# Solving Piston Secondary Motion of Internal Combustion Engines

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## INTRODUCTION

This project aims to develop a numerical model of piston secondary motion of internal combustion engines. The piston dynamics are mainly driven by the pressure load and connecting rod angle. Ideally, the piston has only one degree of freedom, which is the reciprocating motion along the cylinder axis, which is usually referred to as primary motion. More realistically, the piston has six degrees of freedom. The secondary motion of piston is made up of translation perpendicular to the cylinder and piston pin axis (usually called piston lateral motion), and rotation about the piston pin axis (usually called piston tilt). This motion is very important in understanding the piston skirt lubrication, slap noise and friction. The piston's motion due the remaining three degrees of freedom is currently believed to be significantly smaller than both its primary motion and secondary motion and is neglected for the purpose of this project.

## NUMERICAL MODEL

Figure 1 shows the power cylinder system, which is made up of the crankshaft, connecting rod, wrist pin, piston, ring pack and cylinder bore. For the purpose of this project, we assume that the crankshaft is rigid, and carries only rotational motion, whose angular speed is defined as a constant. At each crank angle position, the primary motion of the piston is first solved directly. The combustion pressure trace and piston is used to calculate axial forces acting on the piston. Axial force balances on the piston and wrist-pin are then used to define the piston – wrist-pin and wrist-pin – connecting-rod axial forces. Then a moment balance around the large end of the connecting-rod, assuming that both the large end and the small end are ideal, frictionless, pin joint, is used to get the side force acting on the piston from wrist pin. This force, in some sense, is the driving force for piston lateral motion. Then the lateral force and moment balances on the piston must then be satisfied and are used as the basis for the iterative solution method.

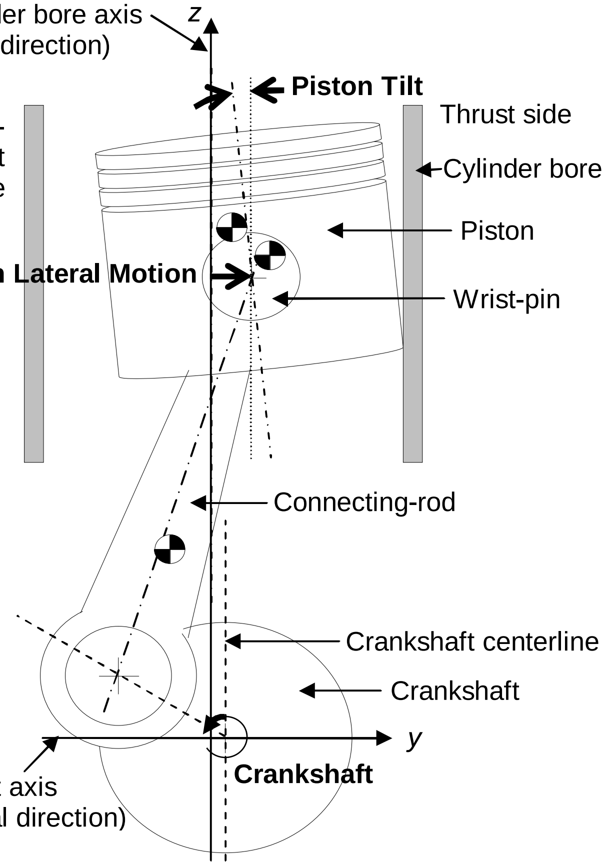


Figure 1: System Schematic.

PISTON – CYLINDER BORE INTERFACE – The piston – cylinder bore interface model is the critical component of this piston secondary motion model. This interface is essentially made up of three components:

- the deformed piston surface geometry, taking into account piston motion,
- cylinder bore geometry, which is considered to be stationary and rigid,
- oil film between the piston skirt and cylinder liner, where the hydrodynamic pressure is generated.

<u>Piston Geometry</u> – The piston's shape is specified via a nominal radius, radial variation along the thrust axis as a function of axial position, and ovality as a function of axial position, as shown in Figure 2.

