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THERMODYNAMIC ANALYSIS ON COMPRESSIBLE VISCOUS FLOW AND NUMERICAL MODELING STUDY ON PISTON/CYLINDER INTERFACE IN AXIAL PISTON MACHINE

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Abstract. The fluid film behavior in the lubricating gap between piston and cylinder bore in axial piston machines is the main focus of this paper. The thermal behavior of the compressible viscous flow in the thin lubricating film formed between piston and cylinder bore plays a critical role on the interface performance and energy dissipation, therefore, deserves a thorough analysis. The temperature distribution in the fluid domain, as well as the heat flux from the fluid domain to the solid domain, follow the first, and the second laws of thermodynamics, however, are difficult to solve due to the constantly changing boundary conditions. The proposed fluid domain thermodynamic model calculates temperature distribution in the gap flow and the heat flux to the solid parts with a higher accuracy than the currently used fluid domain heat transfer model.

Keywords: Thermodynamic, Temperature distribution, Energy equation, Heat transfer, Piston/cylinder interface

INTRODUCTION

The temperature distribution in the fluid domain of the main lubricating interfaces of axial piston machine is always a key aspect of the fluid film behavior. Not only the fluid properties but also the heat flux applying on the running surface of the solid parts that form the fluid film have critical influences on the energy dissipation and the leakage occurring in this lubricating interfaces In fact, the fluid temperature distribution is a result of the energy dissipation due to the viscous friction, the convection due to the gap flow, the fluid pressure changing with respect to time and space, the conductivity of the fluid, and the conduction between the fluid domain and the solid domain. A fluid and structure interaction problem is then created by the conduction and thermal deflection on the running surface of the solid domain. The temperature distribution in both the fluid domain and the solid domain interact with each other. And, the thermal deformation of the solid bodies under the thermal load changes the shape of the boundary of the fluid film, impacts the pressure build up in the fluid film, therefore the energy dissipation, the gap flow, and the fluid temperature.

The main objective of the paper is to study the fluid temperature distribution in the lubricating interface between piston and cylinder bore in axial piston machine. In the past, many researchers have tried to address this fluid structure and thermal interaction problem for piston/cylinder interface with more or less simplification of the complex physics. Ivantysynova [1][2] published a modeling approach that firstly solves the pressure distribution in the piston/cylinder interface using Reynolds equation for non-isothermal fluid. The viscosity in the fluid domain changes with the temperature, which is calculated considering conduction, convection, and the energy dissipation due to the viscous shear. Olems [3] solved piston micro motion fulfilling piston body force balance between the external load and the fluid film pressure force considering squeeze motion. His model was developed based on the previously published non-isothermal model proposed in [1][2]. Ivantysynova and Huang [4] added the elastic deformation due to pressure into Olems's model. Their model captures the influence of the pressure deformation of the solid bodies on the fluid film behavior including the pressure distribution in the gap. Therefore, the fluid model and the solid parts deformation are solved using an iterative scheme. The pressure deformation in their model is calculated using an influence matrix approach. Pelosi and Ivantysynova [5] further improved the modeling ability of piston/cylinder interface by adding a solid domain heat transfer model and a thermal elastic deformation model. In their model, the fluid temperature distribution generates heat fluxes on both the piston and the cylinder bore running surface. The heat flux on each body then acts as thermal boundaries on a three-dimensional heat transfer model that solves the solid domain temperature distribution. This thermal load can be calculated from the resulting temperature fields, and be used to calculate the solid body thermal deflection. The deformation of the running surface of the solid body changes the shape of the fluid film.

The temperature on the solid body running surface controls the boundary temperature of the conduction calculation in the fluid domain. This fluid structure and thermal interaction problem is solved in their model using another iterative loop. Shang and Ivantysynova [6] published a port and case flow temperature prediction model that calculates the temperature in displacement chamber, inlet/outlet port volume, and the case volume. Those temperatures are essential thermal boundaries for the three-dimensional solid body heat transfer model. Their model put the last piece in the puzzle, enables the piston/cylinder interface fluid film behavior modeling without any support from measurements.

