discretization. The model multiplies this scheme for the number of discretizations.

4.10 Dynamic Pressure Loss

The engine cooling model uses a fluid that can be assumed to be incompressible. This approximation simplifies the calculations for the computer simulation and gives a very low error. This error is small because the engine coolant has a very low compressibility.

In a system where there is a compressible fluid two states are needed to model the volume components. This states can be, for example the pressure and the temperature. Since in the engine cooling system the fluid is incompressible there is only one state in these components, usually the temperature. The pressure is an algebraic variable that depends on the mass flow in the flow components. The mass flow in the flow components depends on a non linear equation. Since there is no pressure as a state between the flow components, there is a non linear system of equations that grows with every flow component that is connected in the system.

The engine cooling system is a loop and a component that has either a pressure or a mass flow as a state is needed to decrease the size of non-linear system of equations. The pump component has the mass flow as a state, but the basic engine cooling system has two loops, one for each branch, so two components that brake the loop are needed. For this reason the dynamic pressure loss component is a component that acts as a normal pressure loss component but has the mass flow as a state.

The main equation comes from the equation of the simple quadratic pressure loss, where:

$$\Delta p = Coef_{\Delta p} \cdot \dot{m}^2$$

Where Δp is the pressure drop between the ports of the component, $\dot{m}$ is the mass flow, and $Coef_{\Delta p}$ is the coefficient that determines the size of the pressure loss.

This formula is changed to have the mass flow as a state. One way to do this is to transform the formula of the simple quadratic pressure loss to have the mass flow in one side:

$$\dot{m} = \sqrt{\frac{\Delta p}{Coef_{\Delta p}}}$$

This is the mass flow depending on the pressure drop quadratic curve. Is the theoretically mass flow if the component follows the quadratic curve. The difference between the theoretical mass flow and the real mass flow in the component gives us the variation of the mass flow, that is the derivative of the mass flow:

$$\frac{\partial \dot{m}}{\partial t} = (\dot{m}_{quadratic\,curve} - \dot{m}_{component}) \cdot \frac{1}{\tau}$$

Where $\frac{\partial \dot{m}}{\partial t}$ is the derivative of the mass flow, the $\dot{m}_{quadratic\,curve}$ is the mass flow that depends on the pressure loss coefficient that we had in the simple pressure loss, and $\dot{m}_{component}$ is the mass flow in the component that will be modified by its derivative. τ is a time constant that will disappear because it is 1.

The formula of the new dynamic component is:

$$\frac{\partial \dot{m}}{\partial t} = \sqrt{\frac{\Delta p}{Coef_{\Delta p}}} - \dot{m}$$

The square root can cause numerical problems with small values of Δp. For this reason we use a function in the model that calculates the square root everywhere except near zero, where it is replaced with a cubic polynomial with a finite derivative.

4.11 R.P.M Component

When the whole circuit is built, there is a source that determines the r.p.m of the pump, another one that determines the heat that goes to the engine, and another one that decides the air flow that enters the radiator.

For the simple model that is built it could be good to have a component that has the r.p.m of the crank shaft as an input and allows to connect its outputs to the pump, the radiator mass flow and to the engine heat flow. This component represent all inputs that depend on the r.p.m of the engine. This r.p.m component will be used to control the inputs in an easy way, with only one source that can be a table that will be the r.p.m of the crank shaft.

The r.p.m of the pump is usually coupled to the r.p.m of the crank shaft. For the simple engine cooling that we have we can assume that the r.p.m of the crank shaft is linearly coupled to the r.p.m of the pump.

The nominal point of the pump is at 6672 r.p.m, assuming that the nominal point of the engine, at 3000 r.p.m, will be the same. This is the relation between the r.p.m of the pump and the crank shaft:

$$r.p.m_{pump} = r.p.m_{crank\ shaft} \cdot \frac{r.p.m_{pump\ nominal}}{r.p.m_{crank\ shaft\ nominal}}$$

These nominal r.p.m are parameters in the model and that can be changed in an easy way when it is needed.

The energy that the engine cooling has to take away from the engine has a relation linear to the r.p.m of the crank shaft. This relation is not linear, but it is going to be simplified and it is going to use a relation between the r.p.m of the crank shaft and the heat that goes to the engine and warms it.

In a gasoline combustion engine we can assume that the energy per kilogram of gasoline obtained is 43 MJ/kg (Hamilton, B.). The 25% of the energy produced from the gasoline is useful work, the exhaust gas takes a 33%, the engine cooling dissipates 35% and the surroundings waste another 12%.

Instead of MJ/kg, MJ/l are used to make easier calculations and obtain the heat heating the engine. The energy is 34.2 MJ/l, and the energy that goes to the engine is 11.97 MJ/l. Assume that the consumption of a vehicle can be 8 litres in 100 km circulating in an average speed of 100 km/h. With this data we can make a simple calculation to see which the energy per second going to the engine. This value is 26.6 kJ/s. This values is a rough approximation, but it will be enough to check this simple model.

If we suppose a consumption of 6% we would have a value of 19.19 kJ/s. We can also use this value if we want to see the behaviour of the thermostat valve better. At this value the valve doesn't open that fast and the position of the valve can be plotted in a clearer way.

The equation used in the component is:

$$Q_{engine} = r.p.m_{crank\ shaft} \cdot \frac{Q_{engine\ nominal}}{r.p.m_{crank\ shaft\ nominal}}$$

Where Q is the heat flow in J/s.

The flow of air in the radiator could be approximately between 0.5 and 0.8 kg/s. We are going to simplify the model and assume that the relation between the r.p.m of the radiator and the mass flow is linear. This is not true, but for initial tests this assumption is sufficient. The equation used in the model is:

$$\dot{m}_{radiator} = \frac{r.p.m_{crank\ shaft} \cdot \dot{m}_{radiator\ nominal}}{r.p.m_{crank\ shaft\ nominal}}$$

The model has the structure of a normal block of the Modelica library, with one input where a table is connected with the different r.p.m respect to time, and through the equations gives the flow of air in the radiator, the r.p.m of the pump and the energy that goes into the engine.

With the component of the Modelica library called CombiTimeTable maps can be drown of the r.p.m respect time, where r.p.m variations can be introduced through a table. We can simulate the gear changing, sharp acceleration, smooth accelerations, etc.

5 Engine Cooling System Models

5.1 Introduction

Different models are built to check the components separately before trying to assemble a simple but complete engine cooling system. It is important to decide the structure of these models and check the components well and be sure that all components are correct.

The firsts models presented here are simple models to test the components, but the models progress step by step in complexity to reach the simple engine cooling system.

The simple engine cooling system is closed, the fluid circulates, and there is a split and a junction. These elements have to be tested and build with the components to check them and their interaction in different steps.

The basic engine cooling system that will be built has the structure showed in figure 5.1.1.

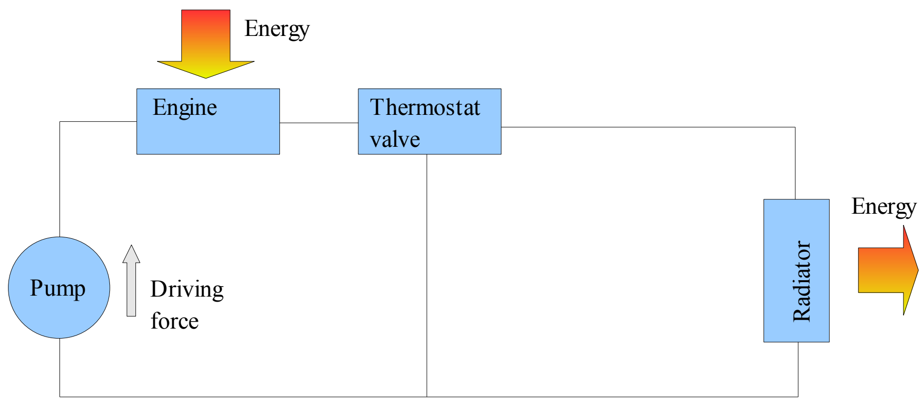

Figure 5.1.1: Simple engine cooling

One of the modelling problems is that we use an incompressible fluid and there is an increase of the temperature and thus of the density of the fluid. Something has to compensate the pressure and the volume expansion of the fluid, otherwise the system will not be physically consistent and a closed circuit will not work. The expansion volume can be placed just after the pump and before the engine. It is a good place to compensate for the increases of pressure due to the variation of the pumps rotational speed.

Some pressure drops will be added to make the circuit reasonable, with reasonable pressures and flow.

The radiator needs an air side that can be controlled (flow and temperature).

We add the R.P.M component to control the pump, radiator and engine. The Dymola scheme is shown in the figure 5.1.2.