Cylinder Bore Geometry - The shape of the cylinder bore is assumed to be a perfect circle.

Compliance Matrix - A compliance matrix is used to describe the radial mechanical deformation of the piston due to the hydrodynamic pressure distribution which is generated in the oil film between the piston skirt and cylinder liner. The compliance matrix used for these simulation results was generated using an FEA model of the piston, and supplied by one of the project sponsors. The diagonal of the compliance matrix, which represents the local deformation of the piston due to a unit load at the same point, is shown in Figure 3a. From this we can note that there is significant variation in surface compliance over the piston skirt, with the lower and central regions of the skirt being much softer than the upper and outer parts of the piston.

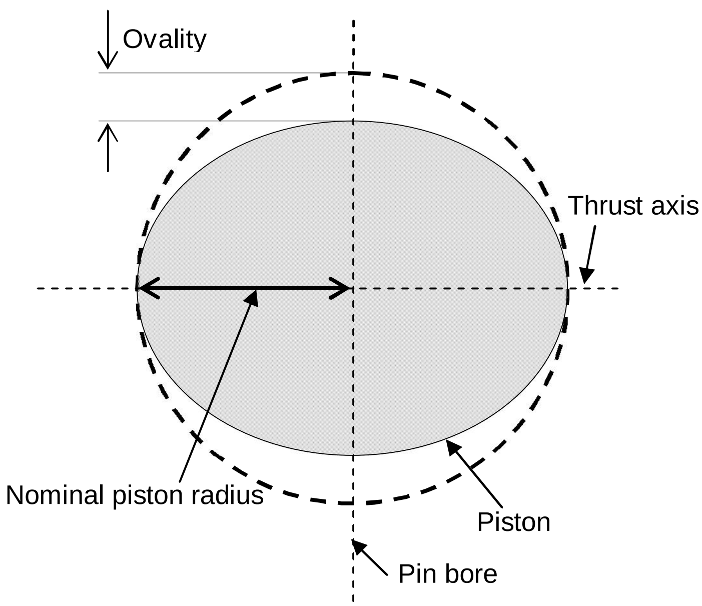


Figure 2: Piston Shape

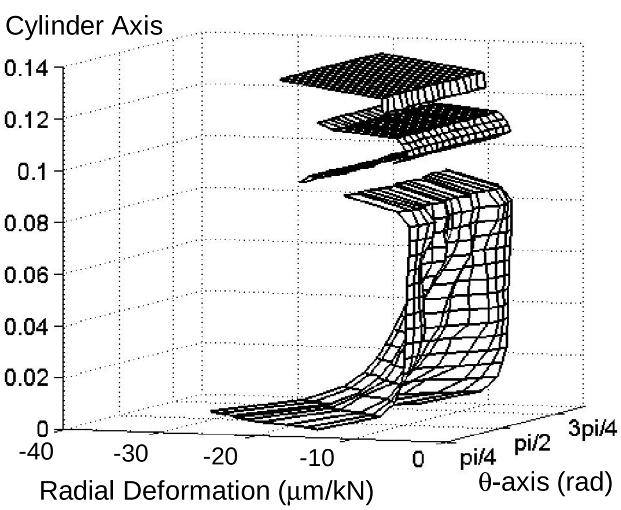


Figure 3: Compliance matrix diagonal

Fluid Model – The governing equation describing the lubrication flow (Figure 4) is Reynolds Equation as follow, in which we assume the viscosity of the oil is constant.

$$\left[\frac{\partial}{\partial z}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial z}\right)+\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial x}\right)\right]=\frac{\partial}{\partial z}\left[\frac{\rho h W}{2}\right]+\frac{\partial(\rho h)}{\partial t}$$

W in this equation is the sliding speed of the piston skirt.

