Designing and analysing the cooling of a medium speed engine piston using MPS method

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Summary

In this paper the method of calculating temperature field for medium speed engine piston is presented. The method combines simple analytical formulas for calculating heat load of the piston, MPS-method of Particleworks software for simulating the cooling effect of oil and finite element analysis for calculating the piston temperature field. This calculation model makes it possible to quickly evaluate several different piston cooling designs in early stage of piston design process. Validation results and comparison of simulation times between MPS-method and VOF-method are shown in the paper.

Keywords

MPS-method, Particleworks, thermal analysis, mesh-free

Introduction

To keep up with the competition in internal combustion engine manufacturing all the key components like piston must be designed and optimized very precisely. During last 50–60 years the brake mean effective pressure (BMEP) of Wärtsilä engines has increased from about 5 bar level of naturally aspirated engines to over 30 bar level of recent W31 engine. Higher BMEP i.e. more work per cycle means also more mechanical and thermal load to components taking part in the combustion process.

High thermal load generates a need for effective piston cooling. Correct piston temperature is needed to maintain steel material properties, prevent hot corrosion and build-up of carbon deposits. Too high temperature leads to excessive thermal expansion and seizure between piston and cylinder liner. Figure 1 demonstrates the principle of piston cooling by oil jet which is one method used for controlling piston temperature. Oil is sprayed from underneath through an inlet hole in piston. Oil splashes inside the cooling gallery of piston cooling it by so called shaker effect. Hot oil then exits piston and flows back to oil sump.

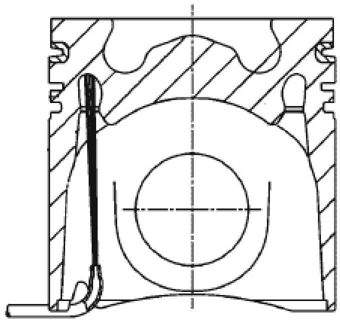

Figure 1: Piston jet cooling principle [1]

When new piston is developed the main focus of CAE work is usually how to calculate mechanical stresses and dimension the piston against fatigue. Nowadays mechanical stresses can be calculated very accurately with sophisticated simulation tools. An aspect that is often left to less attention than solid mechanics is the thermal analysis of piston. Heat transfer related boundary conditions that are being used might be very inaccurate and that way also the temperature field of the piston will be far from real. Ability to simulate realistic piston temperature field has big importance also as starting point of stress calculation. Uneven temperature distribution causes thermal stresses. High temperature has an effect on material mechanical properties as mentioned earlier. These things directly affect outcome of structural analysis. Reason for this inaccuracy of thermal boundary conditions is that they are quite difficult to be determined.

Target of the work was to create a reliable and flexible tool for evaluating many designs which are different when it comes to piston geometry and how the cooling is arranged. It is required that thermal analysis can be done already in early stage of design process so that one can be assured temperatures don’t exceed general limits. Philosophy is not to use time consuming detailed CFD combustion analysis to keep the process quick. Instead, simpler experimental formulas are chosen to calculate thermal load. Most difficult part of the problem is how to calculate the cooling effect of oil splashing inside piston cooling gallery. Convective heat transfer is always coupled to fluid flow and now flow scenario is two-phase free surface flow which is extremely difficult to handle. As said, to keep simulation time short there is no point using traditional CFD methods here neither. Mesh-free MPS method of Particleworks offers a great solution to simulation of fluid flow and determining the cooling effect.

When boundary conditions are found the temperature field is easily solved.

Heat transfer in piston

General transient heat transfer problem can be formulated as below [2]

$$\begin{cases} -\int_{\Omega} (\nabla\psi)^T \underline{k} \nabla T \, dV + \int_{\partial\Omega_q} \psi q'' dS + \int_{\Omega} \psi\left(\dot{E}_g - \rho\dot{u}\right) dV = 0 \\ T = T_0 \text{ on bounday } \partial\Omega_T. \end{cases}$$

This equation forms the basis of solving temperature field with FEM.

Convective heat flux boundary condition of the second term in equation above is expressed [3]

$$q'' = h(T_1 - T_2),$$

where *h* is convective heat transfer coefficient.