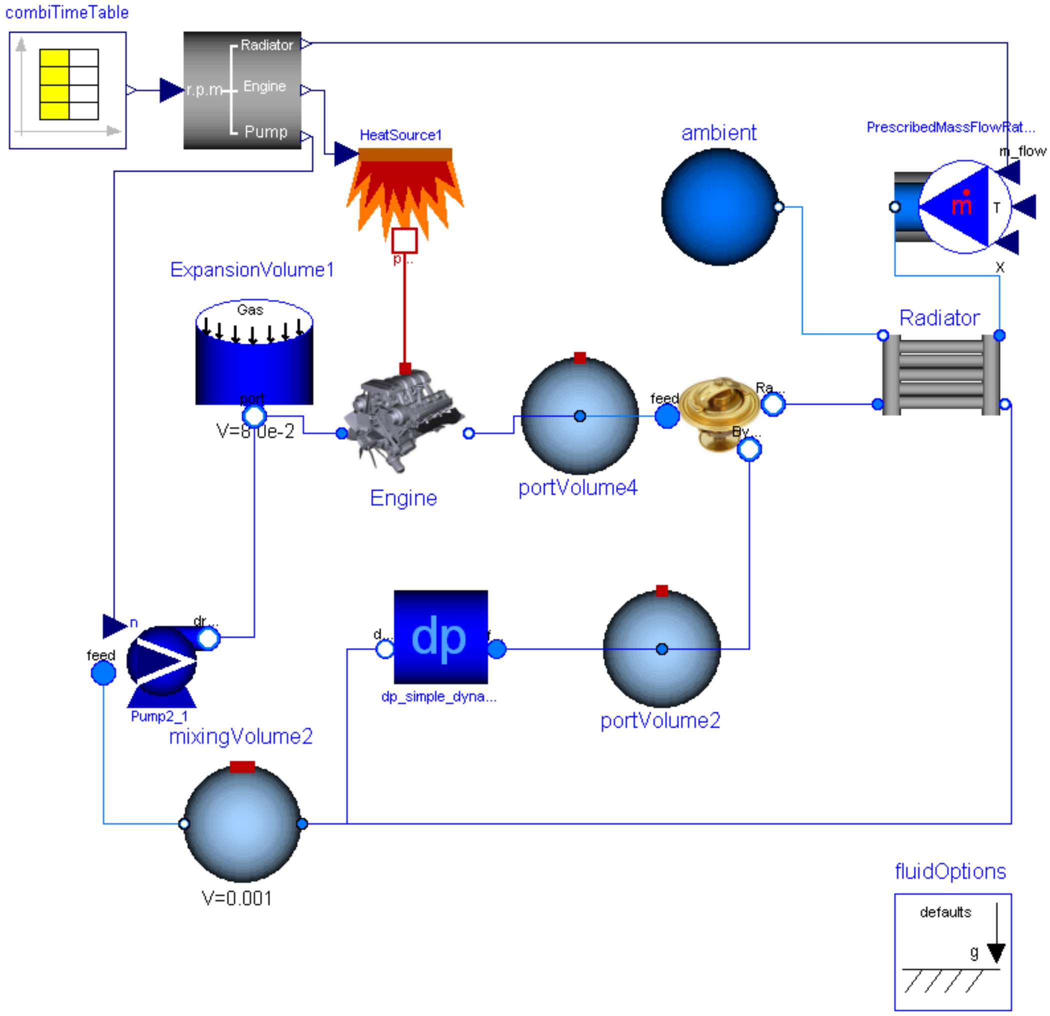

Figure 5.1.2: Dymola model schematic of the engine cooling

The 'combiTimeTable' component lets us introduce points in a table to introduce a r.p.m map. Maps simulating the normal usage of the engine cooling will be used. One example is shown in the map in figure 5.1.3.

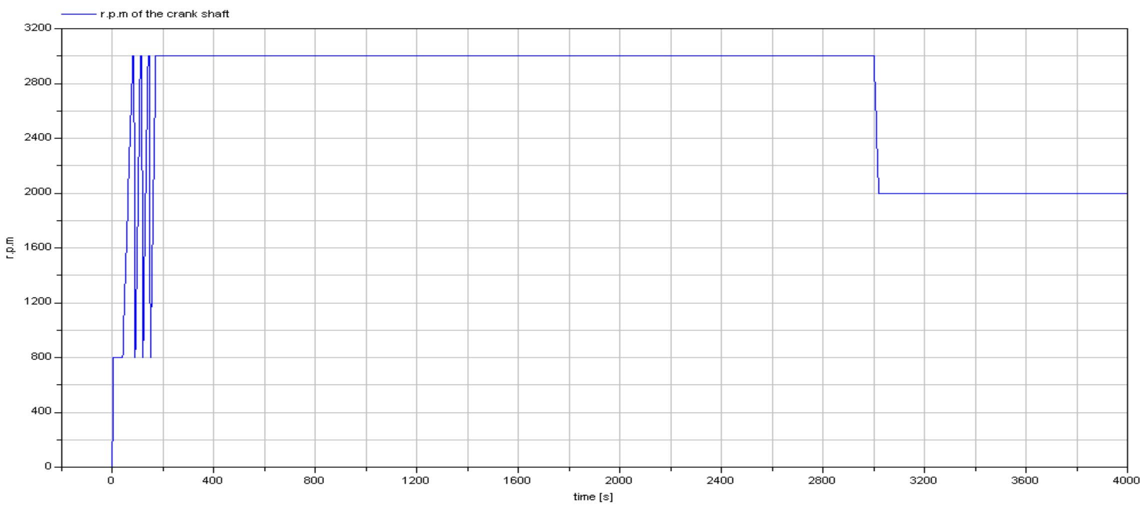

Figure 5.1.3: r.p.m map

This map simulates a possible normal use of the engine. The first part represents when the engine is turned on and the r.p.m of the engine goes from zero to 800 r.p.m. It stays some seconds at 800 r.p.m and then accelerates to 3000 r.p.m in 40 seconds where there is a change of gear. Then the r.p.m go to 800 again and accelerates to 3000 r.p.m in 10 seconds. It represents four gear changes and then it holds 3000 r.p.m until 3000 seconds from the start. This represents driving at constant r.p.m for 50 minutes. Then there is a reduction of r.p.m to 2000 and it holds this revolutions.

A detail of the start to see the gear changes is shown in figure 5.1.4.

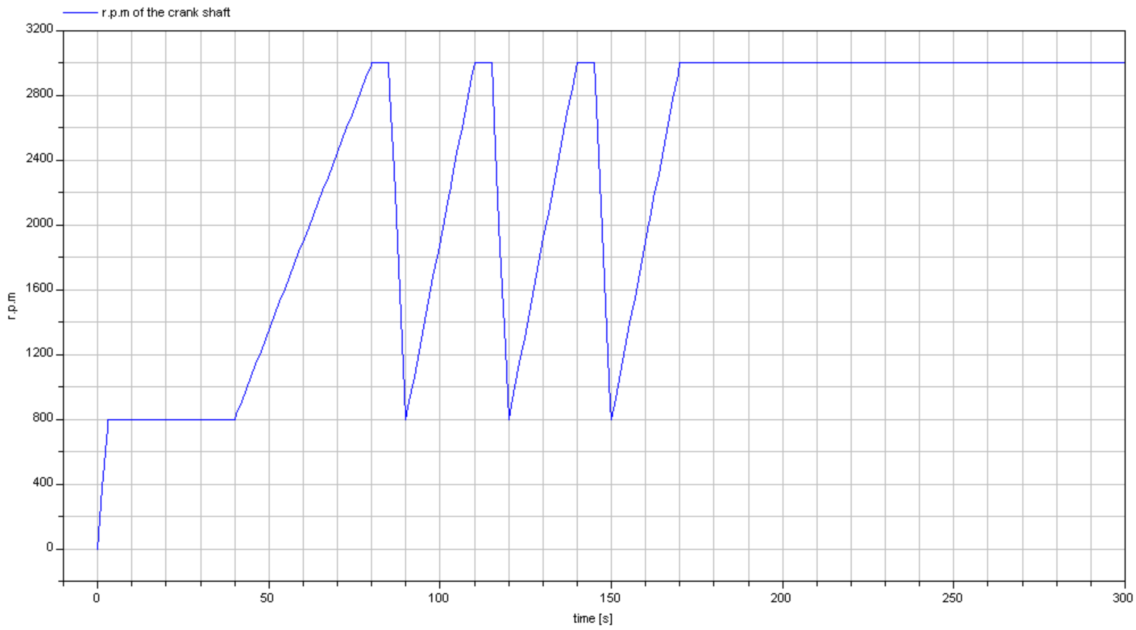

Figure 5.1.4: r.p.m map detail

There is also the component called 'fluidOptions' where to set the initial conditions and definitions of some components can be set.

In chapter 5 the components and its behaviour are analysed. The cycle will be simulated in different conditions to check all the components. When it is necessary some small models will be built to check a specific component.

5.2 Pump Test Model

To test the pump model we use the component called *ambient*. This component acts as an environment where the temperature and the pressure can be set. To test the behaviour of the pump, the pump is connected to two ambient components, one in each side, and the one connected in the drain of the pump has an oscillating pressure. With this set-up, the dynamic pump reaction can be checked. See the Dymola schematic model in the figure 5.2.1.