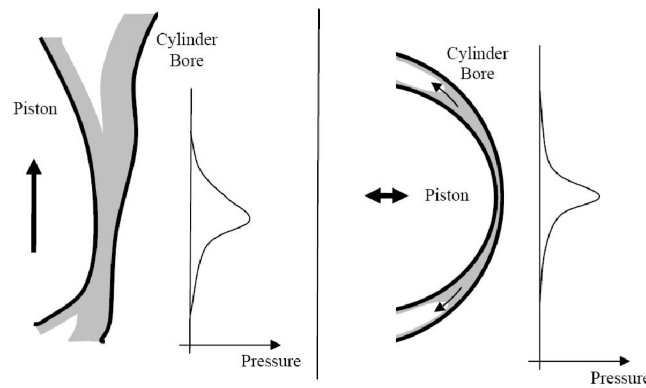


Figure 4: piston-cylinder bore lubrication flow

However, this equation does not take into consideration the fact that the oil will cavitate at locations where pressure drops to cavitation pressure, forming oil vapor. Our fluid model must therefore be extended to include this phenomenon. Then we use a Universal Reynolds equation, based on JFO cavitation theory, as the governing equation. In the full film region, the density of the oil is assumed to be constant, in the cavitation region, the pressure is constant and equals to the cavitation pressure, however, in cavitation region, due to different fraction of oil vapor, the density may vary.

$$\frac{\partial}{\partial z}\left[h^3\frac{\partial(F\Phi)}{\partial z}\right]+\frac{\partial}{\partial x}\left[h^3\frac{\partial(F\Phi)}{\partial x}\right]=\gamma\frac{\partial}{\partial z}\left\{[1+(1-F)\Phi]h\right\}+2\gamma\frac{\partial}{\partial \hat{t}}\left\{[1+(1-F)\Phi]h\right\}$$

In this Universal Reynolds Equation, F is introduced, which is 1 in the full film region and 0 in a partial film region, and a global variable, Φ, which when combined with F allows both the pressure, p, and the density, ρ, to be determined simultaneously throughout the calculation domain. For the full film region, where F = 1, density is constant and equals the density of the oil, and p = Φ, this equation reduce to Reynolds Equation. For the partial film region, where F = 0 and pressure is constant and equals cavitation pressure, [1+(1−F)Φ] describes density of the combined vapour and fluid mixture. Then control volume method is used to discretize the equation.

The boundary conditions for the lubrication flow right now are set to be full oil boundary conditions. That is, there are full oil in the boundaries and the boundary pressure to set to be a constant – the crankcase pressure, which is adequate.

Numerical Algorithm – Combining these equations of motion of the piston, fluid flow and structural deformation, we form a nonlinear system. Consider at a given crank angle position, we can first get the side force acting on the piston from wrist pin by solving the piston primary motion, then what need to do is determine the piston lateral motion and piston tilt. And at the same time the calculated forces and momentums acting on the piston can balance. Now suppose we first have an initial guess of piston secondary motion as well as piston skirt deformation (at each node), then we can get the corresponding height profile between the piston and cylinder. Then we solve the Universal Reynolds Equation to get the hydrodynamic pressure generated in both anti-thrust and thrust side of the piston with this height profile. Then we can find that usually the forces and momentums will not balance, also, the deformation of the piston skirt due to this hydrodynamic pressure may not equal the deformation we initially guessed. So a Newton's method is used to solve this nonlinear problem.

## RESULTS

First, the role of piston skirt deformation is examined. In the case of 660 degree crank angle position, 1um minimum rigid clearance (the clearance when the piston skirt has no deformation and the piston is placed in the middle), Figure 5 shows the pressure distribution is the initial guess of piston skirt deformation is zero.