Shang and Ivantysynova [7] created a temperature adaptive piston design utilizing the previously described model. They proved that changes of the thermal behavior of the solid body including the thermal expansion coefficient and the heat transfer coefficient have critical impacts on the piston/cylinder interface performance. Their proposed bi-material piston showed improvement on energy dissipation with inlet temperatures in the range between -20°C and 100°C.

Currently, the fluid temperature modeling approach of the state-of-the-art piston/cylinder interface does not consider the temperature change due to the fluid pressure change with time, or gap flow through pressure gradient. However, Cheng and Sternlicht [8] and Cheng [9] already included the fluid velocity through pressure gradient in the source term of the energy equation to solve the thermal effect between two rolling and sliding cylinders. In their thermal analysis, the first law of thermodynamics was fulfilled. Burton [10] analyzed the thermodynamics of a viscoelastic film under shear and compression. His study combined the first and the second law of thermodynamic. Xu and Sadeghi [11] reported their thermal EHL analysis of circular contacts with measured surface roughness. A three-dimensional time dependent nondimensional energy equation for the lubricating fluid was proposed which includes the local temperature change with respect to time.

In this article, a more completed numerical fluid film thermodynamic model is firstly proposed for the piston/cylinder interface for axial piston machine. The fluid temperature distribution is solved not only considering the convection, conduction, and energy dissipation, but also considering the temperature change with time, the load pressure changing with time, and the flow through pressure gradient. Both the first and the second law is fulfilled in this proposed thermal model.

In the first part of this paper, the energy equation which is the combination of the first and the second laws of thermodynamics is analyzed focusing on the compressible viscous flow. The concluded physical relationships, was then constructed with the finite volume method, resulting in a thermodynamic model for the given boundary conditions. In the second part of this paper, the proposed thermodynamic model was inserted into a fluid structure and thermal interaction model developed in authors' research group, which allows the prediction of the fluid behavior in the main lubricating interfaces in a swash plate type axial piston machine considering the elastic deformation on the solid parts under both the pressure and thermal load, and the heat transfer in both the solid, and fluid domain. The proposed fluid domain thermodynamic model calculates the fluid domain temperature distribution and the heat flux to the solid domain with a higher accuracy than the currently used fluid domain heat transfer model. In the third part of the paper, the simulated flow thermal behavior are compared between the proposed and the current model. The simulated local temperature in the solid domain is also compared to the measured film temperature from a modified axial piston machine.

TIME DEPENDENT ENERGY EQUATION FOR COMPRESSIBLE VISCOUS FLOW

The foundation of the fluid film thermodynamic model is the energy equation that links the rate of local temperature change to the rate of local energy dissipation, to the rate of the local pressure change, and to the fluid conduction, the fluid convection, the fluid flow through pressure gradient. In order to solve the heat transfer problem numerically, the fluid domain between the piston and the cylinder bore is discretized to a three-dimensional structured numerical domain as shown in figure 1(a). For each three-dimensional finite volume as shown in figure 1(b), there are six faces labeled 't', 'b', 'w', 'e', 'n', and 's' separate the volume from its neighbors located at 'T', 'B', 'W', 'E', 'N', and 'S'. The mass in the control volume is changing due to the mass flow rate through the faces 'w', 'e', 'n', and 's'. The fluid velocity on the gap height direction is assumed to be zero. The temperature and the pressure in the control volume are token from the centroid. The local pressure as well as the pressure in its neighbors are changing due to the changing displacement chamber pressure, the piston sliding and spinning motion, and the piston micro squeezing motion. The pressure is calculated also considering the deformation of the solid bodies, and the fluid properties changing with pressure and temperature. The local temperature is changing due to the conduction on the six faces, the convection with the mass flow rate, the local pressure changing, and the mass flow rate through the pressure gradient. In order to model the previous described problem, an energy equation is used:

$$H = H_0 + \Delta H$$
$$h \cdot m = h_0 \cdot m_0 + (\frac{dh}{dt} \cdot m_0 + h_{in} \cdot \dot{m}_{in} - h_{in} \cdot \dot{m}_{in}$$
$$+ \mu \cdot \Phi_D \cdot dV + \sum_i \lambda_i (\nabla T)_i \cdot A_i) \cdot dt \tag{1}$$

