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THE AIRCRAFT PISTON ENGINE CONJUGATE HEAT TRANSFER MODEL

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Abstract

Maintaining high aircraft's propulsion system reliability requires a good knowledge of engine's heat transfer conditions at each engine running time. Even though the flow around the cylinder may be steady, the heat flux from the engine is not evenly distributed. This is caused by varied engine head and fins geometry and uneven heat transfer coefficient distribution. The lack of knowledge of the local heat transfer coefficient values and time coefficients for the transient heat transfer make it unfeasible to make an analytical model for a given geometry. One transient Computational Fluid Dynamics simulation does not solve the heat transfer fully. Only a conjugate simulation allows an in-depth analysis of a transient heat transfer. The Combustion and species transport fluid simulation is coupled to the temperature field solid simulation. This work presents the methods and results of such conjugate heat transfer simulation. The change of heat flux parameters in respect to time is shown. The results are verified by the real engine measurements.

Keywords: *CFD, fluid, dynamics, conjugate, coupled, heat, transfer, engine, SI*

1. Introduction

Internal combustion engines require adequate cooling. The temperature in combustion chamber can reach the order 2500 K. The cylinder wall temperature is usually kept below 200ºC in order to prevent oil deterioration. Too high temperature is not good for oil and can cause physical and chemical changes in the oil resulting in excessive wear and sticking of the piston rings, scoring if the cylinder walls or seizure of the piston. For that reason, cylinder walls must be cooled. In the process of converting thermal energy to mechanical energy, high temperature is produced in the gas as a result of combustion process. A large amount of heat is transferred from the gases to the cylinder head and walls, and piston and valves. Their temperature increases as the heat is absorbed. The temperature distribution is not even causing uneven expansion of various engine parts. That results in thermal stresses that cause fatigue and cracking. For that reason, the temperature of iron components must be kept below 400ºC and for aluminium components below 300ºC. Cylinder head temperature higher than 220ºC can lead to overheated spark-plug electrodes or exhaust valves, causing pre-ignition. Spark plug and valves must be cool to prevent knock combustion. Cooling the engine too much is not beneficial either. Starting a cold engine is difficult because of poor fuel vaporisation and lower gas temperature. The mixture is less homogenous and the combustion is poor. Lower temperature means lower average gas pressure and lower work per cycle. This means lower specific power and efficiency. At low temperature, friction is much higher because oil has higher viscosity; therefore, cylinder liner wear is increased. The sulphurous and sulphuric acids formed after combustion process can condense at lower temperatures and corrode the cylinder surfaces. Also the higher the engine temperature the higher the thermal efficiency of a cycle. Therefore, it is desirable to keep the engine as hot as it is possible [1]. Advanced engine simulations that contain many complex models require adequate preparation and calculation work planning. It is not practical to use one big computational model for all phenomena. It is a good practice to split the model into separate simulations, which can be calculated independently. In this case, the heat transfer model is split into three models: One external heat transfer conditions model and two coupled heat transfer and combustion models.

Theoretical heat transfer

The heat flux through combustion chamber walls varies between and as high as 10 MW/m^2 in less than 10 ms. The heat flux varies dramatically with position. The flux pattern varies considerably from cycle-to-cycle. There are various empirical models for instantaneous heat flux calculations. Eichberg proposed a formula valid for two stroke and four stroke diesel engines. It has been found that the formula works well for low piston speeds. As the studied engine is not a typical SI engine, the formula was investigated for comparison.

$$h = 7.67 \times 10^{-3} (C_m)^{1/3} (\rho T)^{1/2} . \tag{1}$$

Woshni proposed another widely used correlation formula based on the similarity law of steady turbulent heat transfer. He found that experimental surface temperature method was not suitable because of data scatter. Instead, he used heat balance to determine the total heat transferred to the combustion chamber walls for each complete engine cycle.

$$h = 0.820\, D^{-0.2} p^{0.8}\, W^{0.8}\, T^{-0.53} , \tag{2}$$

where the reference velocity is given by

$$W = C_1 \cdot C_m + C_2 \frac{V_s\, T_1}{P_1 V_1} (p - p_0) , \tag{3}$$

where C_m is the, mead piston speed and coefficients vary depending on work cycle: for the gas exchange process: $C_1 = 6.18$, $C_2 = 0$; for the compression process: $C_1 = 2.28$, $C_2 = 0$; for the combustion and expansion process: $C_1 = 2.28$, $C_2 = 3.24\text{x}10^{-3}$; p_0 is the motoring pressure (in MPa) and V_s is the displacement volume. It must be noted that the formula treats all convection and radiation in a lumped form. Hohenberg modified the Woshni equation to give a better prediction of time averaged heat fluxes. Sitkei also modified the equation based on avaible data changing the coefficient to a form:

$$h = 2.36 \times 10^{-4} (1 + b) \frac{p^{0.7}\, c_m^{0.7}}{T^{0.2}\, d_e^{0.3}} , \tag{4}$$

where d_e is an equivalent diameter:

$$d_e = \frac{4V}{A} , \tag{5}$$

where V is volume, A is heat-absorbing area and the pressure p has units of MPa. The constant b incorporates the additional turbulent velocity and ranges from 0 to 0.35 [2-6].