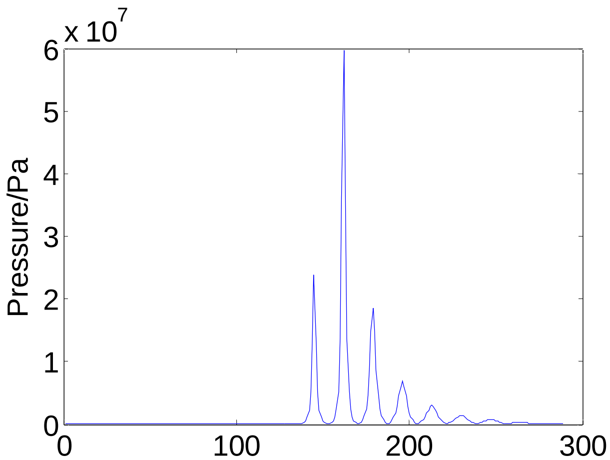


Figure 5: pressure distribution without deformation

We can clearly see the sharp pressure peak with this initially guessed zero deformation profile. Then, the piston skirt will deform due to this pressure load and hence make the gap between the piston skirt and cylinder liner larger, and correspondingly smaller pressure generated. After 5 Newton's iteration, the actual pressure distribution is shown in Figure 6. Then we can find that piston skirt deformation has an important role in the hydrodynamic pressure generation, which efficiently decreases the peak pressure, and makes that the side load be supported by a larger area (since the pressure is distributed over a wider region). Figure 7 shows the actual deformation we get.

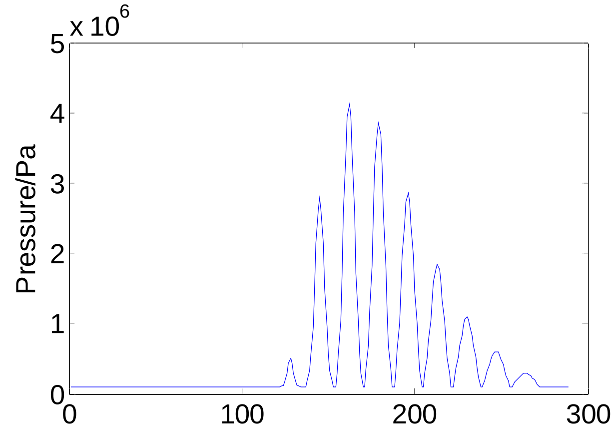


Figure 6: actual pressure distribution

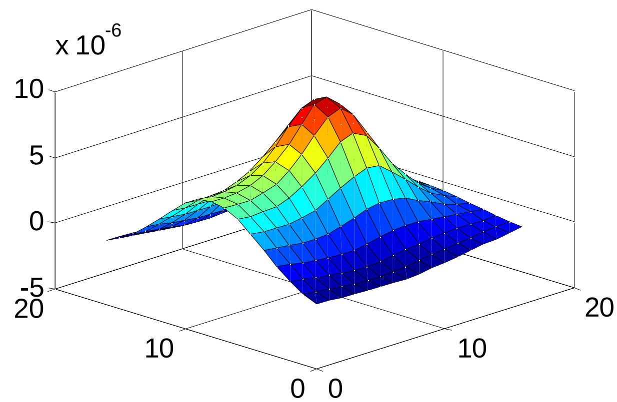


Figure 7: piston deformation

Figure 8 shows the load used in the simulation, which is the force on top of piston due to the pressure in combustion chamber. And we can see at about 360 degree crank angle, due to combustion, there is a load peak.

