# Cylinder Block / Valve Plate Interface – a Novel Approach to Predict Thermal Surface Loads

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#### Abstract

In this paper a novel simulation model for the cylinder block / valve plate interface is described. For the first time, precise estimations of thermal and elasto-hydrodynamic effects are fully coupled together in order to have an accurate description of the fluid film thickness as a function of the main operation parameters. A comparison between simulation predictions and actual measurements of the valve plate surface temperature of a 100 cc unit is presented for different operative conditions. Cylinder block, valve plate and end case body temperature and thermal deformations are also shown, explaining their impact on the fluid film and the interface performance.

KEYWORDS: cylinder block / valve plate interface, elasto-hydrodynamic lubrication, heat transfer, thermal expansion

# 1. Introduction

The lubricating interfaces of an axial piston machines represent a key design element. A significant reduction in energy dissipation and an improvement in efficiency and power density can be reached through a deeper understanding of the complex physical phenomena taking place in these interfaces and utilization of the gained fundamental knowledge within a novel computational design.

# 1.1. Cylinder block valve plate interface

This paper focuses on the cylinders block / valve plate interface, Likewise the other interfaces of axial piston machines, the cylinders block / valve plate interface has to fulfil a sealing and a bearing function. However, its design is problematic, representing a trade-off among opposite requirements. In particular, to limit the leakage flow, the interface should provide a very thin lubricating film; on the other hand this would dramatically increase the energy dissipated by viscous friction. Moreover, to minimize possible conditions of mixed lubrication, with a consequent extensive wearing of the solid parts, an adequate film of lubricant should always be present. The lubricating film

is also responsible of the interface load carrying ability. The external loads acting on the cylinder block are obviously function of the operative conditions (pressure, machine speed and swash plate angle), but they also vary in time, due to the machine kinematics and the oscillating pressure in the displacement chambers. All these external forces need to be balanced by the pressure field in the lubricating gap. The pressure field is characterized by a hydrostatic and a hydrodynamic component. The first is determined by the diffusion of pressure from the fluid film boundaries and can be estimated analytically as in *1*?. The second is instead the result of complex phenomena, all coupled together: cylinders block and valve plate relative inclination and micro-motion, elasto-hydrodynamic deformation and thermal expansion of both the boundary surfaces. These effects have been proved to be fundamental for the machine performance, as shown in *1*16, 17/ relatively to the piston cylinder interface. Unfortunately, being much more difficult to calculate, they have never been completely considered in the design process, which instead has mostly relied on the expansive trial and error practice.

#### 1.2. Previous research

The cylinder block / valve plate interface has been subject of many studies in the last 40 years. Experimental investigation of the clearance and its modification as regard to oil properties, operative conditions and geometric details were carried out by /7-10/. Interesting consideration were found, however the measurements apparatus used in those works could not be assumed as capable of measuring directly the film thickness, but rather of capturing the relative position of the block with respect the valve plate. Other important experimental investigation is found in /11/, with focus on the thermal behavior of the rotating kit.

Analytical models were developed by /1-6/ providing important information to better understand the complex behaviour of the interface. Nevertheless these models have clearly shown to be insufficient even to predict a possible trend regard the interface performance. For this reason, the growing need in computational tool to aid the design process, led the researchers to the development of more advanced CFD based models. In these models new phenomena such as micro-motion, elasto-hydrodynamic lubrication, non isothermal flow and heat exchange with boundary solids were introduced.

