Engine thermal shock testing prediction through coolant and lubricant cycling in Dymola

Eduardo Galindo[1] Rodolfo Soler[1] Alessandro Picarelli[2] Victor Avila[2]

[1]AVL IBERICA SA - VALLADOLID, Spain, {Eduardo.Galindo, Rodolfo.Soler}@avl.com
[2]Claytex Services Ltd. – Leamington Spa, UK, {alessandro.picarelli}@claytex.com

Abstract

In this work, an acausal multi-domain physical system model is used to study the interaction between an internal combustion engine operation and a range of cooling and lubrication system thermal cycling scenarios. Although the model can be used for modelling a wide range of scenarios, this paper concentrates on the application of engine thermal shock test dynamics prediction through coolant and lubricant cycling. An internal combustion engine is load-controlled on a dynamometer. Coolant and lubricant temperature transients are imposed on the engine system. Using freely available and commercial Modelica Libraries within the Dymola environment, the systems integration of the coolant rigs, lubricant rigs and engine is achieved. The rigs and the controllers are validated against test data to create predictive models of such systems for test virtualisation. This allows the user to develop and define control strategies for the tests from desktop, prior to engaging in laboratory tests.

Keywords: Engine testing, thermal-shock, control system development

1 Introduction

Engines need to work under a variety of temperature conditions. Some engine failure modes are caused by temperature cycling which in turn causes thermal expansion and contraction of the components. This phenomenon can induce mechanical stresses which in extreme cases can lead to component failure.

This paper builds on (Picarelli et al, 2014) and seeks to validate engine coolant and lubricant conditioning rigs for virtualisation of test scenarios in order to predict the system behaviour and to tune the control systems prior to the real testing taking place.

In addition to previous tests, where only the engine coolant was conditioned, in this paper we present thermal shock testing where the dynamics of the lubrication system are also included.

2 Thermal shock testing

Many manufacturers carry out thermal shock tests to understand and prevent component failure, as well as to accelerate durability testing of engines and engine components, including cylinder-head gaskets.

Thermo-mechanical fatigue is the term used to describe the type of fatigue in which temperature is varied throughout a cycle. The maximum tensile strain occurs at the same time as the maximum temperature.

Maximum compressive strain occurs at the minimum temperature. The main factor causing thermos-mechanical failure is a large number of temperature cycles. As in fatigue testing, it is possible to accelerate thermal cycling failure modes by increasing the frequency or amplitude of the thermal cycles.

These thermal tests are used to simulate critical conditions inside the engine by circulating a coolant flow with very large temperature gradients occurring over short periods of time (e.g. 30ºC to 120 ºC). This cycle is repeated a several times.

The main task performed in this study is simulation of repeated hot/cold thermal cycles. The engine is cycled between rated power and idle speed. The coolant and lubricant are also cycled between hot and cold temperatures by means of external conditioning units.

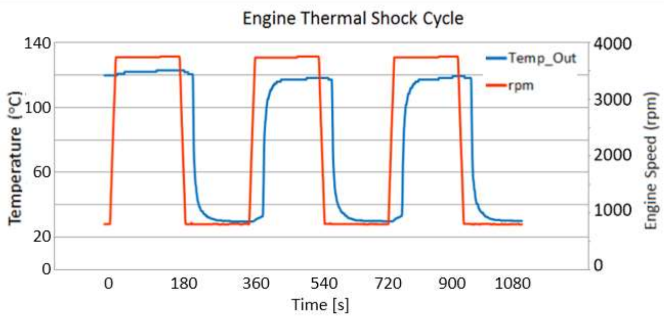


Figure 1. Example of an engine thermal shock cycle. Engine speed shown in red and coolant outlet temperature from engine shown in blue (ºC).

The temperature gradient in the warm up and cooling down cycles is critical to generating the mechanical stresses applied to the engine due to the thermal shock. These kinds of tests allow manufacturers to reproduce the whole life of an engine in about 500 hours for a light duty passenger car and 2000 hours for a heavy-duty vehicle. Manufacturers expend great efforts in obtaining a good correlation between specific tests and the actual lifetime of an engine. Once the correlation is completed, the test must be performed as accurately as possible to preserve this correlation.

Figure 2. Cracks in the valve seat produced by thermal stress.

3 The need for a simulation model

An accurate and representative simulation model allows us to reduce engineering time for the prediction of new tests and design of new systems giving us the ability to predict the behaviour of a given system before manufacturing it.

This simulation ability also allows us to change the test or equipment parameters and foresee their impact on the results. This way, we can have a better view on how the system will behave, so that any specific issue or change can be adapted quickly and easily.

Furthermore, the simulation model has already predicted some unexpected and unwanted behaviours such as pressure spikes, giving the opportunity to make the necessary corrections early enough, thus avoiding additional engineering efforts and potential system failures.

4 Case Study

The thermal shock rig system in this study was intended to test engines from 60 kW to 120 kW, with an engine mass of 90kg to 120 kg. This power was limited by a maximum torque of 130-250Nm and a maximum speed of 6700rpm.