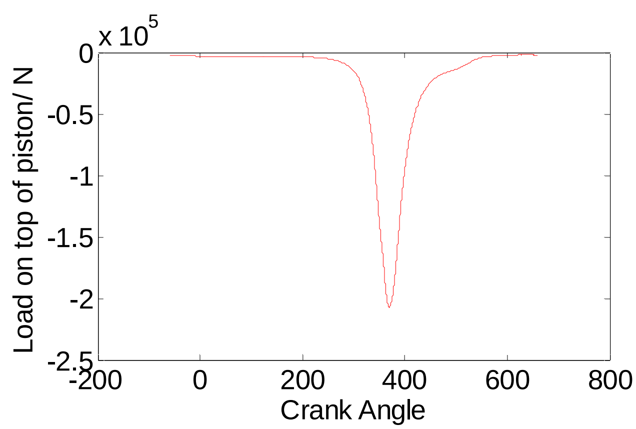


Figure 8: load on top of piston

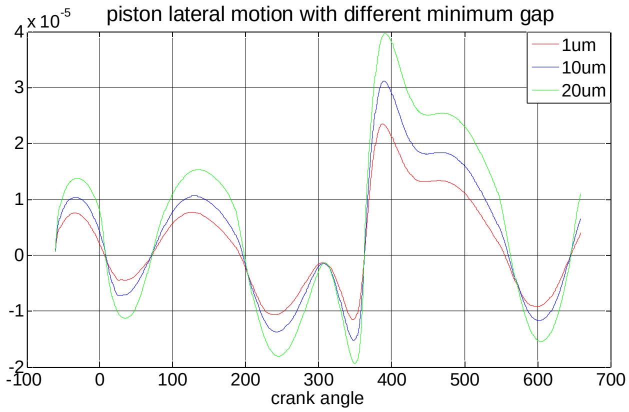


Figure 9: piston lateral motion with different minimum gaps

Figure 9 shows the calculated piston lateral motion for one engine cycle, with different minimum rigid clearances. We can see that at zero crank angle, when the piston is at top dead center, the piston will move from thrust side (positive lateral position) to anti-thrust side (negative lateral position), this is because now the engine is at intake stroke, and the connecting rod pulls the piston down, and due to the negative connecting-rod angle, the piston is pulled to anti-thrust side. Then at about mid- intake stroke, the inertia force will change sign (the piston will change from accelerating to decelerating), the piston then move to the thrust side. See also the peak lateral position at about 380 degree crank angle. It is because now is the power stroke, and from figure 8 we can find that now the load on top of the piston is very large and this force- accompanied with the connecting rod angle will push the piston to the thrust side. And since now the load and correspondingly the side force is very large, the piston skirt will have very large deformation and hence result in a large lateral position.

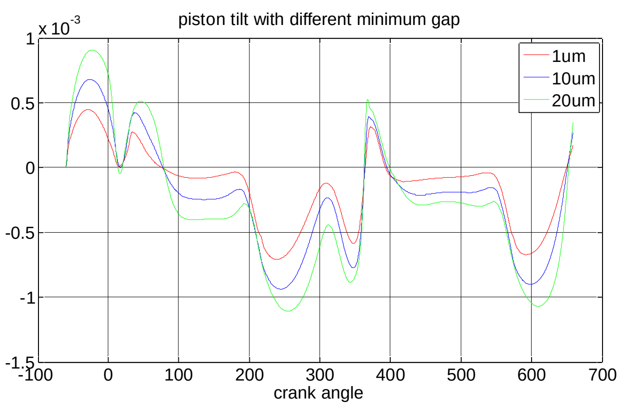


Figure 10: piston tilt with different minimum rigid clearance

Figure 10 shows the calculated piston tilt with different minimum rigid clearances.

From figure 9 and figure 10, we can see piston lateral motion and piston tilt increase with rigid minimum clearance, which is understandable. Since the rigid minimum clearance get larger, the piston will laterally moves more in order to have the same magnitude of deformation (which in some sense represents the side load). Generally, the piston lateral motion is driven by the side force from wrist pin. To illustrate this, figure 11 shows the lateral forces on piston. From this figure, we can see, over the cycle, the primary forces are the side force acting on piston from wrist pin (red line) and the reacting force from cylinder bore (blue line).