In this field remarkable works were published by the research group in Achen and by the team of professor Ivantysynova in Germany and later in the USA. In particular the model developed by Wieczorek and Ivantysynova /12/ was the first attempt of

predicting the film thickness based on the balance of the external forces acting on the cylinder block through the calculation of the hydrodynamic pressure field in the gap. Unfortunately this model has soon revealed to be poor, especially relatively to the cylinder block valve plate interface, being still too simplified regarding the main phenomena involved in the lubrication mechanism. Successive developments in the interface modelling moved towards the elasto-hydrodynamic lubrication direction, as in /18-20/ and /13/, since also experimental evidence underlined the importance of these effects (/10/). The model of Huang introduced the elastic deformation of the cylinder block due to pressure loads through a coupling of the pre-existing model /12 / and the commercial code ANSYS; deformation on the valve plate side was not modelled though. Some years later, Jouini and Ivantysynova (/15/) introduced thermal capabilities in the model described in /13/. The temperature of the surfaces bounding the gap were calculated taking into consideration heat fluxes due to energy dissipation, accounting in a simplified way for the heat transfer through solid parts (cylinder block and valve plate) to the pump case and ambiance. Jouini and Ivantysynova compared their simulation results with surface temperature measurements conducted on a 100cc swash plate type axial piston pump. The simulation results captured the main trend; however the authors concluded that a more precise model was still necessary before this method could be introduced into practical computational pump design.

#### 2. Simulation model description

This paper presents a model for the cylinder block / valve plate interface which for the first time accounts for thermal and elasto-hydrodynamic effects through a unique multiphysics simulation environment. An overview of the new model is represented in Fig. 2. Three main modules, coupled together, embrace all the main physical phenomena in the lubricating interface.

# 2.1. Fluid flow solution

The fluid flow in the lubricating gap is governed by the Reynolds and the Energy equations. The first module solves these equations using the Finite volume method, describing the fluid flow in the lubricating gap, under any possible deformation or inclination of both the boundary solids. The non isothermal fluid flow solution is coupled with a multidimensional Newton-Raphson root finding method, which iteratively changes the micro-motion of the cylinder block relatively to the valve plate until the pressure field in the gap balances the external loads. The squeeze velocity of the cylinder block is then integrated in time to provide its relative position with respect to the valve plate.



Figure 1: Overview of the cylinder block / valve plate interface multi-physics simulation model

A precise estimation of the main oil properties (pressure, temperature, viscosity, velocity) is therefore calculated as a dynamic reaction to the oscillating external loads allowing for the interface load carrying ability, leakages and torque losses to be estimated under steady state conditions over one shaft revolution.

# 2.2. Elasto-Hydrodynamic problem

Under normal operating conditions of the machine the lubrication regime is elastohydrodynamic. In this situation the elastic deformation of the interface's boundary solids has a substantial impact on the pressure field and vice-versa, thus to predict the performance of the interface a rigorous approach to solve this fluid structure interaction is needed. In this work a partitioned fixed point iteration technique was used: pressure and deformation are calculated using different solvers and the results are successively corrected during the iteration process.

For the solution of the Reynolds equation a preconditioned bi-conjugate gradient stabilized solver is used, providing very high computation speed. The calculation of the elastic deformation with the finite element analysis is unfortunately much slower (almost two orders of magnitude) thus, to speed up the process, two sets of "influence matrices" are used: one for the cylinder block and one for the valve plate & end case assembly. The influence matrices technique is based on the offline calculation of the

elastic deformation under a set of reference loads and was already introduced in /13/ for the cylinder block / valve plate interface. Extensive details can be found in /21/.

**Figure 2** shows schematically how the problem is solved. At each time step the film geometry is calculated just by the relative position of the block with respect to the valve plate and the Reynolds equation is solved. With the resulting pressure field the viscosity is updated, the elastic deformations of the gap boundary surfaces are calculated using the "influence matrices" and finally the fluid geometry is corrected on both sides through these elastic deformations. Since the Reynolds equation is affected by both viscosity and film thickness, the algebraic equations obtained with the finite volume discretization will change and consequently they will need to be updated. This makes clear how under conditions of elasto-hydrodynamic lubrication, the Reynolds equation becomes strongly non linear. For this reason, when the pressure residual is calculated at the end of the iteration process (after the correction of the algebraic coefficients in the finite volume matrix) it will most likely be bigger than the permitted tolerance and therefore the aforementioned process is repeated until final convergence. When the film thickness becomes very thin (typically less than one micron) the non linearity in the problem is considerable and a variable under-relaxation



Figure 2: Simplified scheme of the solution loop used to solve the elastohydrodynamic lubrication problem

technique is used.