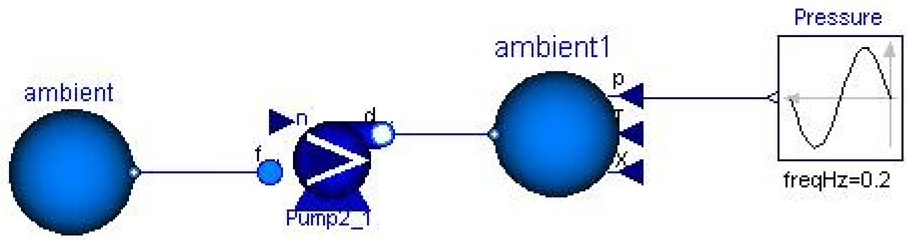

Figure 5.2.1: Dymola model schematic of the pump test

The pressure of the 'ambient1' has to be higher than the pressure of the 'ambient' to simulate the normal pump boundaries conditions. There is 2 bar in the 'ambient' and 2.5 bar in the 'ambient1' and oscillates as a sinusoidal with an amplitude of 0.3 bar, which means that it will go from 2.2 bar to 2.8 bar.

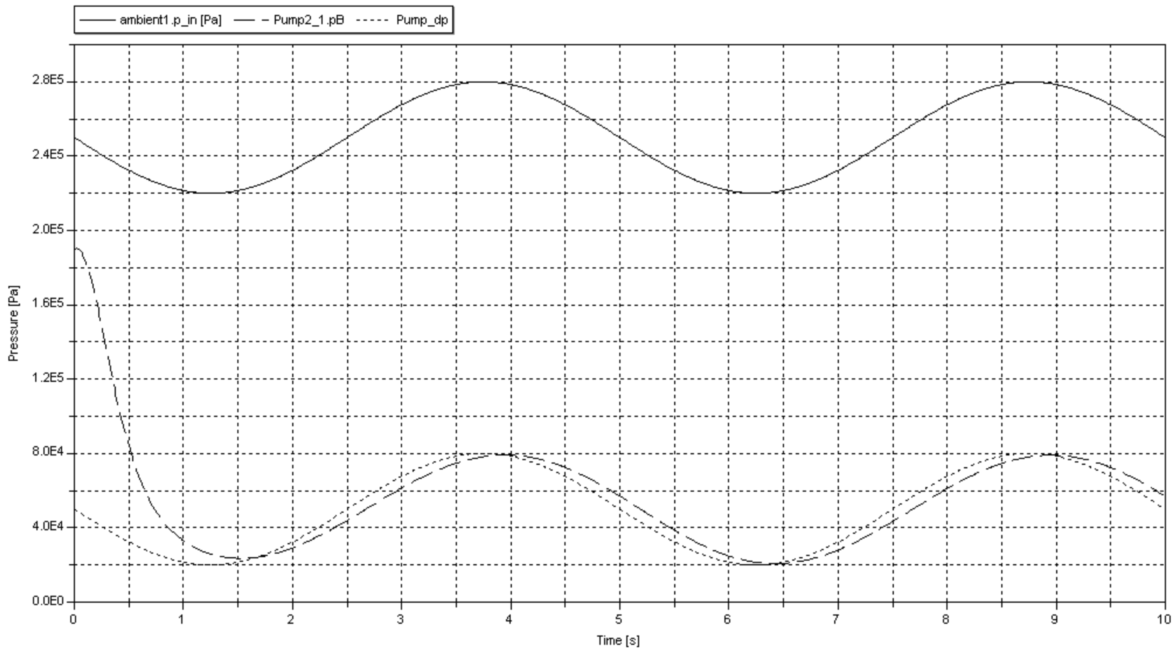

Figure 5.2.2: Pump behaviour

The variable 'ambient.p_in' shows the range where the pump drain is changing its pressure.

The variable 'Pump_dp' is the difference of the pressure on the drain and feed of the pump. It is just the difference of pressure of the both 'ambient1' and 'ambient'. The variable 'Pump2_1.pB' can be interpreted as an “acceleration pressure difference” that is the driving force of the pump. Inside the pump this variable is calculated from the pump characteristics and it is the pressure that the pump is supposed to give for a given flow in steady state.

The figure 5.2.2 shows the dynamics of the system with pB and the difference against the Pump_dp.

The mass flow inside the pump is changing with the pressure changes. When the pump has a big pressure difference between the feed and the drain, the mass flow will be smaller because the pump energy is used to pressurise the fluid. Because of this we can see that the mass flow is oscillating with higher mass flow when the pressure difference is lower (Figure 5.2.3).

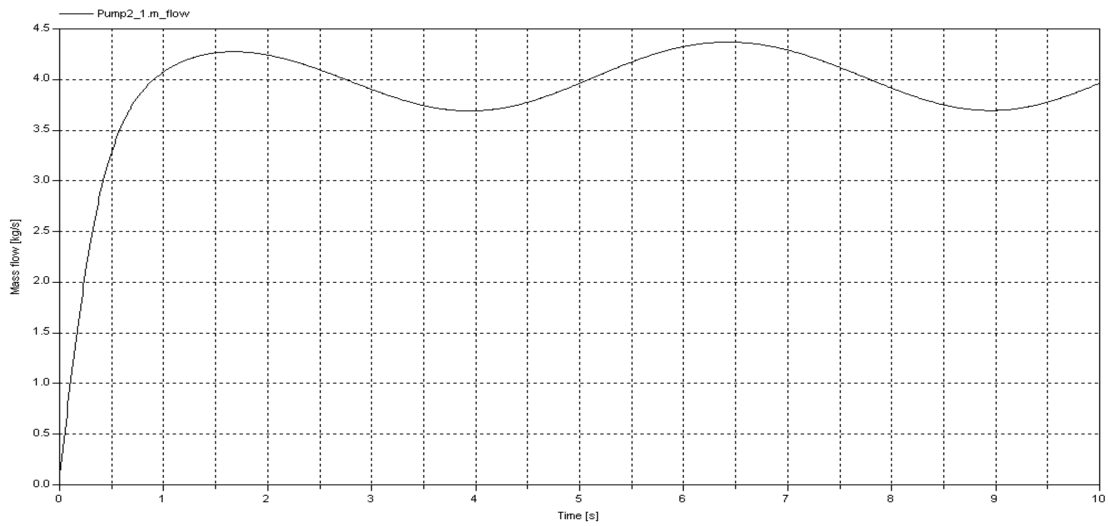

Figure 5.2.3: Mass flow through the pump

At time zero there is no mass flow and it increases until it reaches the steady state point of the pump characteristic. See the oscillations caused for the sinusoidal change of pressure in the figure 5.2.3 (using the same boundary conditions as in plot 5.2.2).

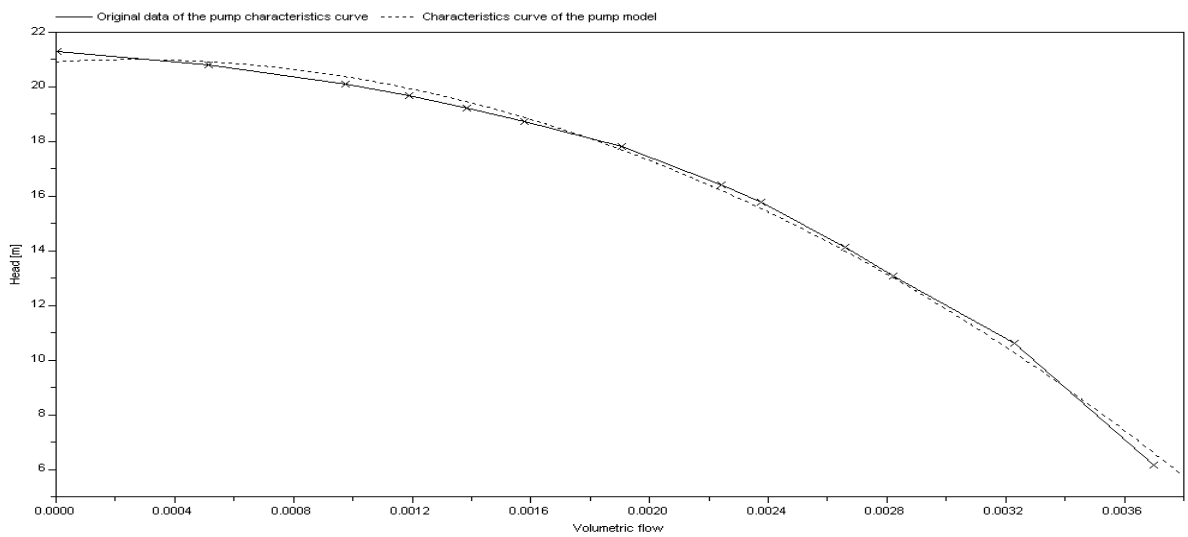

Figure 5.2.4: Pump characteristic curve

Figure 5.2.4 shows the measured flow characteristic data supplied by Daimer Chrisler, and the characteristic curve computed from the data.

5.3 Expansion Volume Test Model

To test the expansion test volume, the same boundary conditions can be used as in the pump. With two ambient components and a pressure loss in between, the drain pressure of the pressure loss is a controlled pressure where the expansion volume can be connected to see its behaviour (Figure 5.3.1).