The term on the left hand side of the energy equation expresses the total enthalpy of the control volume at the current time step. The terms on the right hand side are expressing the total enthalpy at the previous time step, the enthalpy changing with the in and out going mass flow, the energy dissipation due to the viscous friction, and the conduction on each face. The specific enthalpy in the proposed energy equation is a function of pressure and temperature:

$$h = h_0(T_0) + \int_{T_0}^{T} c_{p_0} dT + \int_{p_0}^{p} T\left(\frac{\partial v}{\partial T}\right) dp + \int_{p_0}^{p} v dp \tag{2}$$

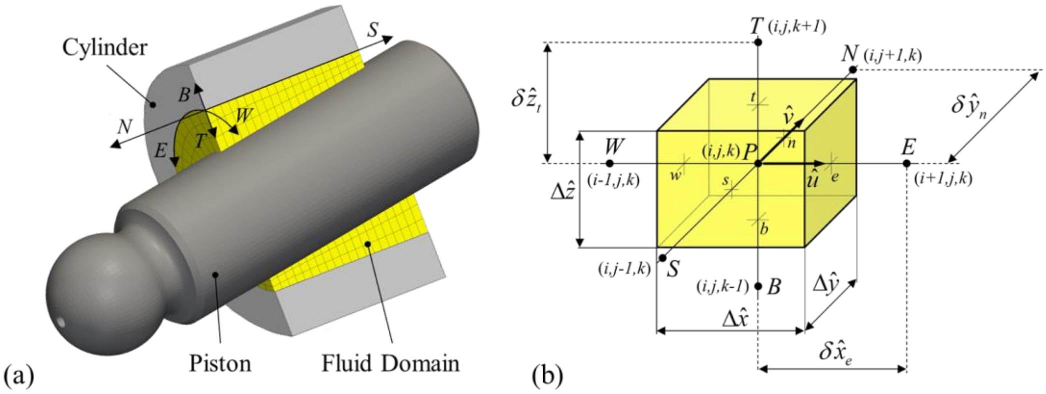

FIGURE 1. The three-dimensional finite volume structure for fluid domain (a), and the finite volume definition (b) Pelosi and Ivantysynova [12]

Combining Eq. 1 and Eq. 2, the local temperature yields to a function of neighbor's temperature and the source term:

$$T_P = f(T_T, T_B, T_N, T_S, T_W, T_E, s) \tag{3}$$

Where the source term is a function of the local previous temperature, current and previous pressure, and the neighbors' pressure:

$$s = f(T_o, p_o, p_P, p_T, p_B, p_N, p_S, p_W, p_E) \tag{4}$$

The source term is pre-defined before the temperature distribution calculation. And the fluid domain heat transfer problem is solved through a Gauss-Seidel over-relaxed method with the solid bodies' surface temperature as boundary conditions.

Comparing to the previous energy equation proposed by Ivantysynova [2], this presented thermodynamic model solves the temperature changing rate instead of absolute temperature. Therefore, an initial condition at time equal to zero is needed, and for the research study presented in this publication, it is set to be the linear temperature distribution between displacement chamber temperature and the case temperature.