# 2.3. Heat transfer problem

The temperature is a very important parameter in the interface behavior, for many reasons. First of all, the oil properties (viscosity in particular) are strongly temperature dependent and therefore a good prediction of the film thickness can be achieved only if the temperature field in the gap is properly estimated. Moreover, the heat generated in the gap due to the viscous friction is partially taken out by the leakage flow, but still a considerable amount heats up the boundary solids, causing their thermal expansion.

Under normal operative conditions, the thermal expansions of the rotating kit of axial piston machines are generally of the same order of magnitude of the elastohydrodynamic deformation and besides, being non uniform, are responsible of important hydrodynamic effects. In this regard, very interesting and important findings have already been published in /17/ relatively to the piston / cylinder interface.

# 2.3.1. Cylinder block and valve plate temperature fields

The temperature fields in the cylinder block, valve plate and end case assembly is defined by the conductive form of the energy equation:

 $\nabla \cdot (\lambda \nabla T) = 0 \tag{1}$ 

Equation (1) is discretized through an unstructured formulation of the finite volume method. This allows the complex geometry of the solid parts to be described with a high level of detail. For the sake of brevity, the details of the solver implementation will be addressed in future publications, in favor of a thorough discussion of the boundary conditions, which are of fundamental importance. In this work, two types of boundary conditions were used:

- *Neumann boundary*: a fixed heat flux (uniform or non uniform) is applied to the boundary surface
- *Mixed boundary*: a heat flux is applied to the boundary surface through the equation  $q = \alpha (T_w T_\infty)$ , where  $T_w$  is the boundary surface temperature,  $T_\infty$  is the surrounding temperature and  $\alpha$  is the convection coefficient.

The Neumann boundary condition is used on the gap boundary surfaces of cylinder block and valve plate, because the heat flux to be applied is calculated from the fluid flow module, through the solution of the Energy equation in the gap. The mixed



Figure 3: Example of convection coefficient calculation on the HP side of a 100 cc unit, for a speed of 1000 rpm



Figure 4: Boundary surfaces used to calculate the temperature field for cylinder block and valve plate & end case assembly

boundary condition is used on all the other surfaces of the cylinder block, valve plate and end case, where instead the surrounding temperatures are assumed to be known (from measurements). The convection coefficients however are unknown and their estimation as a function of the machine geometry and operative conditions represents a problem.

The values of the convection coefficients in the different regions of the cylinder block, valve plate and end case used in this work were obtained through CFD analysis. Unfortunately due to limited space reasons, the details of this CFD investigation, together with a thorough presentation of the results can't be reported here and they will be addressed in a dedicated publication. **Figure 3** just shows an example of the convective coefficient calculation in the high pressure region of the machine, with a fixed temperature condition and a speed of 1000 rpm. In Fig. 4 are depicted the boundary surfaces used for the cylinder block and the valve plate & end case assembly thermal analysis. The associated boundary conditions are listed in Table 1.