The actual engine used in the real test was a 4-cylinder gasoline engine with a maximum torque of 130 Nm and a maximum speed of 5600 rpm, yielding a maximum power of 76.2kW.

The physical model is tested in two relatively different scenarios, which are as follows:

1. Thermal-shock test with low temperature gradients for heating and high temperature gradients for cooling, running between 110ºC and -30ºC. This test also includes the cooling of the engine's oil down to -20ºC. In advance we'll refer to this test cycle as "Thermalshock 1":

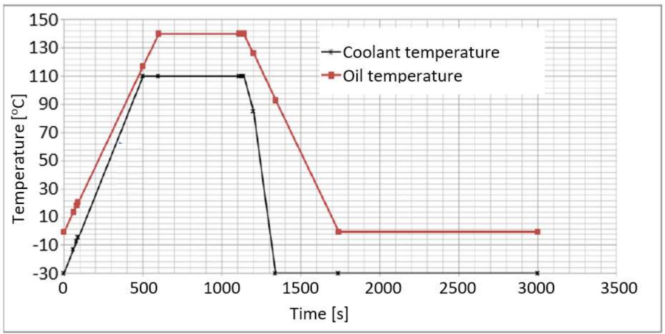


Figure 3. Temperature path for the engine coolant (black) and oil (red) for Thermalshock 1.

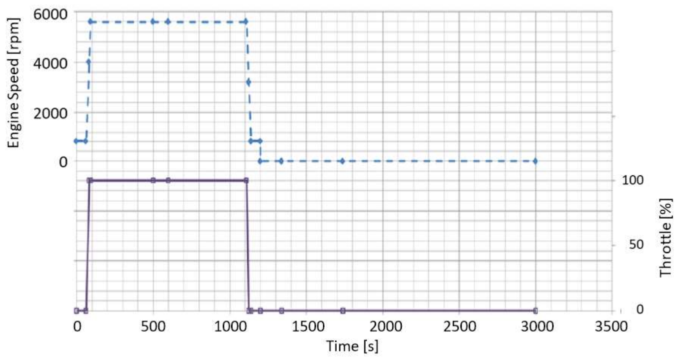


Figure 4. Throttle position of the engine for the Thermalshock 1 (solid line) and engine speed (dashed line).

2. Hot and cold test, with low temperature gradients for both cooling and heating, but with high frequency heat transients produced by quick variation on the engine throttle position. Hereon we will refer to this test cycle as "Thermalshock 2".

DOI
10.3384/ecp17132189

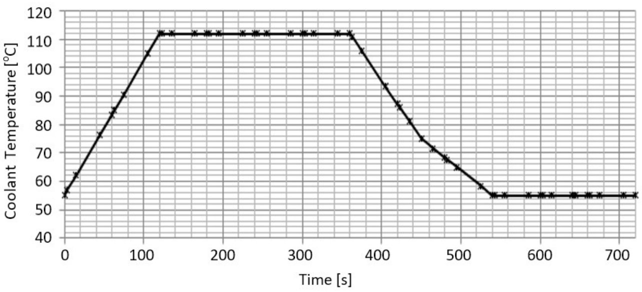


Figure 5. Temperature profile for the engine coolant and oil for the Thermalshock 2.

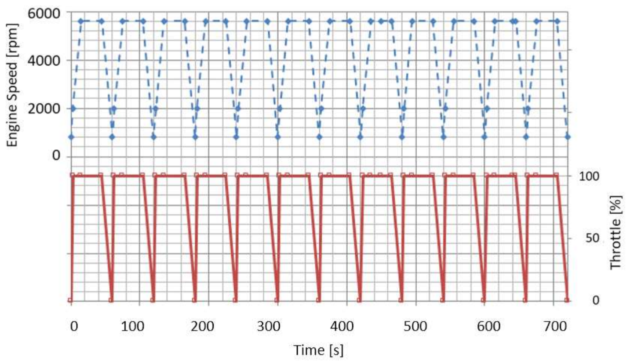


Figure 6. Throttle position of the engine for the Thermalshock 2 (solid line) and engine speed (dashed line).

These two tests are important in order to check the durability of their internal combustion engines, specially focused on the endurance of the head gaskets which are particularly affected by the thermal stress.

4.1 Thermal Shock Equipment Concept

The equipment consists of several fluid conditioning devices all connected to each other and/or the engine (Figure 7). Given that the thermalshock test itself has two well differentiated parts (hot part and cold part), there are two cool-ant conditioning devices and a further device that switches the connection of the engine between them.

Since the oil has to be conditioned too, the engine is connected to a heat exchanger on the gallery connections (the engine's oil pump is responsible for the flow), and to an oil cooling device on the sump.