2. Model setup and validation

External flow conditions model

In order to find out the heat transfer coefficient for the external engine walls a steady state computational fluid dynamics model was made with an aid of ANSYS Fluent software. The engine model included the surrounding geometry and the in-flight conditions of the flow. The heat flux through the walls was set constant so the heat transfer coefficient values could be accurately noted. With constant heat transfer rate there is always some temperature difference so there is never a case of $h = 0$ or close to 0. This can happen when a constant wall temperature is set according to the fundamental convection equation:

$$q = h(t - t_\infty)\,, \tag{6}$$

where q means the heat flux density, t is the temperature and h is the heat transfer coefficient. This simulation provides the heat transfer coefficient map for external engine surfaces. It is then used as a boundary condition in further simulations.

Coupled heat transfer model

Coupled heat transfer model consists of combustion transient simulation coupled with heat conduction transient simulation. Combustion simulation contains the combustion model and species transport model and is defined by the moving mesh that incorporates piston and valve motion. Heat conduction simulation intends to calculate the temperature map and heat loads of the engine. Both simulations are coupled together and synchronised at the synchronisation points. The heat flux is calculated in both models basing on equation (6). Fig. 1 shows the flow of coupling parameters. The thermal results can be obtained in the paper [7]. Multi material properties are implemented in the model by dividing the computational domain into to sub-domains, coupled by conformal interfaces. For thermal boundary layer, hybrid wall treatment was used as it is proven to provide good results. Both models and coupling were made with an aid of AVL Fire software [8].

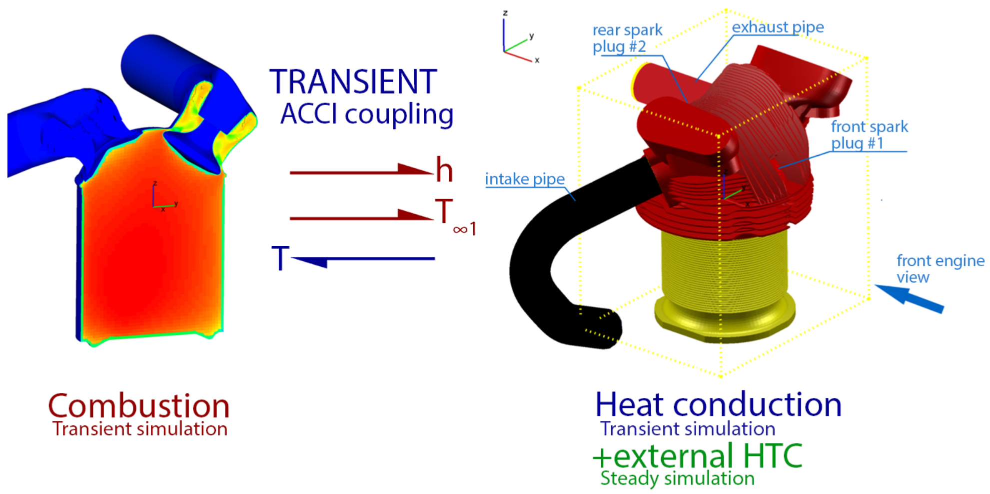

Fig. 1. Coupling between the combustion and heat conduction simulations

Model validation

The CFD#3 simulation is intended to calculate chemical and thermodynamic fluid properties over multiple engine running cycles. The model is developed based on known measured engine parameters, mainly from the engine test stand measurements. The main parameter that describes the operation processes in combustion engines is pressure measured in a combustion chamber. Fig. 2 presents the comparison between the real engine running pressure measurement and simulation results. The measurement was performed with a fibre optic pressure sensor. Infrared images were taken during engine running under take-off power conditions. If the comparison is evaluated some differences in the cooling conditions of both cases should be considered. The flight condition in the simulation assumes that an aircraft is in motion relative to the air. Therefore, cooling air velocity is not only induced by a propeller but it also results from the free stream positive velocity. The IR images were taken when the engine was placed on a test stand. Therefore, the cooling conditions were slightly worse. This can be judged from the higher temperature on the following images. Fig. 3 presents the thermographic image of the third cylinder (right side of the picture) and simulation results (left side), taken from the rear side of the engine. The unified scale temperature field is presented there. The highest temperature of 370 K can be seen in the region near the exhaust duct.