Surface	BC type	Values
Bushings	Neumann	q = 15000 [W/m²]
Gap	Neumann	From gap
Dc	Mixed	h = 2000 [W/m <sup>2</sup> K ], $T_{\infty} = 0.5(T_{HP}+T_{LP})$
case_inner	Mixed	h = 1000 [W/m <sup>2</sup> K ], $T_{\infty}$ = $T_{case}$
case_outer	Mixed	h = 1500 [W/m <sup>2</sup> K], T <sub><math>\infty</math></sub> = T <sub>case</sub>
case_groove	Mixed	h = 1000 [W/m <sup>2</sup> K ], $T_{\infty}$ = $T_{case}$
Case	Mixed	h = 1500 [W/m <sup>2</sup> K], T <sub><math>\infty</math></sub> = T <sub>case</sub>
Metal	Mixed	h = 4000 [W/m <sup>2</sup> K], T <sub><math>\infty</math></sub> = T <sub>case</sub>
LP_port	Mixed	h = 1500 [W/m <sup>2</sup> K], T <sub><math>\infty</math></sub> = T <sub>case</sub>
LP_drill	Mixed	h = 3000 [W/m <sup>2</sup> K], T <sub><math>\infty</math></sub> = T <sub>case</sub>
HP_port	Mixed	h = 1500 [W/m <sup>2</sup> K], T <sub><math>\infty</math></sub> = T <sub>case</sub>

HP_drill	Mixed	h = 3000 [W/m <sup>2</sup> K], $T_{\infty}$ = $T_{case}$
Air	Mixed	h = 50 [W/m <sup>2</sup> K], T <sub>∞</sub> = 25°C

 Table 1: Boundary conditions used for the cylinder block and valve plate & end case

 assembly temperature field calculation, for a machine speed of 1000 rpm

#### 2.3.2. Thermal loads and thermal expansion

Once the body temperature is known, the corresponding thermal expansion can be easily calculated through a FEM analysis. In particular the linear expansion or contraction of all the body dimensions  $\varepsilon_T$  can be associated with the thermal loads through the volume integral

$$\boldsymbol{F}_{T} = \int_{V} \boldsymbol{B}^{T} \boldsymbol{C} \boldsymbol{\varepsilon}_{T}$$
(2)

where *C* is the constitutive matrix for an isotropic material with elastic modulus *E* and Poisson ratio v and *B* is the strain-displacement matrix. Equation (2) is simply summed to the other body and surface loads to get the working force deflection equation for the element, which represents the base of the finite element analysis.

$$\boldsymbol{k} \cdot \boldsymbol{u} = \int_{V} \boldsymbol{B}^{T} \boldsymbol{C} \boldsymbol{B} d\boldsymbol{v} \cdot \boldsymbol{u} = F_{E} + F_{T} = F_{E} + \int_{V} \boldsymbol{B}^{T} \boldsymbol{C} \boldsymbol{\varepsilon}_{T} d\boldsymbol{v}$$
(3)

Knowing the expression for the element stiffness matrix k, the nodal forces vector due to external loads  $F_E$  and the nodal forces vector due to thermal induced stress  $F_T$ , the nodal displacements vector u is solved, upon imposing the constraints conditions. Equation (3) is referred to a single element, while the solid is discretized using a large number of them. Therefore, between the element processing and the body nodal displacements solution, the assembly process constructs the master stiffness matrix Kand the master equation, related to the whole discretized solid. The typical working temperatures of axial piston machines generate consistent thermal expansions. For the 100 cc unit analyzed in this work, the thermal deformations of cylinder block, valve plate and end case assembly varied approximately from 10 to 150 microns in a speed range of 1000 to 3000 rpm and pressure range of 100 to 350 bar. However, what actually affects the lubrication behaviour is not the absolute deformation, but rather the relative one. In particular, what is important is just the change in flatness of the gap's boundary surfaces introduced by the thermal expansion. Figure 5 shows this concept: on the left the overall thermal deformation of the valve plate & end case assembly, on the right the relative deformation of the gap surface: the uppermost point of the deformed surface is defined as the reference (with a deformation equal to zero)



**Figure 5:** Relative deformation of the gap boundary surface on the valve plate side whereas the deformation of all the other points is calculated as a difference respect this reference.