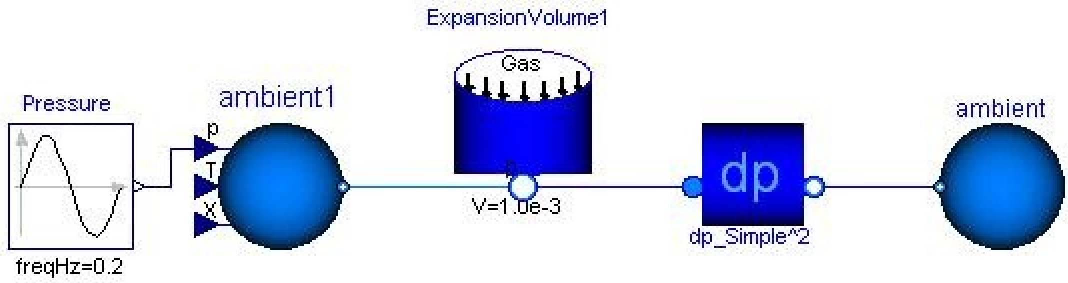

Figure 5.3.1: Dymola scheme of the expansion volume test

Setting the initial conditions the expansion volume has a volume of gas and a volume for fluid. The pressure variates this volumes. If we check the volumes inside the expansion volume we can see how they act if they are subject to the pressure variations (Figure 5.3.2).

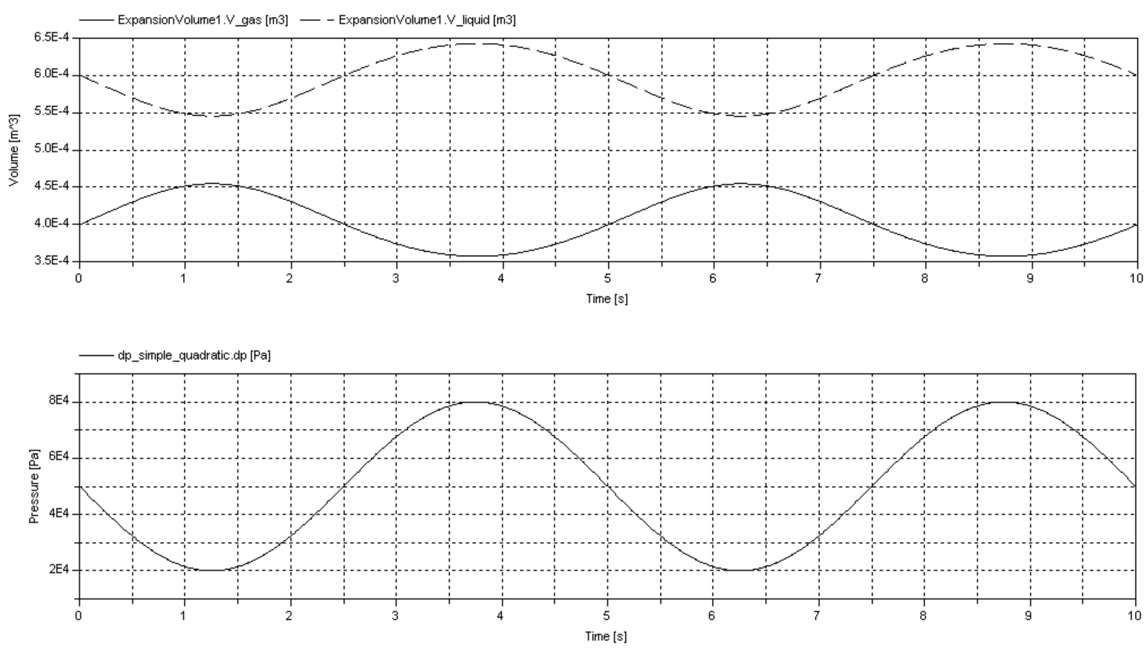

Figure 5.3.2: Expansion volume behaviour

The gas volume is higher when the pressure is lower because of the expansion of the gas. Then the volume of the liquid decreases and vice versa.

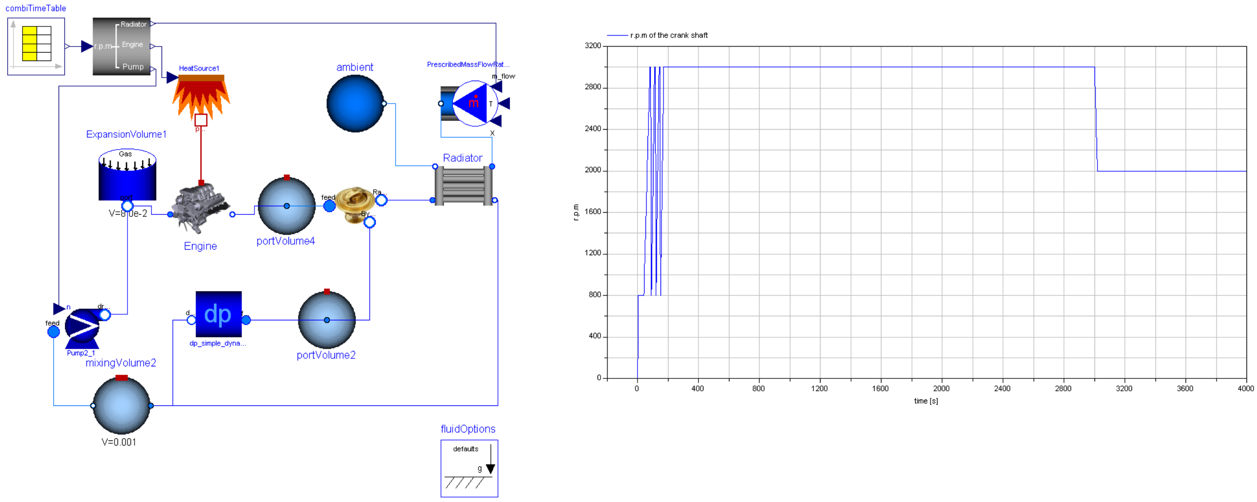

Figure 5.3.2: Engine cooling system and r.p.m map

To check the behaviour of the expansion volume in an engine cooling cycle, the engine cooling cycle presented in the introduction will be used, with the same r.p.m map (Figure 5.3.2).

The behaviour of the volume depends on the pressure on the drain of the pump. We have to set a reasonable initial conditions of the Expansion Volume. The initial pressure of the gas to calculate the mass of gas (inside the model) will be similar to the initial pressure of the pump.

The expansion volume makes sure that pressure variations upstream of the pump are smooth. The behaviour of the volume and the pressure in the pump drain (that is the same of the expansion volume) is plotted in the figure 5.3.3.

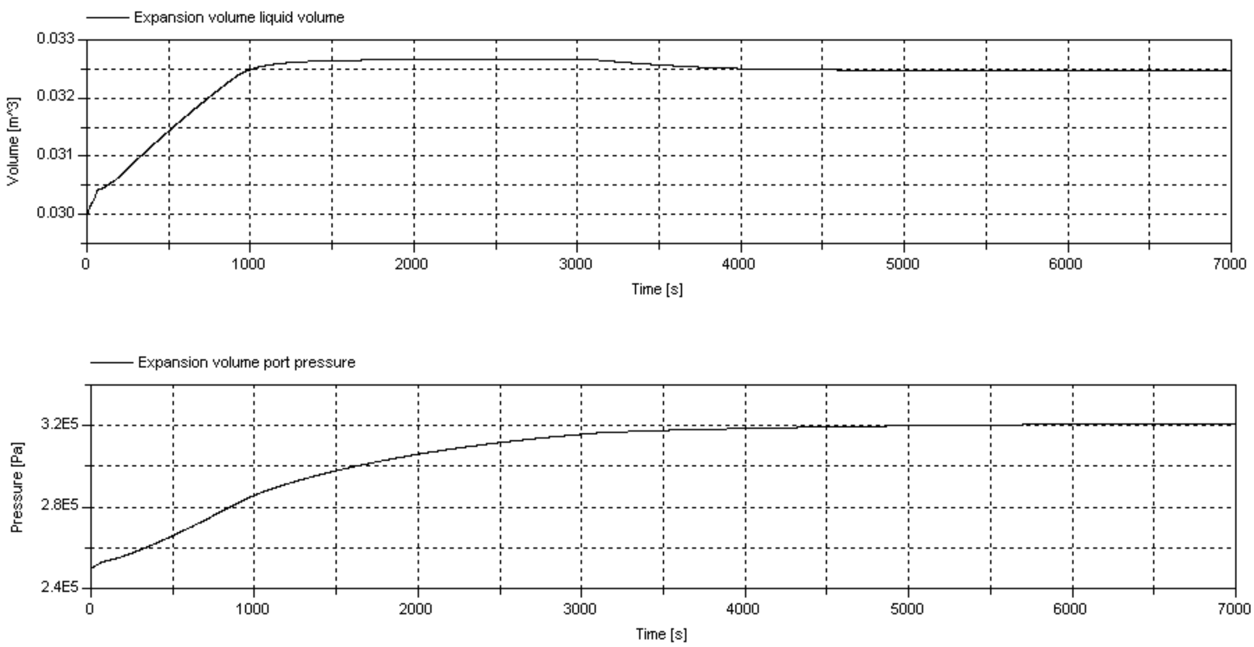

Figure 5.3.3: Volume of liquid and pressure of the expansion volume

When the pressure rises the volume of liquid increases and compensates the sharp r.p.m changes of the pump. During start, the pump speed rises sharply and there are gear changes. The liquid volume of the expansion volume fluctuates because of the gear changes as it is plotted in the figure 5.3.4.