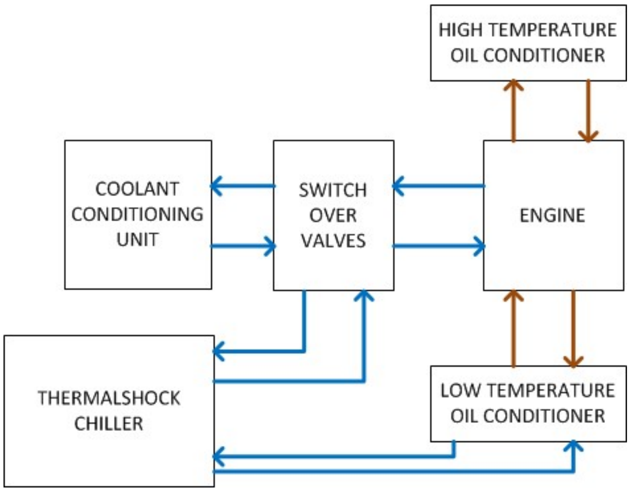


Figure 7. Complete system's simplified P&ID (Piping and Instrumentation Diagram).

4.2 Thermal Shock Testing Equipment

Coolant conditioning unit:

Composed of a pump and a 3-way valve that directs the coolant through a heat exchanger (for cooling) or a heating resistance (for heating).

Switch over valves:

A device composed of several 2 way pneumatic valves that allows the engine to be connected either to the coolant conditioning unit (Consyscool) or to the thermalshock chiller. This way the valves connect the engine to the coolant conditioning unit during the hot phase, and to the chiller during the cold phase.

Thermalshock chiller:

Composed of a water chiller specially designed and built for engine thermalshock testing. A pump flows the coolant from the inertia tank to the engine and to the oil cold heat exchanger.

Figure 8. Thermalshock chiller picture showing the large water tank on the left and the controllers and valves on the right hand side.

The design of this chiller is specially customized for engine testing, with special features like a on-standby design that allows the system to be ready for a thermalshock at any time (Figure 8).

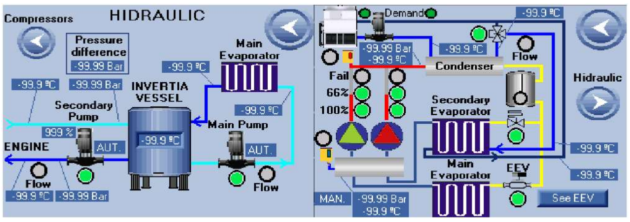


Figure 9. Thermalshock chiller controller screen.

The controller of the chiller is custom-designed for the engine testing process, with several programming parameters, interface with the testing facilities and remote control for operation and diagnosis (Figure 9).

High temperature oil conditioner (see Figure 12):

Consisting of a plate heat exchanger which is connected to a conditioning unit (similar to the engine coolant conditioning unit). The oil is cooled by means of a cold-water heat exchanger.

Low temperature oil conditioner (see Figure 12):

Consisting of an oil pump (variable speed), and a heat exchanger cooled by the same chiller used for cooling the engine coolant.

5 Model Development

5.1 Engine Model

The engine type used on the rig is a 1.8l turbo-petrol inline 4-cylinder engine.

The engine model in these tests is a thermal representation of the real engine which includes heat rejection from combustion to the coolant, lubricant and the engine's thermal mass.

The engine heat release to coolant and lubricant has been defined as a fraction of the crank power and varies depending on engine speed and load. The fraction value is determined from steady state tests by calculating the power required for the coolant and lubricant temperature changes between the inlets and outlets of the engine circuits. The fluid paths within the engine are represented by a single pipe having average diameter of the passageways and the measured total engine pathway volume and surface area. The pipe dimensions are adjusted to achieve the required flow velocities through the engine.

The engine thermal mass used in this study is a lumped thermal mass and is not split by subsystem. More detailed models are available within the Claytex Engines library for studies which require higher level of engine thermal mass discretisation. The engines library was used in a previous study (Dempsey *et al*, 2009; Dempsey *et al*, 2012; Dempsey *et al*, 2013; Picarelli *et al*, 2014).

The coolant pump of the engine is replaced by electric coolant pumps within the rig which can be controlled to deliver specific flows or flow profiles. The lubricant pumped by the engine lubricant pump itself when the engine is running. An electric lubricant pump within the rig is used when the engine is switched off.

The heat transfer from the engine to the coolant is calculated by means of a Nusselt Number correlation, calculated specifically for this engine. The Nusselt Number (Nu) correlation is then used within the pipe model which represents the coolant path within the engine. Due to the fact that the thermal mass of the engine is of lumped type, the volume model used to represent the mass of coolant within the engine has one thermal node. The same Nu correlation can be implemented with multiple node fluid pipes derived from the Modelica.Fluid library should a more detailed thermal discretisation be required and will be the subject of further work when engine CAD data becomes available. This will also increase the predictive capabilities of the model. The exact same Nusselt

DOI
10.3384/ecp17132189

Number heat transfer approach is used for the heat exchangers in the rig model described in section 5.2.