FLUID STRUCTURE AND THERMAL INTERACTION MODEL FOR PISTON CYLINDER INTERFACE UTILIZING THE PROPOSED THERMODYNAMIC MODEL

The fluid structure and thermal interaction model developed in authors' research group allows the prediction of the fluid behaviors in the piston/cylinder interface. As shown in figure 2, the fluid pressure and temperature are solved through a finite volume method. The Reynolds equation is discretized in a two-dimensional grid to calculate the pressure distribution in the gap area. The pressure deformation on the solid bodies is then calculated using an off-line influence matrix method. The fluid structure interaction problem can be solved using an iterative method. And the converged pressure and the fluid film thickness are used by the proposed thermodynamic model to calculate the fluid temperature distribution and update the fluid properties. The heat flux on the fluid/solid domain boundary can be calculated from the resulting fluid temperature gradient at each time step. An average heat flux on the running surface of both piston body and cylinder block body for the whole revolution is obtained after each simulated revolution. At the end of each revolution, the solid bodies' temperature is calculated based on the average heat flux, and the thermal deflection is then calculated based on the body temperature. This thermal deformation is used to update the fluid film shape for the next revolution, and the converged heat flux and body temperature can be obtained using an iterative method.

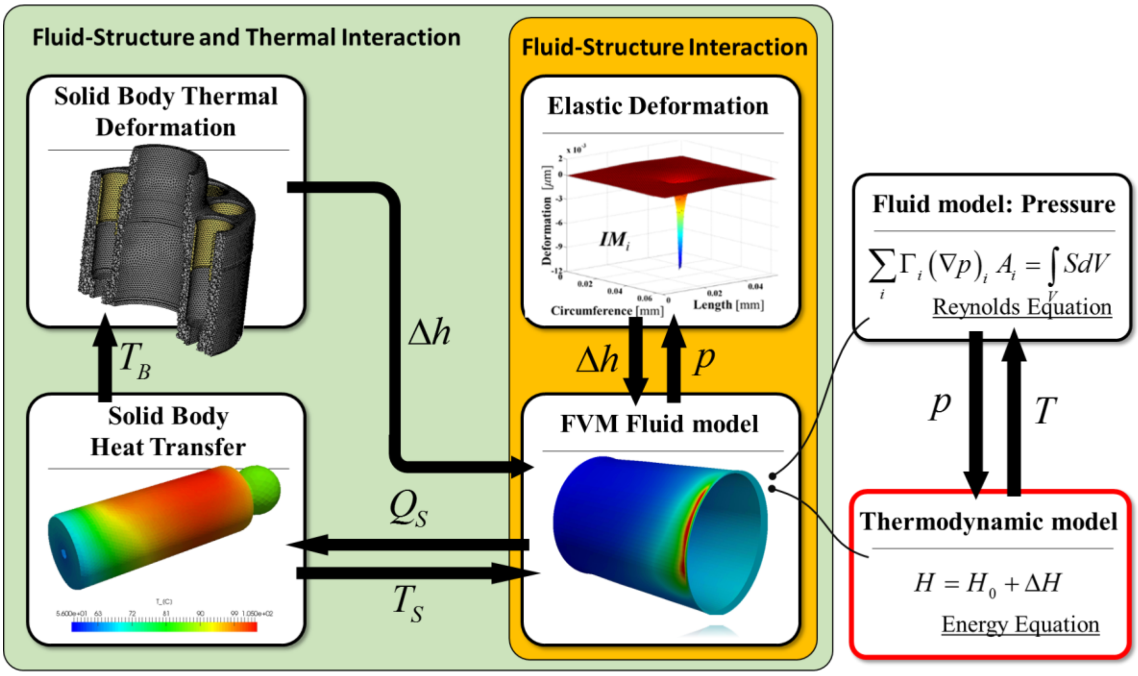

FIGURE 2. Simulation scheme for piston/cylinder interface.

Figure 3 shows a cross section of the resulting pressure and temperature distribution from the above-described model with the proposed thermodynamic model. The piston spins counter-clockwise relative to the cylinder bore and drags the fluid from zone C through zone B into zone A. As shown in the left-hand side of figure 3, the pressure is building due to the wedge shape of the fluid film and the piston spinning motion. The pressure of the fluid flow through zone B increases, as well as the fluid enthalpy. The high energy dissipation in zone A due to low film thickness also increases the fluid enthalpy. The increased enthalpy is represented by the fluid temperature as shown on the right-hand side of figure 3. The high temperature in zone C is due to the convection and conduction from zone B. Due to the low film thickness in zone B and zone A, more energy is transferred to the solid parts.