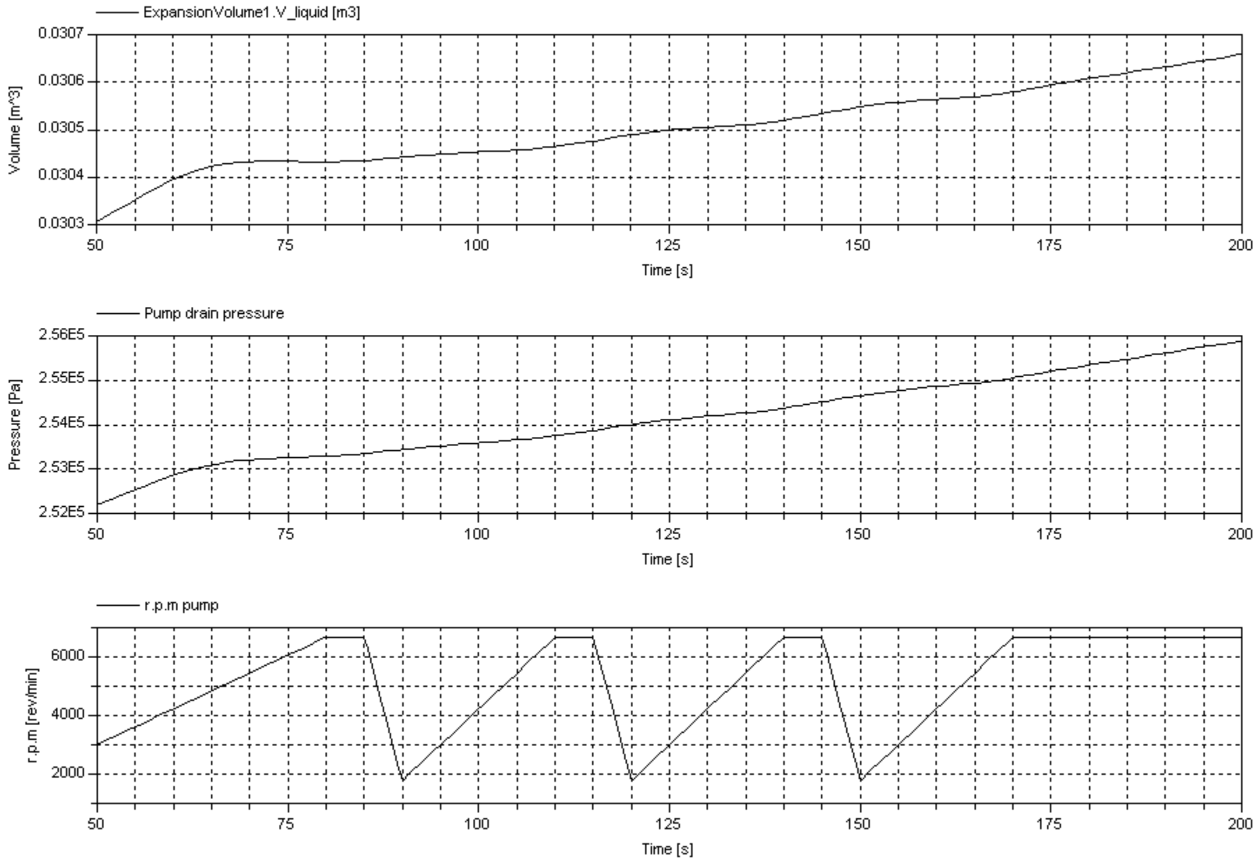

Figure 5.3.4: Liquid volume fluctuations due to the gear changes and pump r.p.m

The volume helps to keep the pressure fluctuations small. The overall pressure level increases because of the fluid expansion due to rising temperatures.

The total volume of the expansion volume for this tests is 8.0e-2 m^3, and the gas takes 5.0e-2 m^3. The temperature is 293.15 K and the start pressure of the gas is 2.5e3 Pa. The other parameters are the default ones where the mass of gas is calculated through the ideal gas formula, the time constant for heat transfer gas-liquid is 10 s as a default, the default gas constant is 287 J/(kg.K) and the heat capacity is calculated through the gas constant.

These parameters are exaggerated, not real, but we use them to see the behaviour of the component. In section *5.4 Engine Cooling components analysis* this component and the system will be analysed with more realistic parameters.

If the expansion volume is bigger or the initial pressure is smaller, the volume compensates the circuit pressure in a different way. We can see what happens changing the volume and the pressure.

To see the behaviour of the pressure in the expansion volume, a variation of the initial volume of gas can be done. The pressure will be different in the gear change zone when the initial gas pressure is 7.0e-2 Pa and when it is 5.0e-2 Pa (Figure 5.3.5).

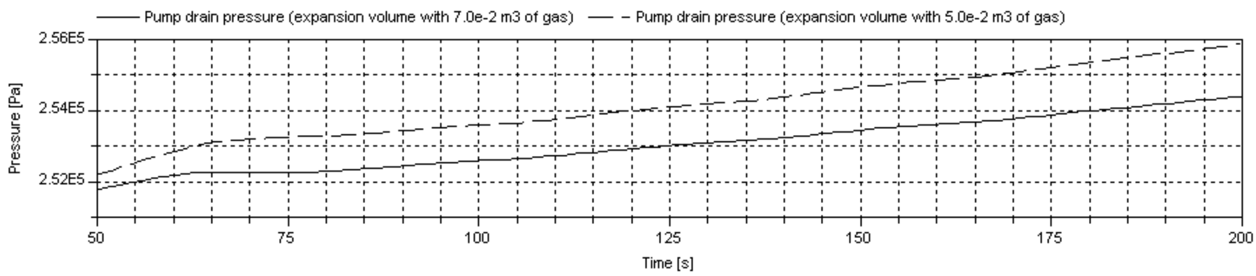

Figure 5.3.5: Gas volume effect on the pressure of the pump drain.

The pressure on the pump drain is smoothed by the bigger capacity of compression of the gas cavity in the expansion volume. The pressure will be smaller. This happens because the gas volume allows less displacements if it is smaller, so the pressure will rise with a steepen gradient.

The effects of choosing a different volume of gas can be seen with this model. This could be useful to decide the component we want to use, looking on the pressure effects of this component on the whole circuit.

For example lets see what will happen if we have winter conditions or normal conditions. Imagine that this system is used at 25ºC or at -10ºC. The behaviour of the system will be different, and the expansion volume that is needed can be different. In reality the vehicles are used in a wide range of temperatures, because of this it is good to check what happens at different conditions but the expansion volume is the same, it is not changed for different conditions. See figure 5.3.6.

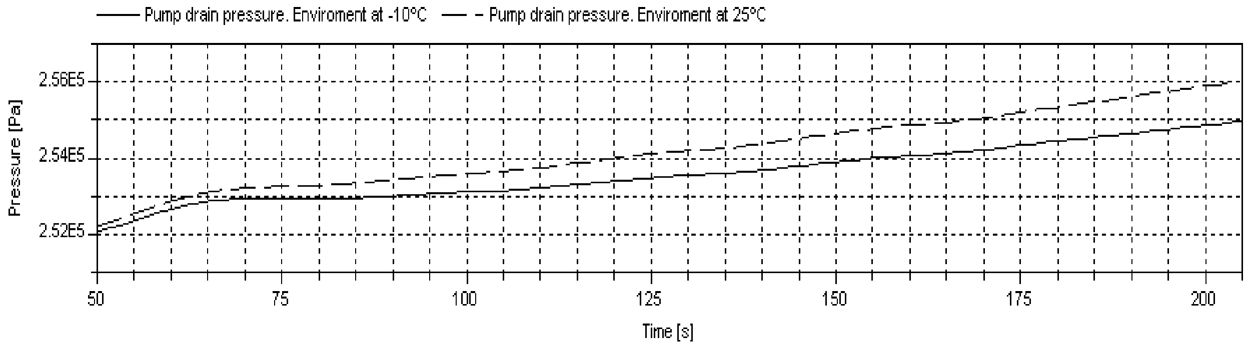

Figure 5.3.6: Pressure in the pump drain at different temperatures

The pressure at the pump drain will be the same as the expansion volume. If a cycle at high temperature or low temperature is used, it can require different characteristics of the expansion volume (to be optimal). The pressure in a system that works at low external temperatures will be smaller, and because of this it can use an expansion tank that is smaller, with a smaller gas mass inside, although in real vehicle it won't be changed because it is wanted to work well in all conditions.

It can be good to check which the effects of having a bigger gas chamber in the engine are. Compare the temperatures and the behaviour of the engine cooling in both conditions. The volumes that are used are exaggerated, the real ones are much smaller, but this ones are good to see the behaviour in clearer.

The temperature in the first discretization and in the last for a initial volume of gas of 5.0e-2 m^3, and 7.0e-2 m^3 are displayed in the figure 5.3.7.