5.2 Rig Model

The thermal shock rig must be able to supply preconditioned coolant to two different flow conditioning units and by means of a switch valves device controls the engine fluid temperatures. The rig described in this paper uses a 2000 litre coolant tank kept at temperature with fixed set-points and an external source which supplies water permanently. The water tank is kept at constant ambient temperature and the coolant tank is kept at low temperature, around -30 ºC (Figure 10. Complete thermal shock rig with water tank (1), coolant tank (2), **Mean Value** engine model (3)). The tanks are required to also smooth out and absorb any temperature fluctuations in the rig, in addition to these two tanks, there are also some small expansion tanks included throughout the rig to absorb any possible pressure, temperature and volume fluctuations.

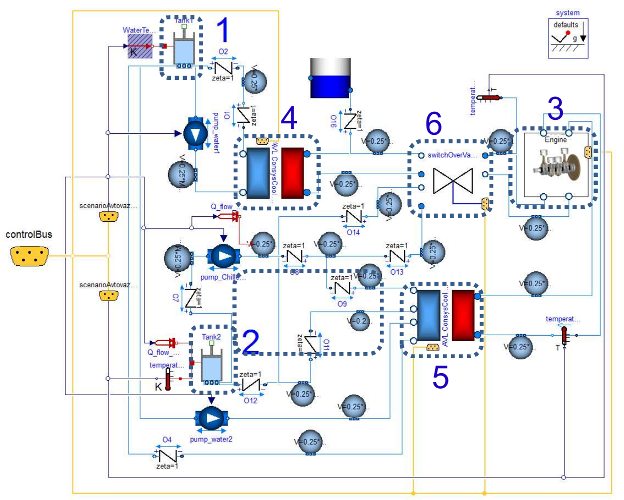


Figure 10. Complete thermal shock rig with water tank (1), coolant tank (2), Mean Value engine model (3), coolant conditioning (4), lubricant conditioning (5) and hot/cold switchover valve (6).

At particular points in the cycle, the switchover valves (Figure 11) model and the internal valves of the conditioning units before described, are controlled to channel either hot or cold coolant through the engine. These changes in coolant and lubricant temperatures yield the required thermal shock for the engine to experience and operate through.

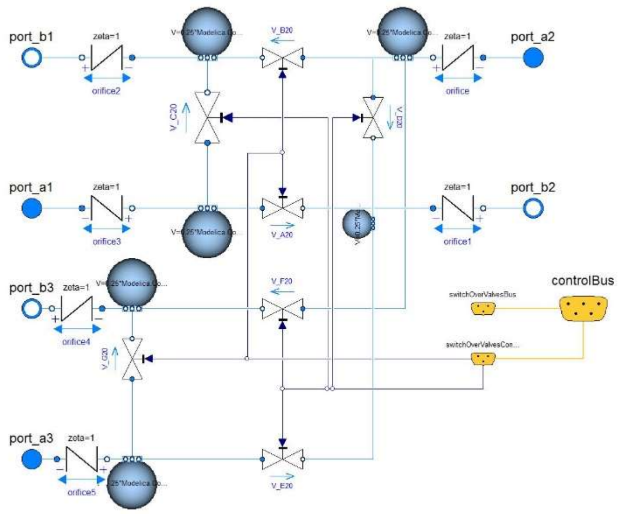


Figure 11. Switch over valve model used in the rig to lead the coolant from the different heat exchangers through the engine.

Both lubricant conditioning units (see **Error! Reference source not found.**) are included in the same model (Figure 12. Oil conditioning unit DiagramFigure 12).

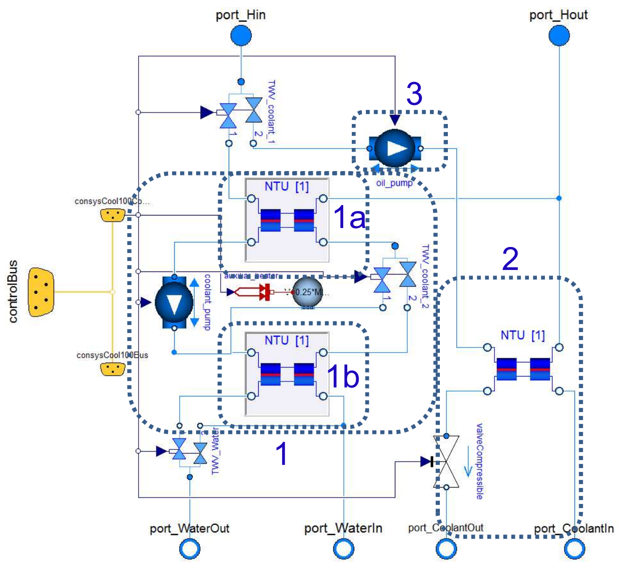


Figure 12. Oil conditioning unit Diagram: High temperature conditioner (1), plate heat exchanger (1a), cold water heat exchanger (1b), low temperature conditioner (2) and electric pump (3).