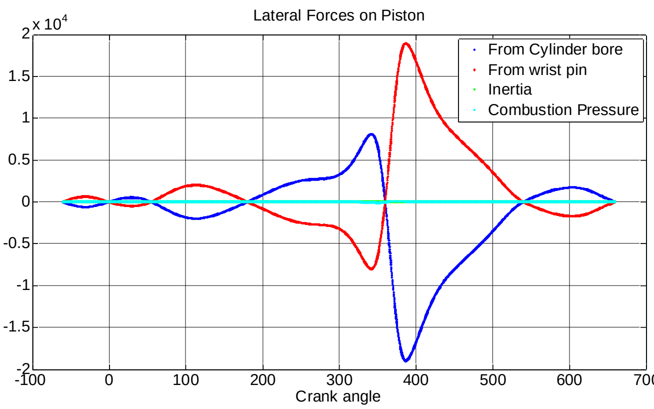


Figure 11: lateral forces on piston

Then the influence of ovality on piston lateral motion is examined. Figure 12 shows the simulated piston lateral motion with different ovality. The result shows with larger ovality, the piston has larger lateral motion. The explanation is with larger ovality, the piston shape (viewed from top) is more flat, and hence will have smaller resistance while the piston is sliding. Then we can imagine with the same rigid minimum clearance, larger ovality means smaller hydrodynamic pressure can be generated. So, for the hydro-pressure to balance the same amount of side force from the wrist pin, the piston must moves more, that is, the actual height between piston and cylinder must be smaller. This is why larger ovality may cause larger piston lateral motion.

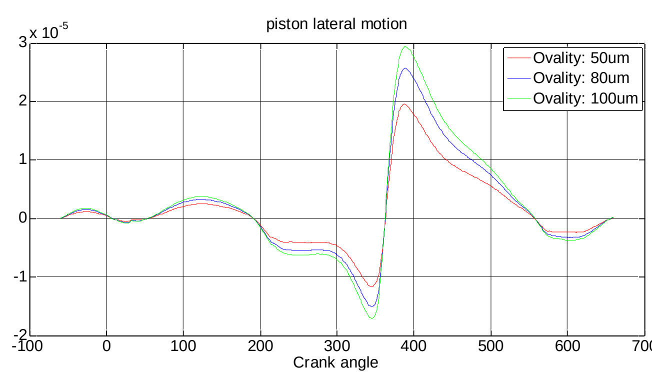


Figure 12: piston lateral motion with different ovality

Figure 13 shows the piston tilts with different ovality and the explanation is similar.

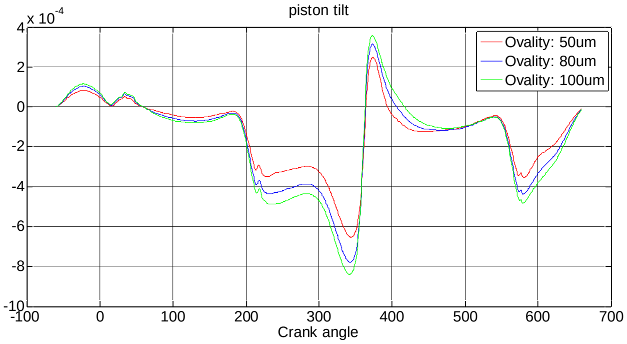


Figure 13: piston tilt with different ovality

Figure 14 shows the piston lateral motion with different engine speeds. And we can see with larger engine speed, the maximum piston lateral motion (which happens at about 380 crank angle) gets smaller. This is because as the engine speed increases, the sliding speed of the piston skirt over the cylinder liner increases. Clearly, from Reynolds equation, we know a larger sliding speed will help develop larger hydrodynamic pressure with other parameters the same, so the piston only needs to move less to generated the same amount of hydro-pressure to balance the side force from wrist pin.