Thermal boundary conditions

Figure 2 summarises the combined heat transfer problem of piston cooling.

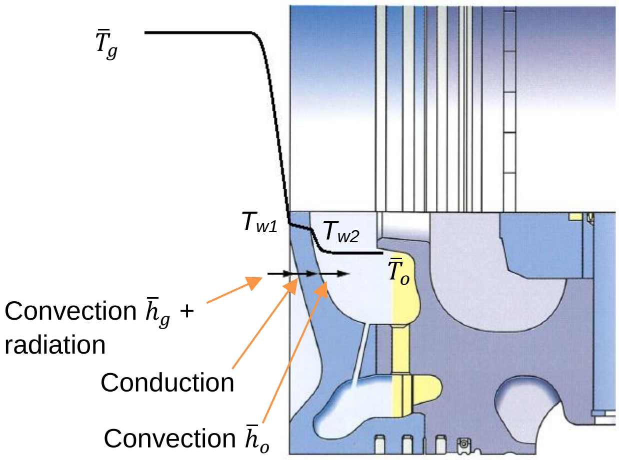

Figure 2: Combined heat transfer in piston

Heat is transferred by convection and radiation from hot gases inside cylinder (temperature $\bar{T}_g$) to piston wall which is at temperature T_{w1} by heat transfer coefficient $\bar{h}_g$. Temperature $\bar{T}_g$ and coefficient $\bar{h}_g$ are assumed to be constant over the whole top surface of piston crown. This is reasonably good approximation for gas engines where heat load is more uniform due to premixed combustion. In diesel engines spatial distribution is needed to take into account injection pattern. From top surface heat is conducted through metal to cooling oil side wall that is at temperature $T_{w2.}$ From there heat is transferred to oil at temperature $\bar{T}_o$ with convective heat transfer coefficient $\bar{h}_o$.

In this application empirical correlation for thermal load chosen is so called Woschni [4] formula. It relates cylinder instantaneous heat transfer coefficient to piston movement, cylinder gas temperature and pressure. The formula can be written

$$h_g = \frac{k_f}{D_c} \mathrm{Re}^b = K_1 D_c^{b-1} p_c^b w^b T_g^{K_2 - 1{,}62b}.$$

Information about valve timing and compression ratio are also included in Woschni formula. Gas temperature inside cylinder is calculated using ideal gas law and cylinder pressure is measured using cylinder pressure sensor, which is anyhow available in Wärtsilä gas engines.

Instantaneous heat transfer coefficient and reference temperature are time-averaged over one full engine cycle using following formulas.

$$\begin{cases} \bar{h}_g = \dfrac{1}{\Delta\phi} \displaystyle\int_{-360°}^{360°} h_g \, d\phi \\ \bar{T}_g = \dfrac{1}{\Delta\phi \, h_{mg}} \displaystyle\int_{-360°}^{360°} h_g \, T_g \, d\phi \end{cases}$$

Particleworks 6.1.1 has a model implemented for calculation of heat transfer coefficient. The model is based on analytical results of flat isothermal plate in parallel flow. Software is used to solve cooling effect of oil i.e. spatial distribution of heat transfer coefficient $\bar{h}_o$ on cooling gallery surfaces. Unlike heat transfer coefficient reference temperature $\bar{T}_o$ is taken as a one constant value.

Solution procedure

General principle of iterative solution procedure and coupling of Particleworks with FEM is shown in figure 3.