The rigs are modelled using the Modelica.Fluid and Modelica.Media libraries (Casella *et al*, 2006) with some customized components from the Claytex library which incorporates advanced functionality within the components both for visualization and enhanced model efficiency. The fluids used match that of the rig in terms of properties and are a mixture of 50% Ethylene Glycol and water with linear compressibility for the coolant side and Oil with constant compressibility for the lubricant side.

The controller for the tank cooler is of on/off type and starts to cool with 40.4 kW of power when the tank fluid temperature has deviated from the set point by +1 ^{0}C.

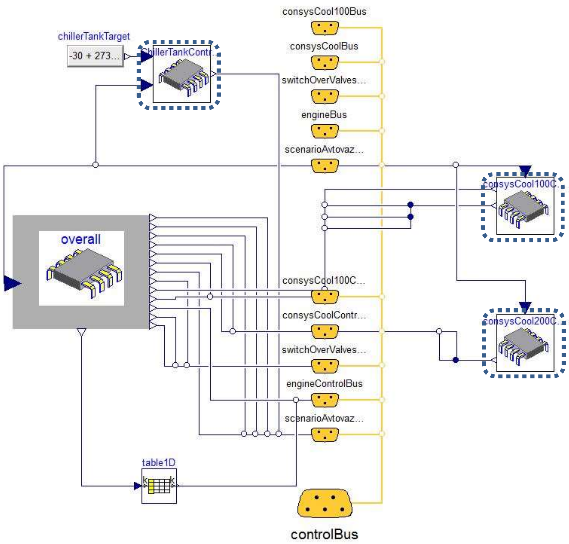


Figure 13. PID controllers for controlling the coolant and lubricant conditioning units and chiller tank to maintain the corresponding fluid temperatures close to the set points.

To control the 3-way valves, within the fluid conditioning devices, a Modelica.StateGraph model was used which is shown below (Figure 14). The valves are operated to route the coolant and the lubricant through the heat exchangers or bypassing them to restore desired temperature targets at particular points in the cycle.

The same type of StateGraph model controls the throttle pedal position which is cycled from 0-100% in a similar phase to the engine speed (Figure 15).

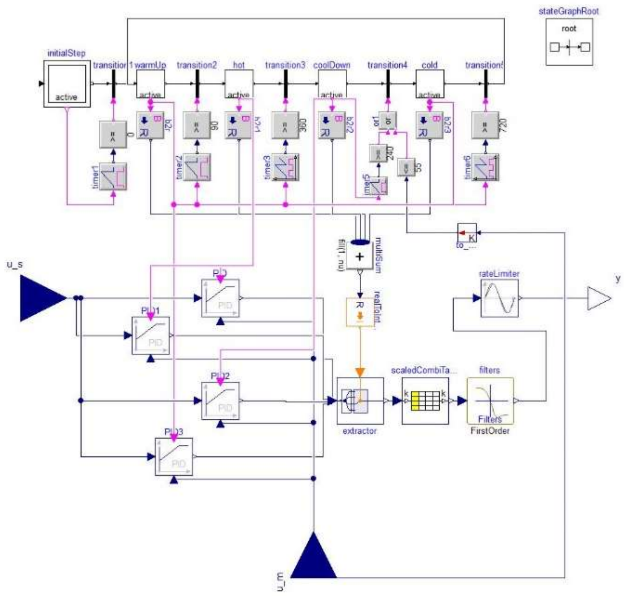


Figure 14. StateGraph controller for the internal coolant conditioning unit 3-way valve.

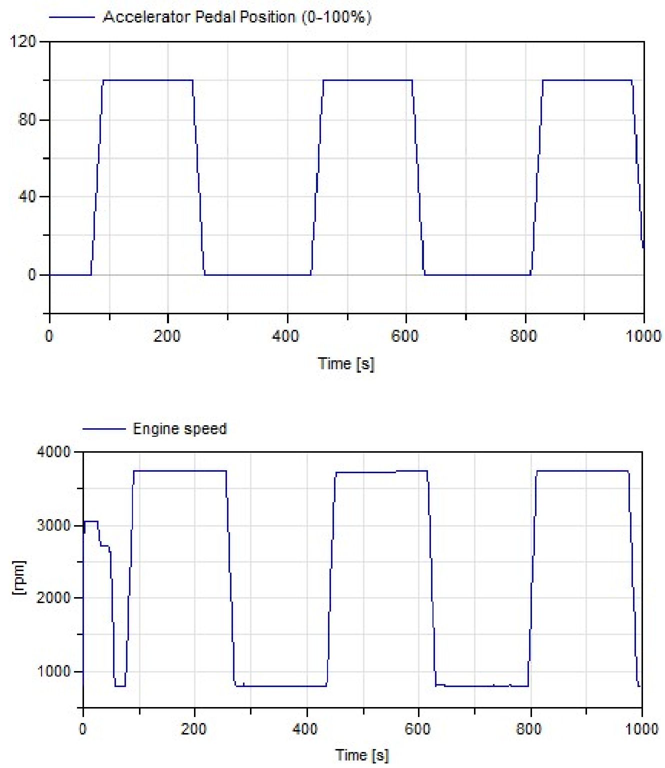


Figure 15. Resulting accelerator pedal position (top) and engine speed (bottom) for the thermal shock test.