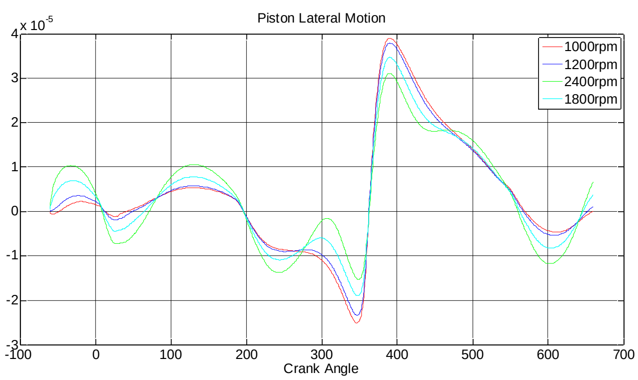


Figure 14: piston lateral motion with different engine speed

Another thing that is interesting is the cavitation phenomena. Figure 15 shows a pressure distribution of thrust side at 660.5 crank angle. Now we can see now the cavitation region appears in the lower part of the piston (the left part in the picture where is pressure is constantly cavitation pressure, 0.955bar). Now the piston is sliding upward and at this situation, the cavitation is caused by the sliding of the piston as well as the curvature of the piston profile. Figure 16 shows a pressure distribution of thrust side at 365 degree crank angle. Note that now there is one more cavitation region-in the upper region of the piston. The reason for this cavitation is not due to the sliding of the piston, but due to that now the piston is tilting with its upper region leaving the thrust side.

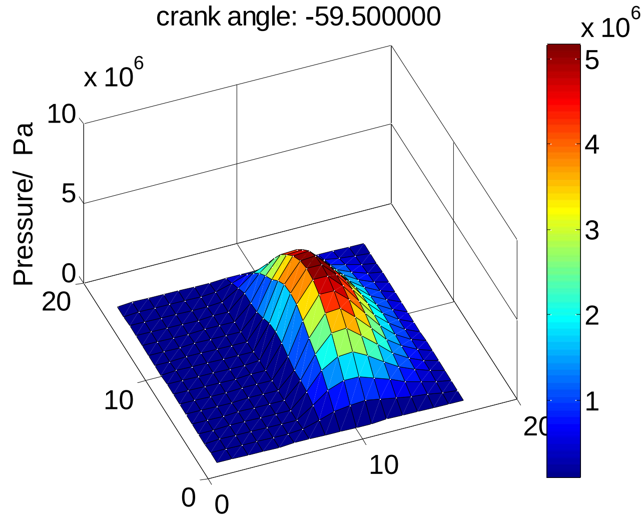


Figure 15: pressure distribution of thrust side

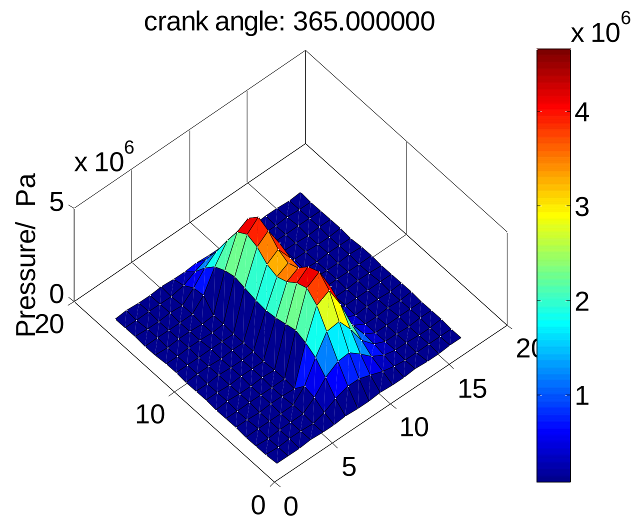


Figure 16: pressure distribution of thrust side

## SUMMARY

A numerical model of piston secondary motion is developed, with both cavitation and piston skirt deformation taken into consideration. The model gives meaningful results. Also, the model is very fast, only about 10 minutes is needed to run for one engine cycle. This course project is also my research project, and in the future, more realistic geometry will be used, for example, the piston themal expansion and the cylinder bore distortion. Also, right now, I use a 17 by 17 grid, which is sufficient for the piston deformation, but is not sufficient for the calculation of hydrodynamic pressure. So finer grid(which will be more costy) will be used in future to improve accuracy.