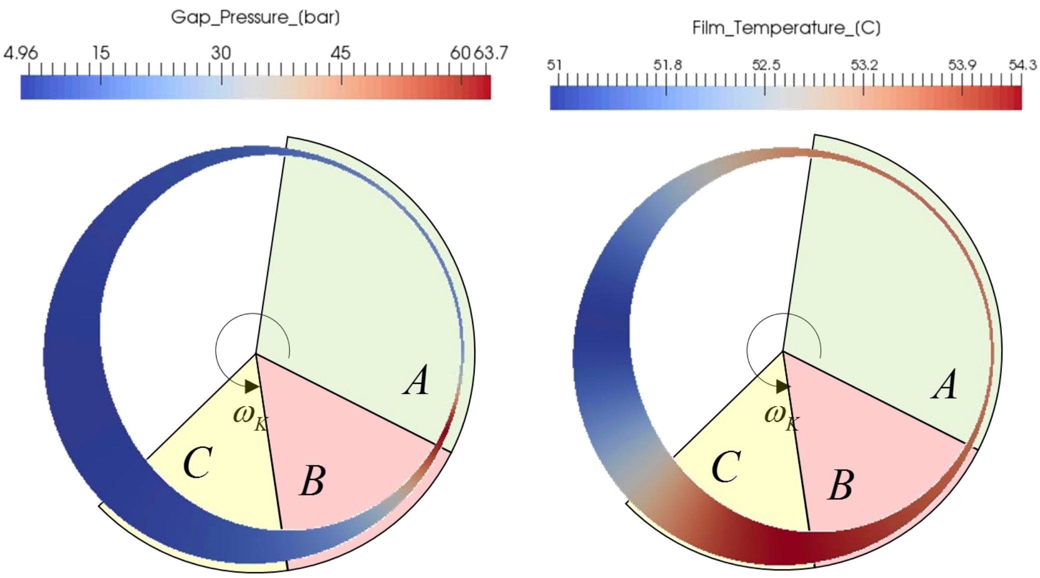

FIGURE 3. Cross section for pressure and temperature distribution in piston/cylinder lubricating interface

Comparing the temperature distribution in the piston/cylinder interface using the previous model that proposed by Pelosi and Ivantysynova (2012), the new thermodynamic model calculates higher fluid temperature thanks to the consideration of the pressure changing with respect to time and space.

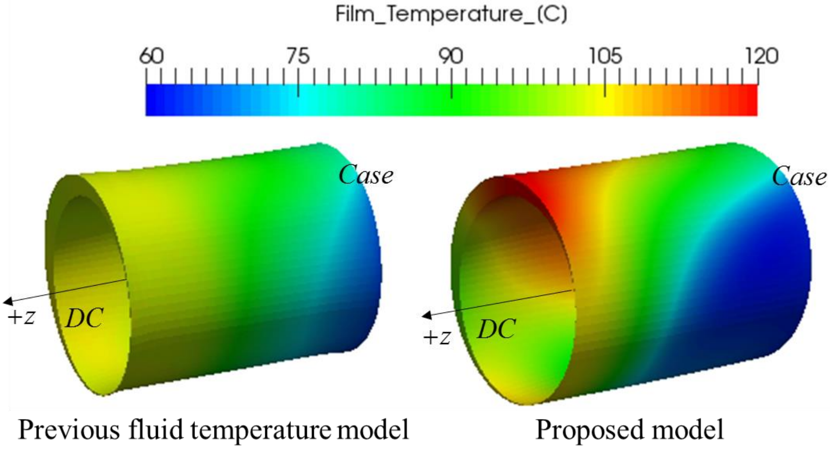

FIGURE 4. Temperature distribution comparison between previous model by Pelosi and Ivantysynova [12] and the proposed new thermodynamic model.