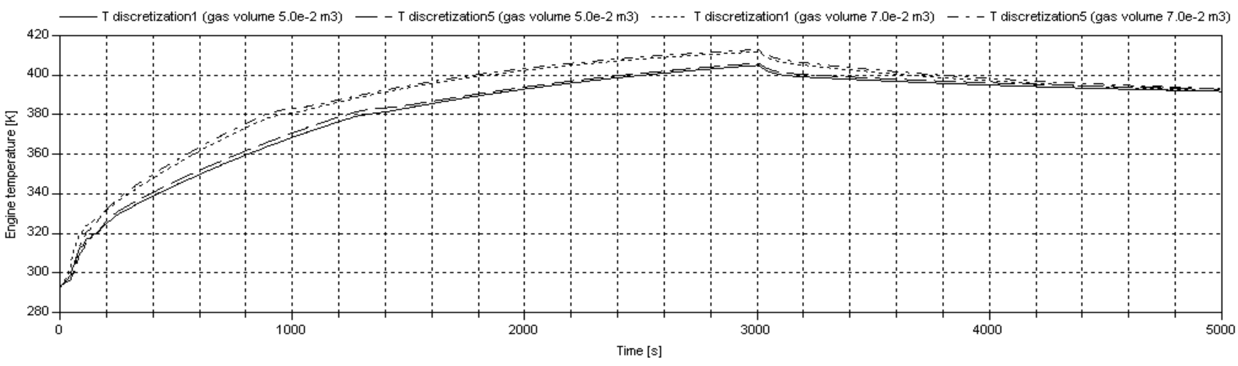

Figure 5.3.7: Engine temperature for different expansion volumes

With the smaller gas volume there is a better refrigeration of the engine, only around ten degrees, but this shows that the expansion volume can have an indirect effect in the engine cooling effectiveness. This difference of temperatures can happen because of the different working point of the two models with different gas volume. The increase of pressure can increase the flow through the radiator branch and cool down the engine better. The engine is cooled with better results but the pressures increase, and this could not be good. The user can use the library to find an optimal component for each engine cooling cycle depending on its work conditions, or he can just check the behaviour of the cycle in different conditions.

5.4 Engine Cooling Components Analysis

The other components will be analysed together to see how they interact.

Using the same engine cooling cycle and the same map, as in the other chapters, we can see what happens in the engine, the radiator and the thermostat valve.

The external air temperature is constant and set to 25ºC. The behaviour of the system will change a lot with the external temperature changes.

The engine block is at one initial temperature and will be heated by the internal combustion and it will increase the temperature. The engine cooling has to control this temperature and keep it within safe limits. It is important to see what happens in the model and see how the engine mass temperature changes and the effectiveness of the engine cooling.

Figure 5.4.1 show that engine block temperatures:

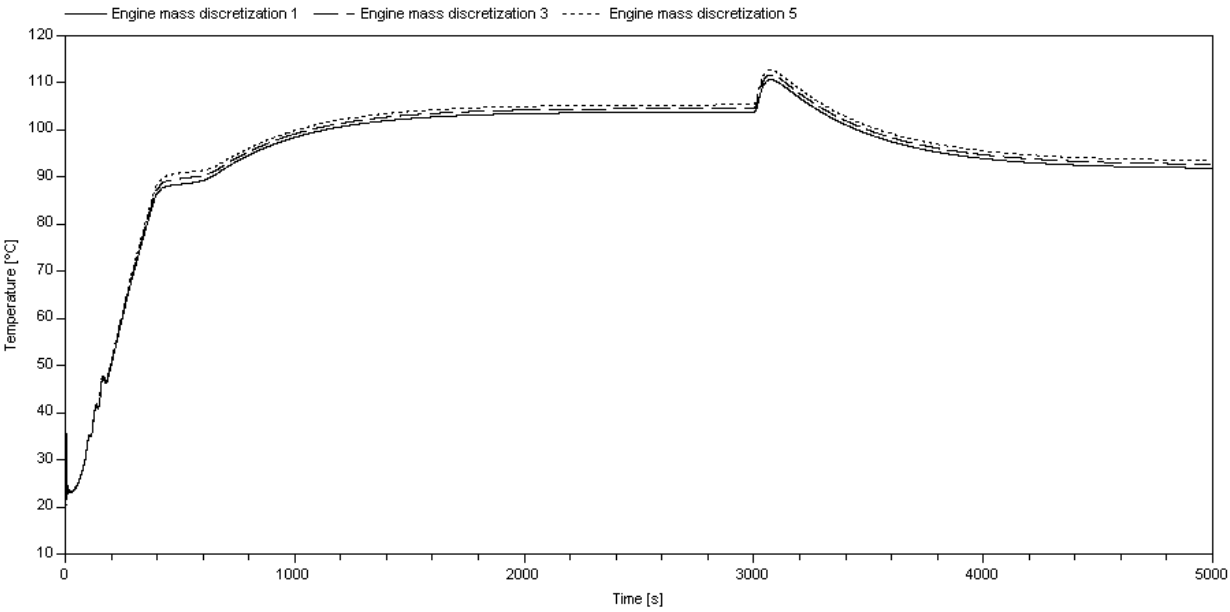

Figure 5.4.1: Temperature of the engine in the different discretizations of the mass

The mass is heated and its temperature is increased. The engine cooling cools it down and reaches an equilibrium. The discretized metal mass elements at the beginning of the flow have a lower temperature than the ones at the end. This is because the coolant gets heated for the first components and there is a temperature difference between the first and the last component that is about 1 or 2 degrees in this example. This is what happens in real engines. The temperature is not ideal in the engine mass and has different zones with different temperatures. This is not the ideal condition, but the designers have to find a good average temperature for the engine.

In the beginning the temperature of the first discretization is higher than the temperature on the last discretizations, but then changes when the engine temperature rises (Figure 5.4.2).

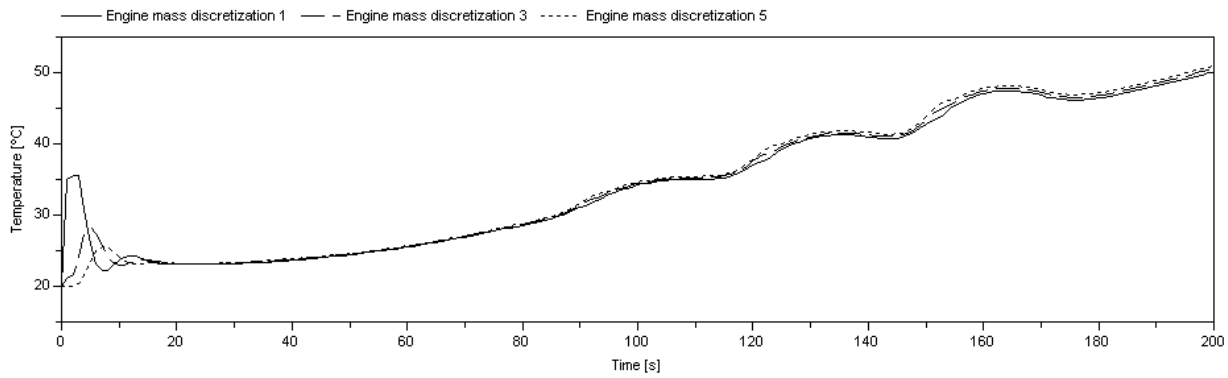

Figure 5.4.2: Engine temperatures detail for the gear changing zone

This happens because at the start the temperature of the fluid is lower than the temperature of the engine, so the coolant is cooled, and it takes away less temperature on the first discretization.

The model can be used to check the effects of the variation of the parameters. In the convection component there is the heat transfer area in m^2, and the heat transfer coefficient in $W/(m^2.K)$. The heat transfer area is changed and there are temperature effects, for example in the 3rd discretization of the engine mass (Figure 5.4.3).

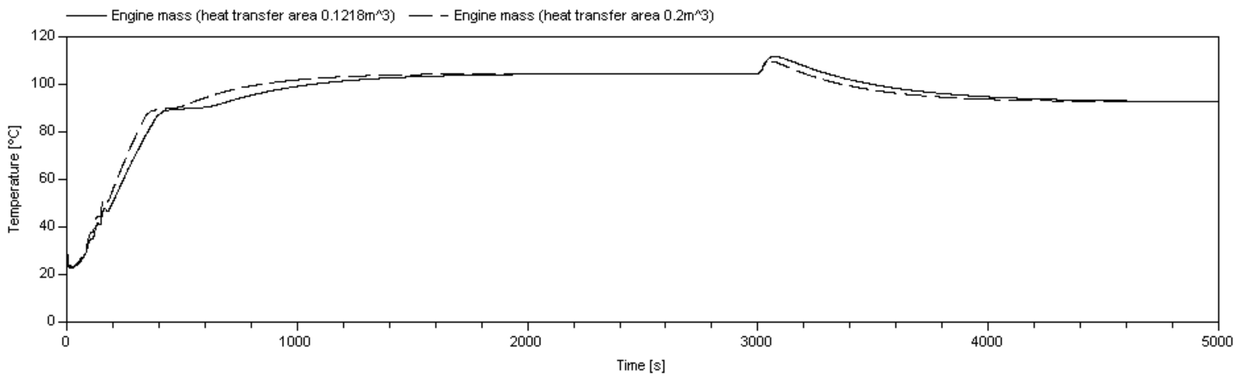

Figure 5.4.3: Engine temperatures with different heat transfer areas

With 0.1218 or 0.2 m^2 of heat transfer area there is a temperature difference of about 10 degrees in the engine block, but only in the zones where the temperature changes faster. When the area is higher the temperature of the block increases slower because it is more cooled.