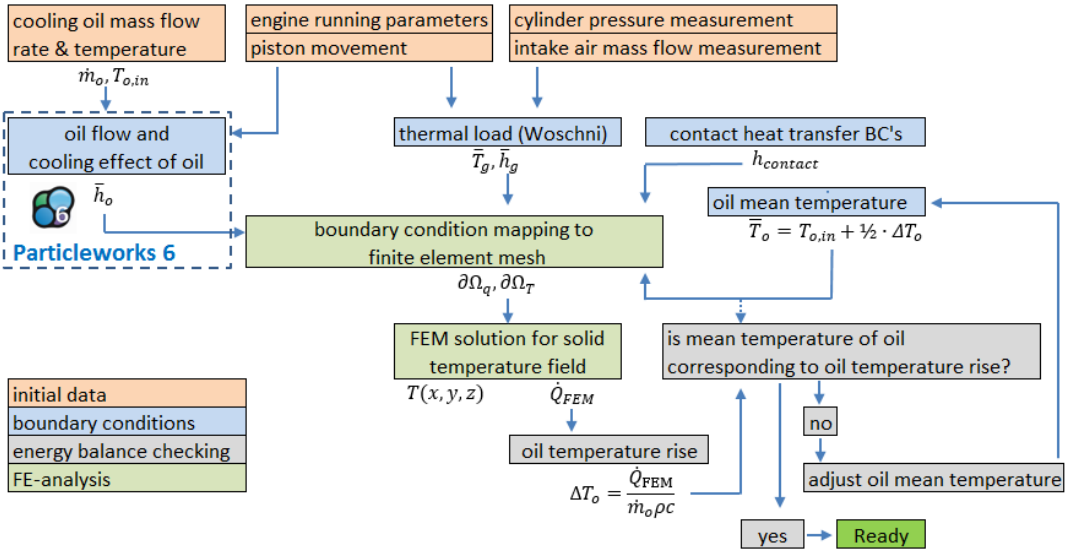

Figure 3: General solution procedure

First, basic geometry and parameters of the engine being analysed must be determined. For example cooling oil mass flow $\dot{m}_o$, and temperature $T_{o,in}$, engine speed, piston geometry, piston movement, cylinder pressure curve and intake air mass flow rate are needed.

Then fluid simulation is performed using Particleworks which gives as a result a time-averaged spatial distribution of heat transfer coefficient inside cooling gallery. The reference temperature of oil $\bar{T}_o$ is guessed at the first round of iteration.

At same time the in-house developed script based on Woschni correlation outputs thermal load, $\bar{T}_g$ and $\bar{h}_g$, for the piston. Gap conductance $h_{contact}$ between piston crown and piston skirt is acquired using simple methods like described in [5] and [6] for example.

When all boundary conditions are determined they are mapped to surface of finite element mesh that describes the solid geometry of the piston. Figure 4 shows that thermal load is applied to red area and cooling effect is mapped to blue areas. Grey area is treated as insulated surface because it's assumed that the major part – at least 90 % – of heat flow goes through red and blue surfaces. Of course there is heat flow through some grey surfaces like top land area but in order to keep the model simple only major contributors are taken into account.

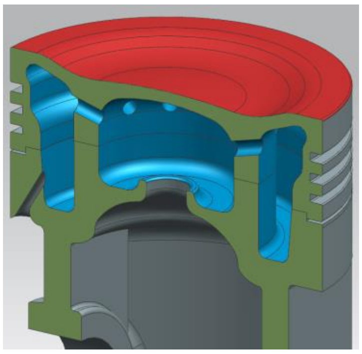

Figure 4: Boundary conditions on different surfaces

After mapping the boundary conditions a steady state FE analysis is performed. Outputs of solution are solid temperature field of piston and heat flow $\dot{Q}_{FEM}$ through the piston. Oil temperature rise equivalent to heat flow $\dot{Q}_{FEM}$ is calculated. A new oil mean temperature based on this oil temperature rise is then compared to initial guess used on the first round of iteration. If two values are different mean oil temperature is adjusted and FE analysis is performed again. This loop will continue until these two temperatures match. Ultimate result after final loop is the piston solid temperature $T(x,y,z)$.

Validation results

To validate the model a simple test series was run on a laboratory engine. Engine is a medium bore Wärtsilä spark ignited gas (SG) engine with mean piston speed 10,75 m/s and specific output of 18,5 kW/litre. One cylinder of the engine is equipped with special piston with thermocouple sensors. Schematic picture on the left hand side of figure 5 shows the locations of temperature sensors. Picture on the right hand side shows oil inlet hole and valve locations. All sensors are located in section A–A.