Figure 4 presents the temperature field analogically to the previous figure, but the image was taken from the engine's front side.

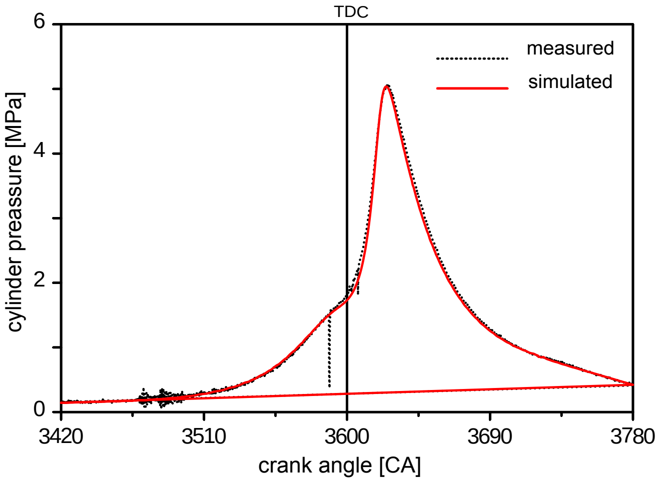

Fig. 2. Cylinder pressure vs. crankshaft angle with measured and simulated in cylinder static pressure [7]

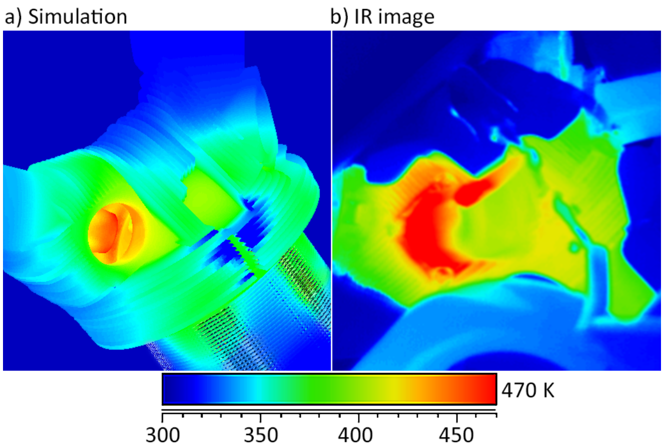

Fig. 3. Temperature distribution on the external engine surface. Infrared image (right side) and simulation result (left side). Engine rear view [7]

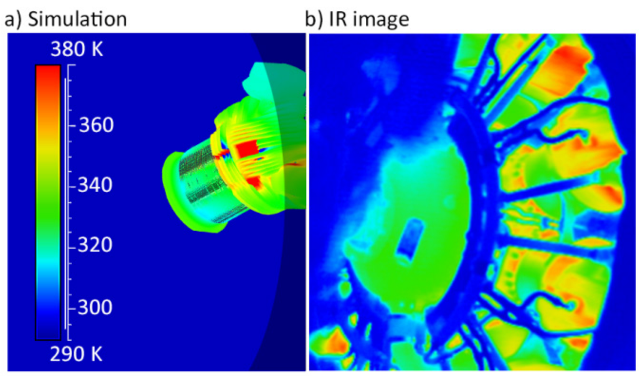

Fig. 4. Temperature distribution on the external engine surface. Infrared image (right side) and simulation result (left side). Engine front view [7]

3. Results

Figure 5 shows the calculated heat transfer coefficient based on Sitkei, Hohenberg and Eichberg empirical formulas. The top dead centre occurs at 360 deg. Those formulas can be used as a good first judgement of the heat transfer in the cylinder but they are not as accurate for different engine types / geometries.

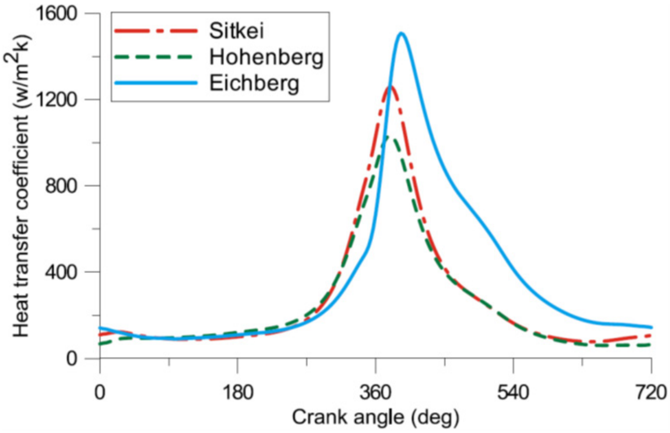

Fig. 5. HTC vs crank angle calculated with empiric Sitkei, Hohenbergand Eichberg formulas