At about 3000 seconds there is an increase of the temperature of the engine that is higher in the model with a higher heat transfer area. Remember that at 3000 seconds the r.p.m decreases from 3000 to 2000. This is a consequence of the decrease of the air flow through the radiator. The radiator has less cooling capability and the thermostat valve has to open the radiator branch completely to cool down the engine. It takes some time to get a lower temperature than before although the heat going in to the radiator is lower.

Another important parameter is the mass of the engine. The engine mass is the parameter with the longest influence on the engine cooling behaviour.

When all this parameters of the engine are changed, there are also changes in the behaviour of the whole circuit. For example if the temperature of the engine is higher the thermostat valve will open the radiator branch more and there will be more flow in the radiator.

The radiator is a component that is very similar to the engine, the components almost have the same internal structure. The radiator dissipates the energy from the coolant to the surrounding air in a cross-flow arrangement heat exchanger. In the radiator there are different parameters like the heat capacity of the heat capacitor component, or the convection areas, the heat transfer coefficient for the convection component, etc. Obviously these parameters will effect the behaviour of the radiator and the behaviour of the engine cooling system.

– **Summer and winter conditions test:**

A realistic performance test for the system is to investigate the behaviour of all components of the circuit if the cycle is driven under winter conditions and then simulate under summer conditions. This means that the air that goes into the radiator will be at a low temperature for winter conditions and a high temperature for summer conditions. This test is used to see if the results are reasonable.

The r.p.m map will be the same as in the previous engine cooling model, and the temperatures of the air will be -20ºC for the winter conditions, 25ºC for the normal conditions and 40ºC for the summer conditions.

The most important parameters of the components:

The **engine** is a 100kg mass engine with 0.1218m^2 contact area between the engine and the coolant. The heat transfer coefficient parameter is 2000 W/(m^2.K) and the heat capacity of the engine is 500J/(kg.K). The nominal heat flow in the engine is 26.6 kJ/s.

The **expansion** volume has a gas volume of 0.5e-3m^3 and a total volume of 1.5e-3m^3.

The **pump** has its nominal point at 6672 rev/min and has the pump curve characteristics according to data provided by Daimler Chrysler.

The **radiator** has a mass of 3 kg and is made of aluminium, with a 5 m^2 of contact of the air and the pipes (convection area) and 0.8 m^2 of convection area for the coolant. The nominal air mass flow is 0.5 kg/s. The convection area for the air side is 7 m^2 and for the coolant side is 0.8 m^2. The heat transfer coefficient for the air side is 150 W/(m^2.K) and for the coolant side is 2000 W/(m^2.K).

The **thermostat valve** uses the hysteresis data provided by Daimler Chrysler.

The whole circuit has around 3.5 litres of coolant. The amount of coolant can change the behaviour of the system, basically the time response.

The figure 5.4.4 shows the valve position depending on the air temperature.

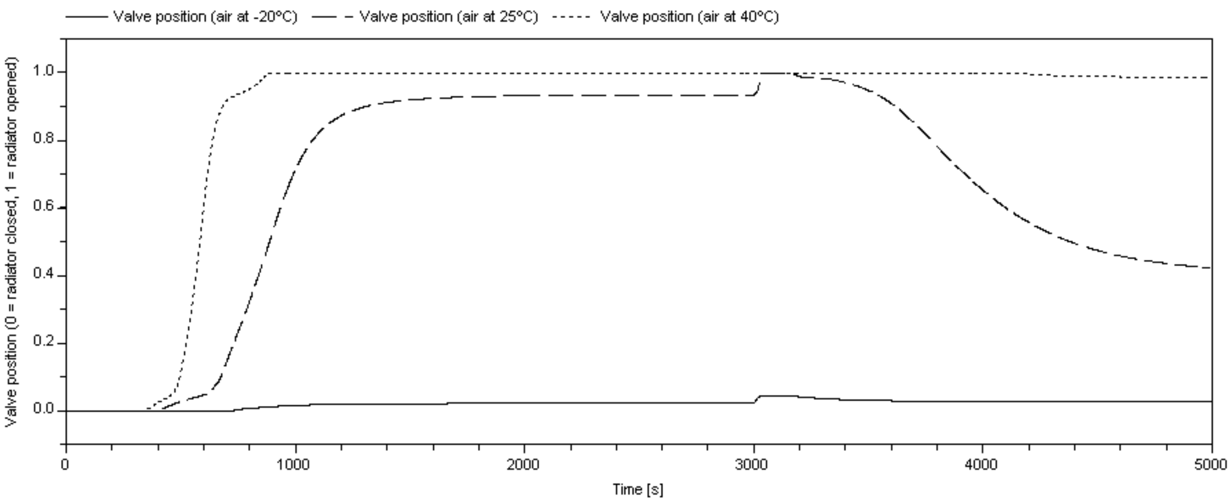

Figure 5.4.4: Valve position depending on the air temperature

The results are reasonable. For the summer conditions the thermostat valve is completely open, which means that most of the coolant goes through the radiator. For the normal conditions the valve opens almost completely but only for a short period. For the winter conditions it just opens a little bit.

The engine will have different temperatures depending on the external conditions. In the winter conditions the temperature of the engine will be lower than in the summer conditions. This is due to the higher cooling capacity when the ambient temperature is lower.

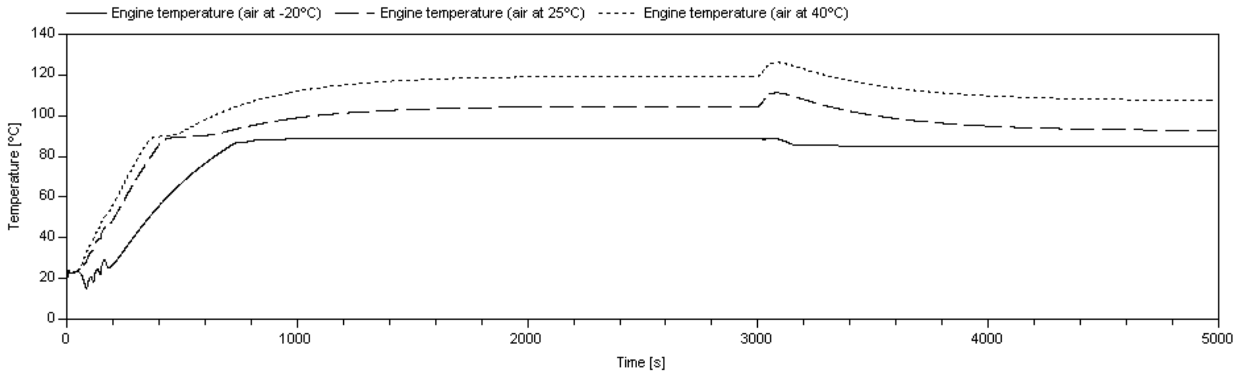

Figure 5.4.5: Engine mass temperatures with different air temperatures

The air that goes into the radiator goes out at a different temperature depending on the energy given by the engine. The air goes in at 40ºC for the summer conditions, at 25ºC for the normal conditions and at -20ºC for winter conditions. See at what temperatures the air exits from the heat exchanger in the different cases in the figure 5.4.6. This air flows across the engine, and could be interesting to observe the temperatures under all conditions.

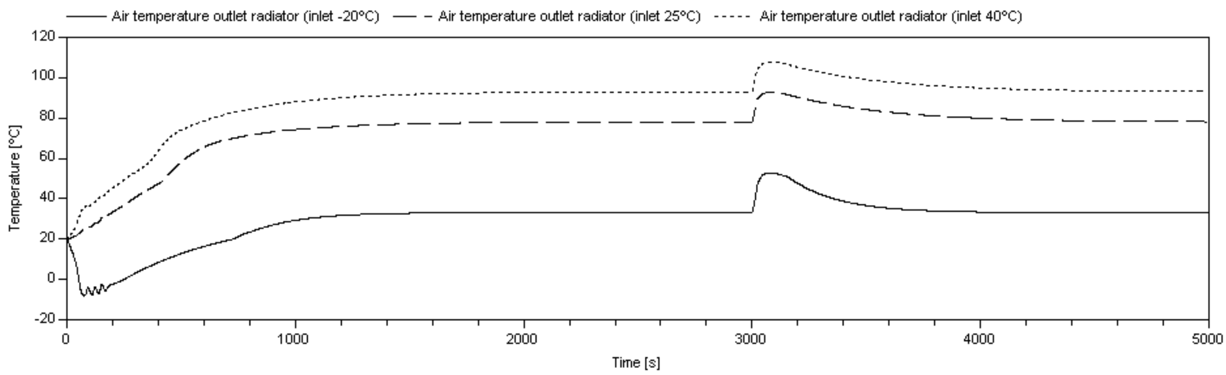

Figure 5.4.6: Air temperature in the outlet of the radiator

The temperature at 3000 seconds rises sharply. At this point the r.p.m of the crank shaft is reduced from 3000 to 2000, and the incoming heat to the circuit decreases, but at the same time the air flow in the circuit is reduced because of the decrease of the r.p.m of the fan. For this reason there is an increase of the air temperature for some minutes until it gets down again. The temperature at the end point is higher, but the temperature of the engine is lower. This happens because of the high effect of the reduction of the air flow in the radiator.