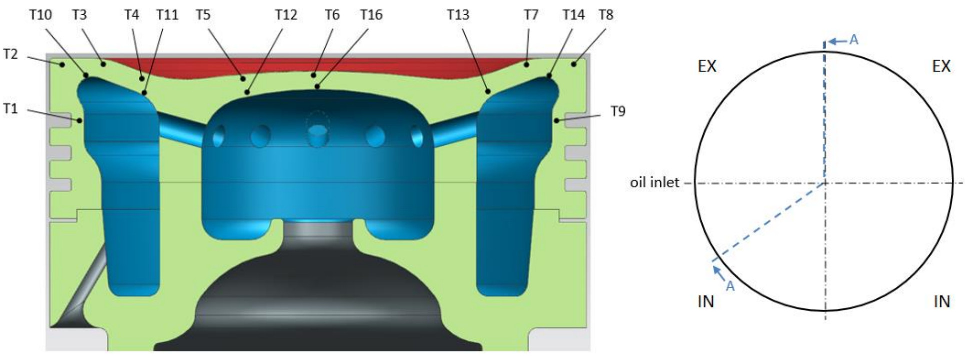

Figure 5: Temperature sensor locations inside piston

Test series according to table 1 was run to see if calculation model is able to handle changes in engine load, cooling oil feeding rate and engine speed.

Engine load (BMEP)	cooling oil mass flow rate		
	1,5 x standard	standard	0,6 x standard
100 %	nominal speed	nominal speed	nominal speed
75 %	nominal speed	nominal speed	nominal speed
50 %	nominal speed	nominal speed	nominal speed
10 %	nominal speed	nominal speed	nominal speed
100 %	-	-	0,95 x nominal
100 %	-	-	0,90 x nominal
100 %	-	-	0,85 x nominal

Table 1: Test matrix

Figure 6 shows an example result set from one point of test matrix. Test this test point was run with 100 % engine load, standard oil feed rate and nominal engine speed. Simulated temperatures are matching well to the measured ones. Root mean square (RMS) error of 15 measurement points is 21,36 °C that can be considered good result.

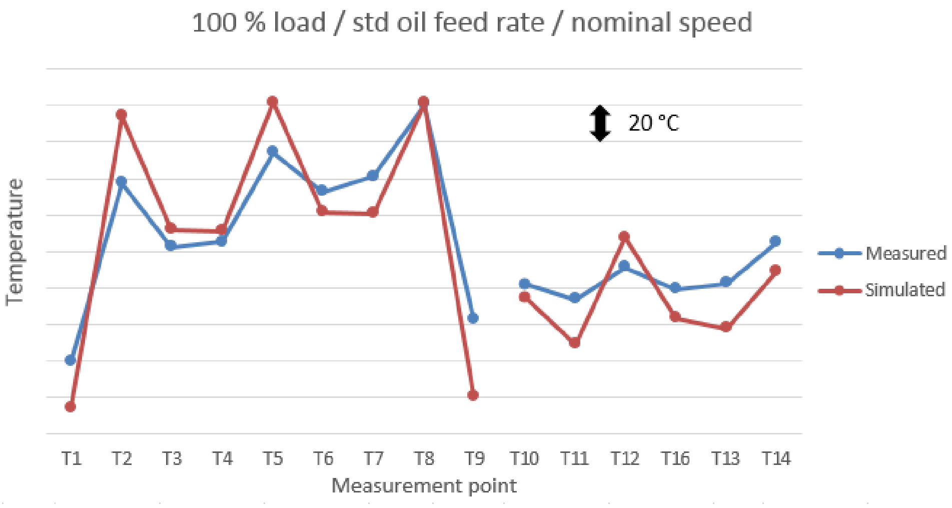

Figure 6: Example result set from validation measurements.

Calculation model underestimates piston ring groove temperatures (T1 and T9) probably because no heat load is applied to top land and 1st ring groove area. Simulation gives rather much hotter temperature T2 than measurement. This is caused by uniform heat load in simulation which doesn't take into account the fact that inlet valve side of the piston is usually cooler than exhaust valve side. Future task is to develop the heat load distribution to get effect of valves included. Temperatures T10–T16 that are located closer to cooling gallery surfaces are generally hotter in measurement than in simulation. Sensor T15 wasn't working properly during test run.

Figure 7 shows correlation between measured and simulated temperatures. One dot represents one temperature measurement point. Different colours represent different engine loads but they include all three values for cooling oil feed rate. On the left hand side engine load is varied and on the right hand side engine speed is varied. Solid blue lines is ideal correlation. Dashed blue line is trendline fitted to all data points and R^2 value is calculated for this trendline.