# 3. Numerical results

Figure 6 shows a comparison between measured and simulated temperature fields of



 $\Delta p$  = 300 bar, n = 1000 rpm,  $\beta$  = 100%

Figure 6: Comparison of the simulated and measured valve plate surface temperature field for two different operative conditions.

measured

simulated

the valve plate surface for two operative conditions. The simulated temperature fields come directly from the output of the simulation model, whereas the measured temperature fields were obtained by interpolating the data from the 28 thermocouples installed underneath the valve plate surface. In addition, for a better comparison, the thermocouples position and the corresponding temperature values are clearly represented in Fig. 6 in both the cases. The measured temperatures at the low and high pressure ports and at the pump casing were used as an input in the simulation model and are clearly indicated in Fig. 6. Additional information about the test stand can be found in /15/.

The results in Fig. 6 represent a substantial improvement respect the outcome of /15/: for both the operative conditions the complex temperature field is generally very well matched. In particular, the two regions at higher temperature at the top and the bottom are accurately predicted by the simulation model in term of position, extension and magnitude (the maximum relative error is below 13%). Very interesting is the low temperature region on the HP side, which was observed experimentally and confirmed by the simulation model.

The interpretation of this particular nature of the temperature fields in Fig. 6 may appear difficult, especially if just the experimental data is available. Only the knowledge of the film shape, gap flow and overall heat exchange in the machine can help the understanding; this information is fully provided by the simulation model. The lubricating film predicted by the simulation model, corresponding to a shaft angle of  $^{\circ}$  for the first operative condition in Fig. 6 is represented in Fig. 7. It can be noticed that the regions at lower film thickness correspond exactly to the two high temperature



**Figure 7:** Lubricating film shape and pressure field, for a shaft angle position of  $0^{\circ}$  and operative condition of  $\Delta p$  = 300 bar, n = 1000 rpm and maximum swash plate angle. The magnification factor in z direction is 1000x.

areas in Fig. 6. In fact for the same value of circumferential velocity of the oil (owing to the cylinder block dragging) its gradient in the z direction becomes much higher in these regions and therefore the viscous dissipation results much stronger than anywhere else. This particular gap shape is determined by the downward deformation of the gap's periphery in the *x* direction, which is explained by the thermal deformation of the valve plate & end case assembly shown in Fig. 5.

The concave shape of the cylinder block sealing surface (due the cylinder block thermal expansion) besides the elastic deformation of both the bodies under the high pressure field, make instead the gap thicker at lower radius, with maximum thickness around the high pressure port. This natural modification of the gap determines a moderate leakage (0.5 l/min), mainly localized in the high pressure region. Here the oil flows all the way through the sealing land, from the displacement chambers and valve plate port to the machine casing. Although this pressure driven component of the oil velocity field partially contributes to the viscous dissipation, its main effect is the cooling of the high pressure region (because a constant flow of fresh oil replaces the overheated one) explaining the experimental evidence.

# 4. Conclusion

In this paper a novel simulation model for the cylinder block / valve plate interface was described. For the first time, precise estimations of thermal and elasto-hydrodynamic effects were fully coupled together providing an accurate description of the fluid film thickness as a function of the main operative parameters.

The model prediction of the valve plate surface's temperature field was very well confirmed by measurements under different operative conditions. The temperature field in the gap and in the boundary solids is directly influenced by the film thickness: a wrong prediction of the correct gap shape would lead to unreasonable leakages and heat fluxes and consequently to major discrepancies with the experimental evidence. According to this consideration, the simulation model seems to capture all the main physical phenomena involved in the machine operation, appearing much more mature than its predecessors for practical computational machine design.

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# 6. Symbols

р	Pressure	N/m <sup>2</sup>
Т	Temperature	К
q	Heat flux	W/m <sup>2</sup>
α	Convection coefficient	W/Km <sup>2</sup>
λ	Thermal conductivity	W/Km
В	strain-displacement matrix	m <sup>-1</sup>
с	constitutive matrix for an isotropic material	N/m <sup>2</sup>
Е	Linear expansion	-
k	Element stiffnes matrix	N/m
u	Nodal displacement vector	m