DOI 10.3384/ecp17132189

6 Results

6.1 Initial Results from Dymola

Before the actual rig commissioning tests were undertaken, the Modelica system model was already finished, and showing the following expected results:

Thermalshock 2:

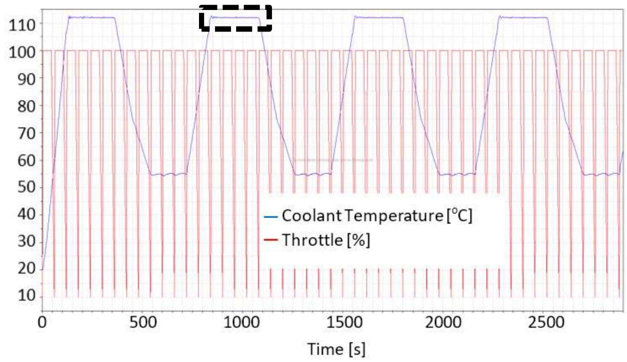
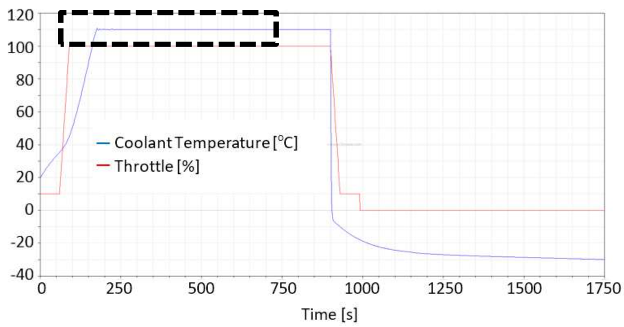


Figure 16. Dymola initial Thermalshock 2 stability estimated as ±0.6ºC with only low frequency and amplitude oscillations.

Thermalshock 1:



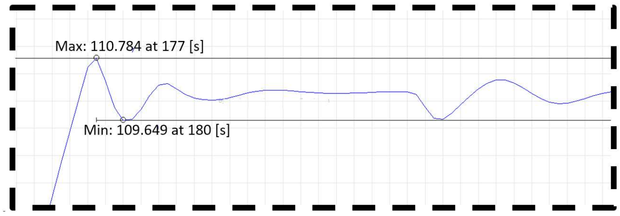


Figure 17. Dymola initial Thermalshock 1 stability estimated as ±0.5º C with only low frequency oscillations on the hot phase an asymptotic cooling up to -30ºC on the cold phase.

6.2 Experimental Results

After the commissioning of the actual real life system, the following data was gathered:

Thermalshock 2 experimental results:

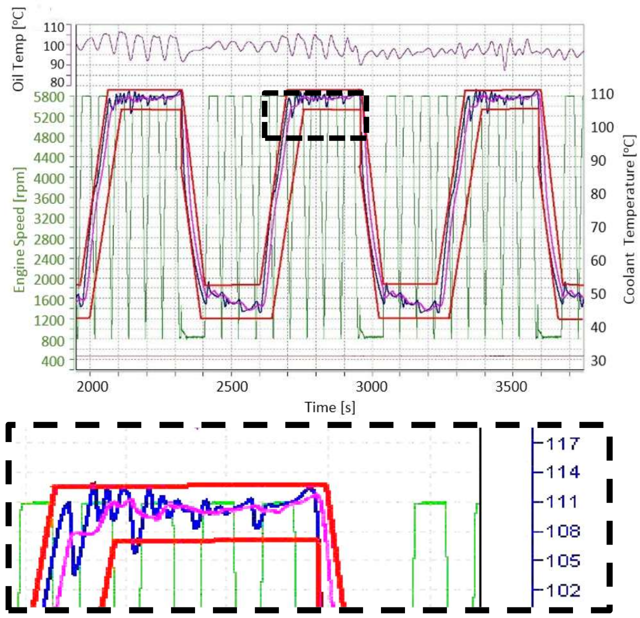


Figure 18. Real Thermalshock 2 results, with higher frequency oscillations and a maximum amplitude of ±3ºC.

At first sight it was seen that the expected accuracies and oscillation frequencies were underestimated.