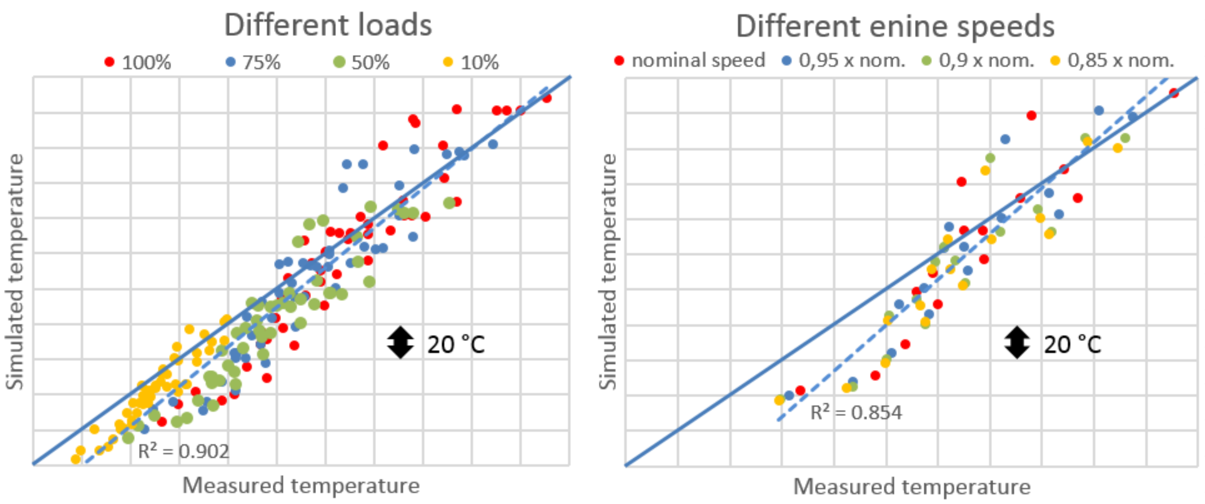

Figure 7: Correlation between simulation and measurement.

It can be noted that model is able to follow the change in engine load and the change in oil feed rate. Especially the highest temperatures that are the most important are very near ideal blue line. On the other hand prediction of temperatures with varying engine speed is not as accurate. Overall the model is giving good results. RMS error in all cases is in range 7…24 °C.

Conclusions

CFD software Particleworks that utilizes MPS method is an effective tool for free surface flow simulation. The basis of heat transfer coefficient model implemented to the software might seem very simple at first but turns out that it calculates the cooling effect of splashing fluid very accurately. Validation measurements showed that model is able to predict effect of main parameters that are related to piston cooling.

Thanks to mesh-free nature of the method pre-processing time of simulation is minimal. The whole pre-processing workflow starting from piston 3D-geometry import from CAD and then setting up a simulation with Particleworks' QUI can be done in only couple of hours. With traditional mesh based methods at least few days is needed to do the pre-processing. The actual simulation time is short too. Post-processing is done with programs own features and self-made excel macros.

Table 2 compares simulation time between Particleworks and another commercial CFD software that uses Volume of Fluid –method (VOF). Basically similar case is simulated using both software.

	MPS-method Particleworks	**VOF-method**
physical time (one piston stroke)	0,1 s	0,1 s
simulation time	5 h	24 h
number of CPU cores	12	160
particle diameter (MPS) average element size (VOF)	1,75 mm	2 mm

Table 2: Comparison of simulation time

With Particleworks the simulation time is 80% shorter even though only a fraction of CPU cores is used. Additionally, VOF method seemed quite unstable at the beginning so extra iterations were needed to get converged results. Even then the method couldn't beat the accuracy of MPS method. Time to perform the analysis with the new model developed is quick enough so evaluation of different piston cooling designs can be done parallel to rest of the piston design process.

Method has already been used to solve cooling problems that were faced with one new piston design project. Piston cooling gallery geometry was unfavourable and oil wasn't properly flowing through the piston causing piston to heat up more than expected which ultimately lead to a piston seizure. Existing design was simulated with Particleworks and problem was discovered. After a new simulation and geometry modification the problem disappeared.

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