Figure 6 shows the results of the simulation. The heat transfer coefficient for chamber and liner surface is shown. The liner heat transfer coefficient peaks to around 1600 W/(m^2·K) close to the top dead centre and then peaks again to around 2600 during the combustion phase. The empirical functions underestimate the heat transfer compared to the CFD results [9]. After the top dead centre, second peak can be recognised. It occurs during the combustion phase and is a result of increased combustion induced turbulence and gas expansion. For the chamber surface, the heat transfer coefficient is highest during the combustion phase and after exhaust valve is open inducing high flow velocity.

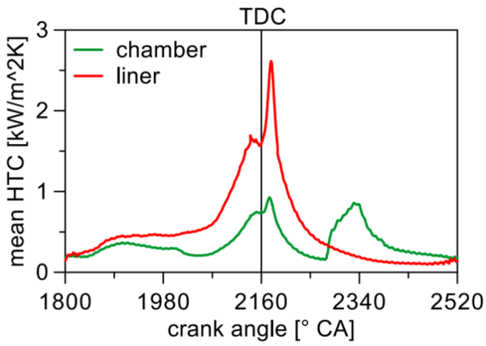

Fig. 6. Simulated heat transfer coefficient on cylinder and liner walls vs. crank angle

The conjugate heat transfer simulation provides an accurate data for further engine running simulations. After the temperature is established, the gathered heat transfer coefficient data can be used as a constant boundary condition taking the reference temperature as an average in-cylinder gas temperature. It seems that the heat transfer coefficient is lower for the chamber surface compared to the liner. Different observation can be noticed about the heat flux density (Fig. 7). The heat flux density is higher for the chamber surface as the gas temperature is high during the combustion phase, around the top dead centre. The cylinder head temperature is higher as well compared to the cylinder. During the compression stroke, the heat exchange to the cylinder walls is minimal as the temperature difference is very small. Therefore, most heat flux density can be observed during the combustion phase. The piston heat load is comparable to the chamber heat load except the chamber has higher heat transfer coefficient during the exhaust phase as the gases pass through the valves.

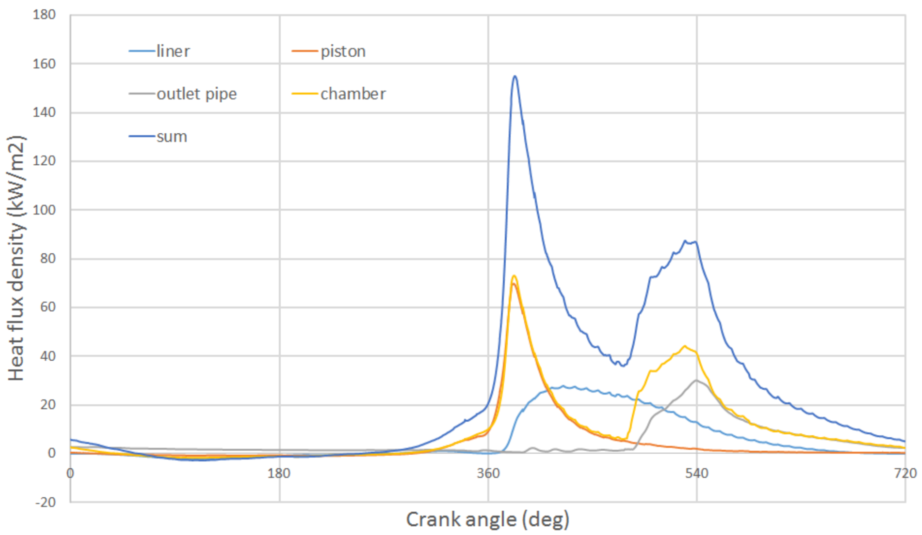

Fig. 7. Simulated heat flux density vs. crank angle position for liner, outlet pipe, piston and chamber surfaces

4. Conclusion

The heat transfer in combustion engine is a very complex phenomenon. For every different surface, the parameters like heat transfer coefficient or the temperature difference varies vastly, changing the heat flux in a hard to predict way. Empirical models like Eichbergs, Hohenberg's, Sitkei's or Woshini's can be used but there can be quiet a large error associated with it. Especially when a non-standard engine is considered. So it is recommended to use those models for first assumptions. For accurate heat transfer and temperature field calculations a conjugate three-dimensional CFD, simulation can be performed as shown in this paper. Model validation proves a good accuracy and provides many results that cannot be obtained with different methods.

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