6.3 Model Adjusting

Since the same system model was used for both tests, the model validation strategy was as follows:

For the thermalshock 2 tests, the real PID control parameters used for the test were recorded. Then, these PID parameters were introduced to the Dymola model. Next, the following parameters where adjusted in order to have similar paths on the engine outlet coolant temperature:

- Engine thermal mass and heat transfer coefficient
- 3-way valve actuation speed

The following parameters/model properties did not require adjustment:

- Fluid properties
- Pipe/ducting geometries
- Bend losses
- Heat exchanger pressure drops
- Heat exchanger performance and thermal coefficients
- Pump flow characteristics
- Pump loss characteristics
- Valve losses
- Engine combustion heat release
- Engine combustion heat release
- Engine inertia

The 3 tuned parameters had different effects on the modelled outlet temperature of the system, and modifying them one by one the following results were achieved:

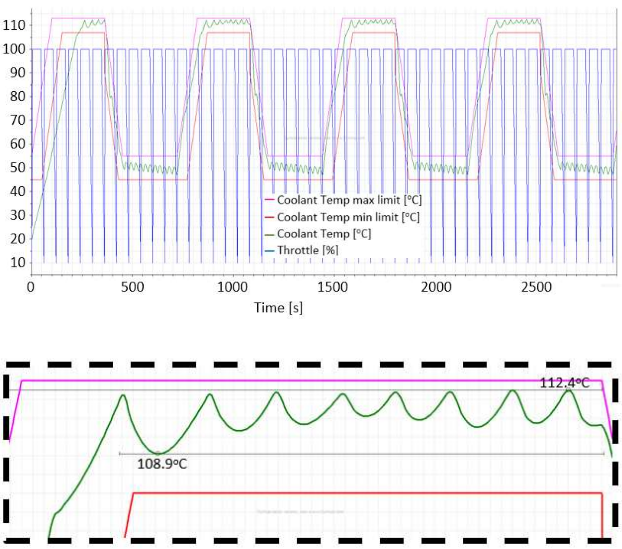


Figure 19. Adjusted Dymola model, running a Thermalshock 2 test, with higher frequency variations and maximum amplitudes of ±1.75ºC on the coolant outlet temperature.

Although the maximum amplitudes measured in the experiment and model results differed (larger in the experiment results: ±3ºC vs. ±1.75ºC, the average temperature and remaining oscillations we of similar value: ±1ºC vs. ±0.8ºC.

6.4 Validation

After checking the model was running accurately simulations on the Hot-Cold tests, all the tuned parameters where frozen, thus obtaining a validated mathematical model.

This model was then used for the Thermalshock 1 test, with the following results:

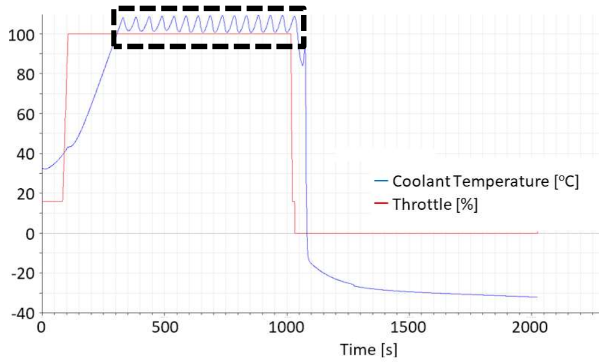


Figure 20. Validated Dymola model, running a Thermalshock 1 test, with higher frequency variations and amplitudes of ±2.7ºC on the coolant outlet temperature on the hot phase, and asymptotic cooling on the cold part.

Where the Thermalshock 1 experimental results were:

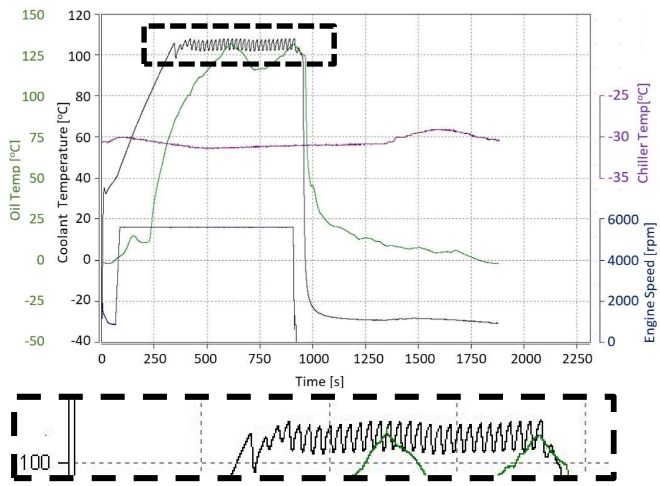


Figure 21. Real Thermalshock 1 results, high frequency oscillations and amplitude of ±2.5ºC.

After model validation and calibration using the hot-cold results, the results for the thermalshock tests were much more realistic, proving that the model can be used for any test made using the same system, regardless of the test conditions.

DOI
10.3384/ecp17132189

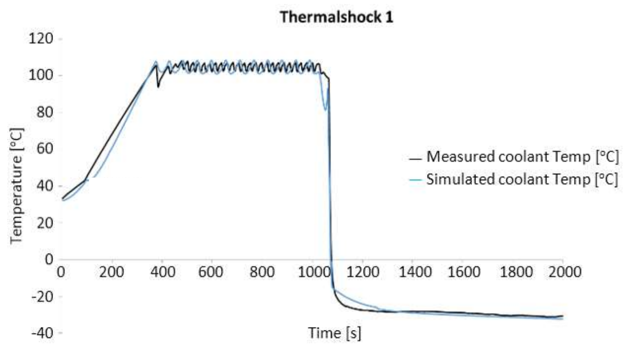


Figure 22. Comparison of the thermalshock 1 real results (black) and the validated Dymola model (blue).