COMPARISON OF SIMULATION RESULTS TO MEASUREMENT USING THE EHD TEST RIG

The EHD test rig has been designed and built by Ivantysynova et al. [12]. The specially designed pump for the EHD test rig, as shown in figure 5, is equipped with nine thermocouples and nine piezoelectric pressure sensors. The EHD pump is designed to achieve the same piston kinematics and dynamics as a commercial axial piston machine. Instead of a rotating block with nine pistons, the EHD pump is designed with a wobbling swash plate and a single piston stationary cylinder block. As shown in figure 5, the shaft rotates the swash plate and drives the piston in and out from the cylinder bore. The thermocouples are placed in the cylinder block pointing to the lubricating gap. The diameter of the tip of the thermocouple is 0.5 mm. The tip of the thermocouples are touching the fluid film, and therefore, forming part of the running surface of the cylinder bore. The nine thermocouples are placed 2.5mm, 3.0mm, 5.0mm, 8.0mm, 14.33mm, 20.66mm, 23.66mm, 25.66mm, and 26.16mm from the displacement chamber end of the running surface of the cylinder. The thermocouple closest to the case end is 2.5mm from the end of the running surface.

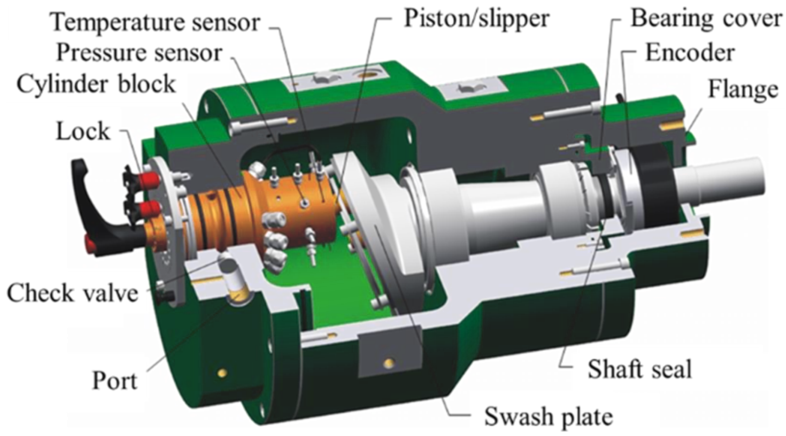

FIGURE 5. Special designed pump for EHD test rig by Ivantysynova et al. [13]

As shown in figure 5, the thermocouples are placed 45° apart. The first and the last thermocouple laid on the same circumferential position but 23.16mm apart axially. A locking device fixes the cylinder block at 180 different angular positions. Therefore, the temperature of the cylinder bore running surface can be measured at 1620 different locations.

The piston that was used for the measurement has a barrel shape as proposed by Ivantysynova and Lasaar [14]. This surface shaping has been considered in the simulation. The operating conditions of the measurement are listed in table 1. The temperature of the case flow and at the high pressure port were measured when the pump reached steady state conditions.

TABLE 1. Operating conditions for EHD test rig.

Operating conditions	
Shaft speed	1000 [rpm]
Differential Pressure	150 [bar]
Case Temperature	55.0 [°C]
Temperature at High Pressure Port	45.0 [°C]
Temperature at Low Pressure Port	43.0 [°C]

When solving the solid body heat transfer problem, two types of boundary conditions are used on both the piston and cylinder of the EHD pump as shown in figure 6. The mixed boundary applies heat flux on the surface of the solid bodies that is calculated from the environment temperature and the convection coefficient. The Neumann boundary applies heat flux that is calculated from the normal gradient of temperature at the surface.