When the system is in summer conditions, it is logical that the pressure in the cycle gets higher than when it is in winter conditions. The gas volume in the expansion vessel will be compressed more or less. In the figure 5.4.7, where the gas volume of the expansion volume and the pressure are plotted, the volume of the circuit that works in summer conditions is smaller than the one at normal conditions, the one in normal conditions is smaller than the one in winter conditions. The rise of the gas volume in the expansion volume determines the maximum system pressure level.

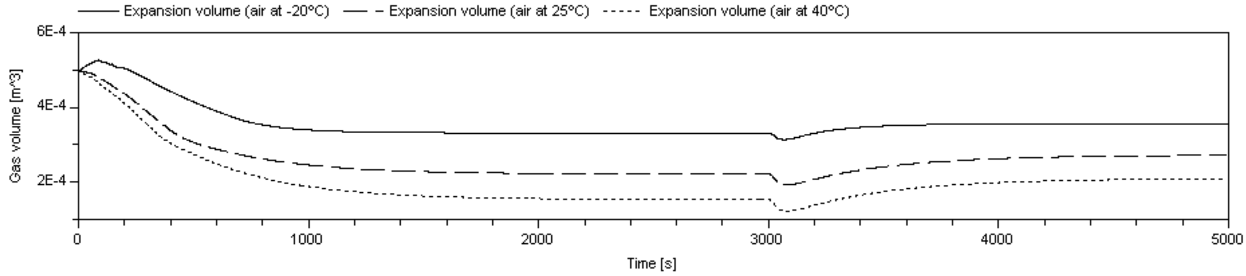

Figure 5.4.7: Expansion volume results: Gas volume and pressure

The pressures also have the expected behaviour being higher in the summer conditions than in the normal conditions and lower in the winter conditions. This pressure has a high difference depending on the conditions. The pressure working point is highly influenced by the temperature of the air that cools down the coolant in the radiator.

The pressure in the expansion device is the same as in the pump drain. It is interesting to see the behaviour of the pump and consumption of the power in the crank shaft and the pressure that the coolant reaches on both sides of the pump. It is important that the coolant is not at a lower pressure that its evaporation pressure to avoid cavitation. The pump will not work properly if the coolant is evaporated or if there is air inside the coolant. The circuit has a limit pressure that can not be exceeded or some pipe fittings will brake.

The pressure downstream of the pump is the pressure plotted in the figure 5.4.7, and the pressure in the pump feed is the pressure plotted in the figure 5.4.8 for the different ambient conditions.

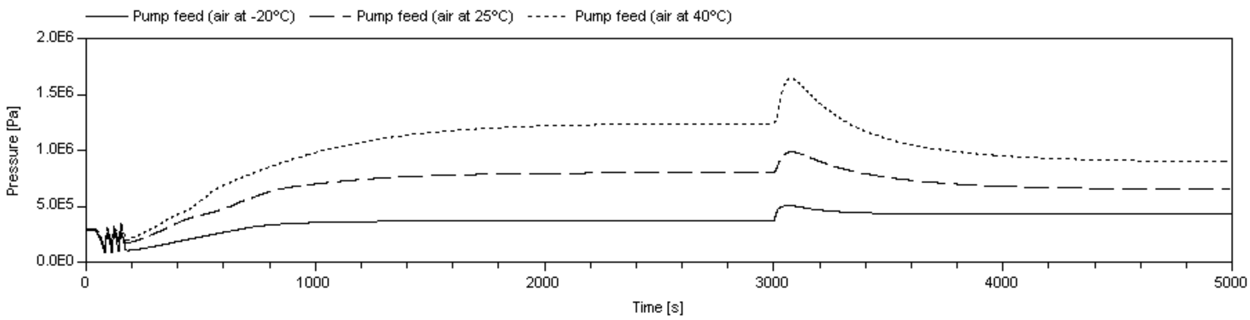

Figure 5.4.8: Pump feed pressure for the different air temperature cases

During the gear change the pressure is lower than 1 bar, and the evaporation pressure of the coolant is near 1 bar. The user of the library, if this case happens, will have to analyse this situation to see if there will be a cavitation problem in the real system.

Another important parameter is the pump power consumption. If the pump is driven by the crank shaft it is interesting to know the parasitic consumption by the pump.

In summer driving the power consumption of the pump is higher than in winter conditions. This can be seen in the model (see Figure 4.5.9).

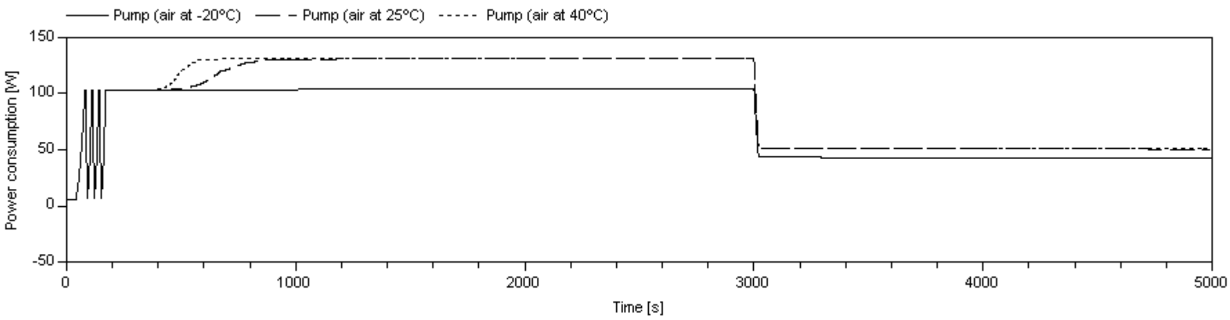

Figure 5.4.9: Pump power consumption for different ambient conditions

The power consumption in the experiment with temperatures of 40ºC and 25ºC is higher than the one at -20ºC. This is a consequence of the cycle working point of the three different cases. The power consumption gets higher with the mass flow through the pump (see Figure 5.4.10).

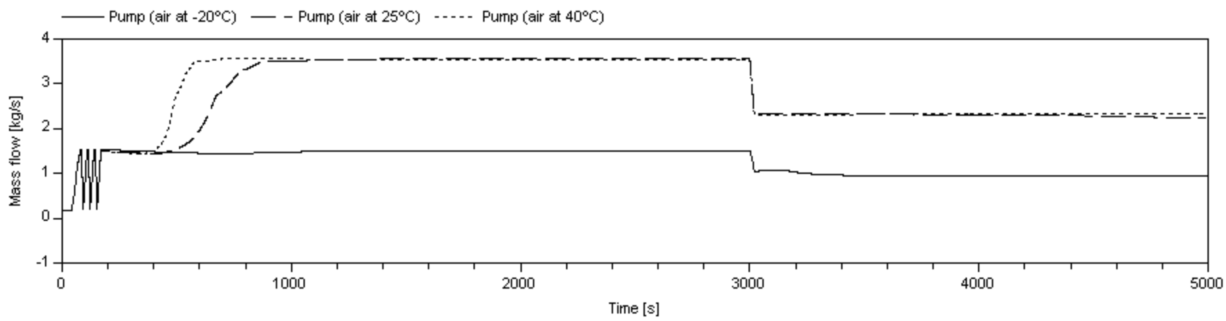

Figure 5.4.10: Mass flow through the pump for different ambient conditions

6 Conclusions

This thesis is the description of the development of a basic Engine Cooling library and some test models. A simple engine cooling system is built and produces reasonable results. The given data to test the Engine Cooling model is for a complete model, for this reason the test of this library is made checking that the results are reasonable. Some simple components were tested with real data separately, but not in the complete Engine Cooling system.

It is easy to build models with a library when the components are robust and consistent, thus the library components have to be well designed and tested. It is also important to set reasonable parameter values, for this reason there are some models and tests in the library that give a reference of this values. The user of the library can use the basic model of the Engine Cooling library and the tests to extend them and build a more specific and realistic model.

The expansion volume is a necessary element for the closed loop models because it compensates the expansion of the coolant due to the temperature changes.

The numerical problem of long non-linear equation systems in the models was overcome by introducing dynamic equations for mass flow, physical equivalent of dynamic momentum balance, in some components.

There was a lot of work to include the large list of empirical pressure drop correlations needed for a complete engine cooling library, but of little academic interest.

The thesis work is a good base for a complete and useful Engine Cooling library. The next steps to follow are to build a more complex and realistic engine and radiator models. Some components will have to be added like holes, pipes and different kinds of bends.

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