Further investigation is required to confirm the reason for which the higher temperature oscillations in the model are half those measured in the experimental results. Further tuning of the valves pressure drops, hence flow speeds in this operation mode might improve the discrepancy.

6.5 Lubricant Dynamics Validation

Despite the rig model being prepared to also simulate lubricant conditioning and heating (

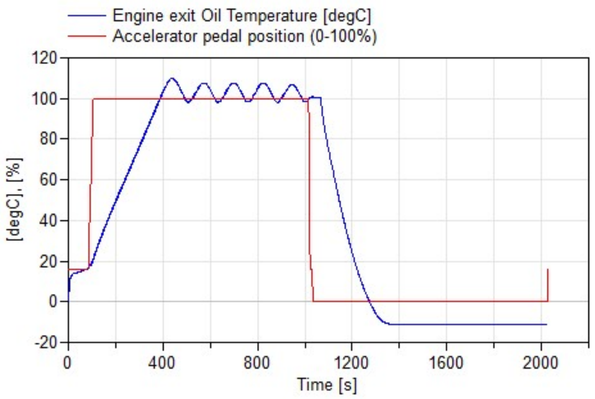


Figure 23), technical issues in the real rig suggested the lubricant measurement data could have been compromised, hence preventing detailed validation of lubricant conditioning (Figure 24).

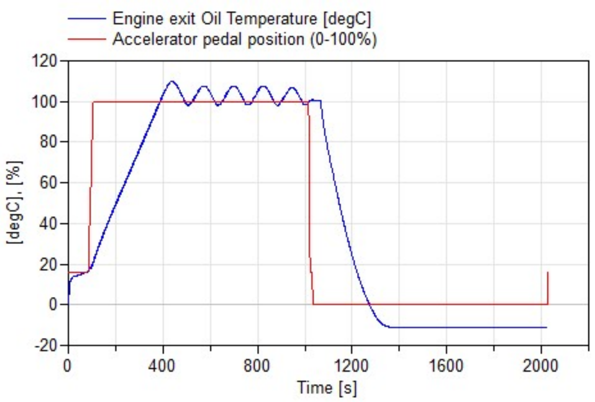


Figure 23. Example of the non-validated results for the lubricant temperature (blue line), running Thermalshock 1 test. Throttle position is also displayed (red line).

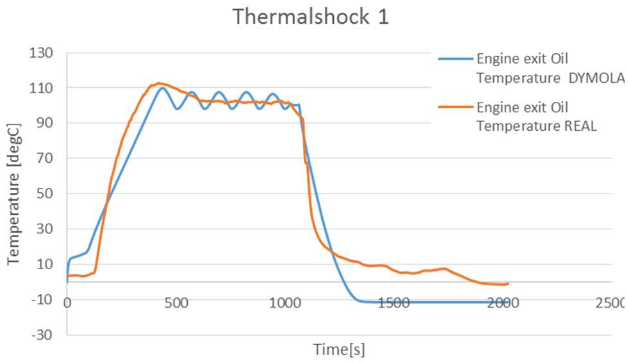


Figure 24. Comparison of the Thermalshock 1 real results (Orange) and the non-validated Dymola model (blue)

This validation will be the objective of a further investigation in the future.

7 Conclusions

After adjusting the Modelica model parameters, the simulation results were much more realistic. Even though the system's hysteresis/entropy is still a parameter that cannot be simulated and produces non-periodic oscillations, the more significant results such as temperature accuracies, heating and cooling times are correctly simulated and can predict an actual behaviour of a system under different test scenarios with an accuracy of ±0.5ºC.

The information about the engine's thermal mass and global heat transfer coefficient will be useful for future projects with similar engines. Even with different engines, these parameters are now a starting point for estimating these values otherwise impossible to know.

References

Casella. F. et al. (2006) The Modelica Fluid and Media library for modeling of incompressible and compressible thermo-fluid pipe networks Modelica Conference, 2006

Dempsey M., and Picarelli A. (2009). Investigating the multibody dynamics of the complete powertrain system. Como, Italy: Proceedings 7th Modelica Conference.

Dempsey M., Picarelli A, Fish G. (2012). Using Modelica models for driver-in-the-loop simulators. Munich, Germany: Proceedings 9th Modelica Conference.

Dempsey M., Roberts N., Picarelli A. (2013) Detailed Powertrain Dynamics Modelling in Dymola – Modelica. IFAC-AAC conference Tokyo, Japan.

Picarelli A., Galindo E., Diaz G. (2014) Thermal shock testing for Engines in Dymola. Lund, Sweden. 10th Modelica Conference, 2014