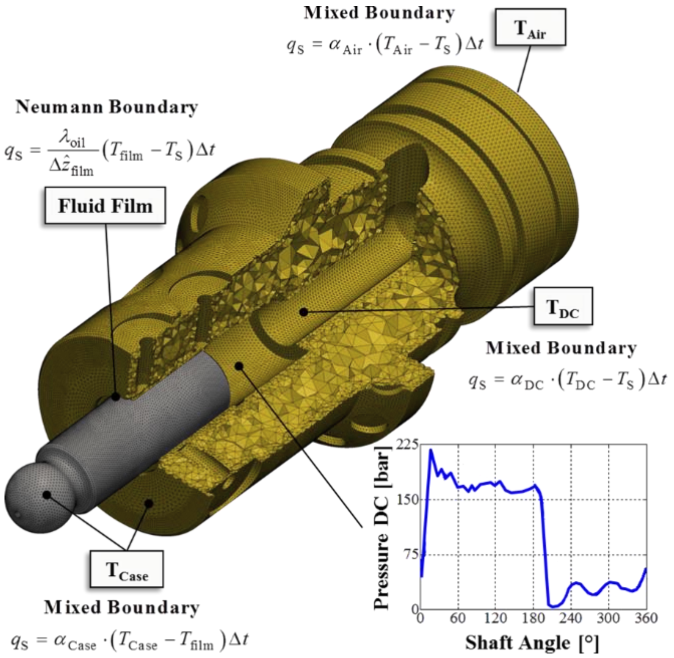

FIGURE 6. Boundary conditions for the solid body heat transfer model, Pelosi and Ivantysynova (2012)

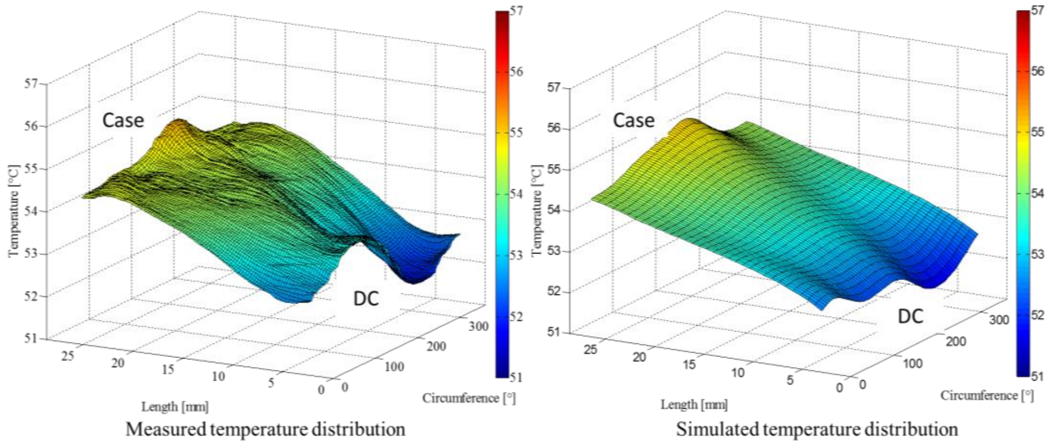

FIGURE 7. Temperature distribution comparison between measurement and simulation at sensor location 1 and 3.

Figure 7 shows the comparison result between the measurement and simulation. The temperature distribution of both the measurement and simulation show similar absolute value and trends. The simulation is able to catch the hot spot near the case end. However, the measured hot spot at displacement chamber (DC) end is not represented in the simulation. Because of the unexpected derivation of the simulated temperatures from the measured profile, the authors very recently re-measured the surface of the cylinder and piston using a surface profilometer. These recent measurements did show that the surface profile of the cylinder bore has changed compared to the profile which was measured before the temperature measurements were carried out, however the original measured surface profile has been used for the simulation. The authors are still rerunning simulations and will present and discuss the new results at the conference.

CONCLUSION

The proposed thermodynamic model for the fluid domain in the piston/cylinder interface added the change of pressure with respect to space and time and rate of temperature change into model. The simulated temperature distribution comparison result between the previous and the proposed model show a significant difference. The previous model underestimated the temperature due to the fact of neglecting compression heat. The simulated temperature distribution using proposed thermodynamic model was compared to the measurement. The comparison result verified that the new model is capable of calculating the temperature distribution in the fluid domain of piston/cylinder interface with a very good accuracy.

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