Proceedings of the World Congress on Engineering 2010 Vol II
WCE 2010, June 30 - July 2, 2010, London, U.K.

Modeling of Piston Top Ring Lubrication by considering Cylinder Out-of-Roundness in Initial Engine Start up

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Abstract. **The piston top ring in an automotive ring-piston system plays a very crucial role during the engine start up and normal operating conditions. A hydrodynamic piston top ring lubrication model in the initial engine start up is developed by incorporating an elliptical cylinder bore out of roundness effect due to liner wear, manufacturing error and cylinder head clamping force. Numerical analysis is presented based on two dimensional Reynolds equation having iso-viscous regime and constant lubricant density. Top ring liner contact profile coupled with piston assembly motion, hydrodynamic pressure field generation and time based lubricating film thickness profile as function of crank rotation are examined and influence of bore out of roundness effect at the time of engine initial start up are investigated. This study suggests that hydrodynamic pressure and lubricant film thickness profiles are affected by cylinder out of roundness due to wear in the initial engine start up conditions.**

Key words: **Piston top ring, Hydrodynamic lubrication, Bore out of roundness, Initial engine start up.**

I. INTRODUCTION

The piston top compression ring plays a vital role in an efficient engine operation as it prevents the combustion gas leakage and allows heat dissipation but contributes towards mechanical friction. Under severe operating conditions, the ring-block interface contributes about 20% of the total engine mechanical frictional loss [1]. Hence, the piston rings are lubricated by oil, the film thickness of which results in low friction and reduced wear [3]. Major factors affecting the oil film thickness are bore distortion, piston speed, lubricant viscosity, top ring face profile, ring flexibility, boundary conditions and surface roughness effect [4].

Many researchers modeled piston ring lubrication in normal engine operating conditions and in different lubricating regimes by considering rigid and elastic contacts, boundary conditions, surface roughness, bore out of roundness, ring fluttering under its groove and ring twist [6-10]. None of them modeled the phenomenon in the initial engine start up conditions. Most of the earlier ring lubrication models were based on the assumption of fully conformal (axisymmetric) top ring and liner surfaces, but in reality liner is not perfectly circular due to manufacturing errors and radial distortion due to cylinder wear, combustion chamber pressure, thermal effects and head clamping force [3]. Moreover, cylinder bore of the used engine clearly indicates excessive wear just below the Top Dead Centre (TDC) due to defective lubrication [17], which implies that under starved lubrication conditions in initial engine startup maximum wear is likely to occur between the top ring and the cylinder near TDC. In normal engine operating conditions most of the piston stroke in one single stroke witnesses hydrodynamic ring lubrication [13]. It is experimentally determined that motoring ring friction prior to engine startup is 50% of its value under firing conditions [14]. It implies that the moment engine starts after cranking (motoring) piston ring friction value shoots up to double the value of that during engine motoring. During normal engine operation at low velocity near TDC and Bottom Dead Centre (BDC), lubricating film is thinner and there are chances of piston ring and liner contact [15]. In the initial engine startup, reduced engine speeds at low loads are likely to maintain thinner lubricating film and the chances of wear due to ring and liner contact are further enhanced. At low load level, top piston ring experiences most critical lubricating conditions close to TDC, when the oil film is the thinnest. An increase in load results in significant increase of film thickness near TDC on the thrust side [16]. In case of initial engine start up, load at idle rpm can be categorized as at lowest low-load levels.

This study models two dimensional top piston ring hydrodynamic lubrication in the simulated initial engine startup conditions by considering non-axisymmetric out of round (elliptical) cylinder for parabolic ring face in sliding direction. Parabolic face profile has the advantage that it tends to be self perpetuating under wear since ring tends to rock inside its groove during reciprocating movement and causes preferential wear of its edges [5]. This model generates hydrodynamic pressure fields and minimum hydrodynamic film thickness profiles as functions of engine crankshaft rotation of 720 degree, apart from calculating the friction coefficient. To do all this, it is essential to take the following logical assumptions:

1. The lubricant is an incompressible Newtonian fluid and the flow is laminar.
2. Side leakage, oil starvation and surface roughness factors are neglected.
3. No relative motion between piston ring and its groove.

Manuscript received March 17, 2010. This work was supported by College of Electrical and Mechanical Engineering, National University of Sciences and Technology (NUST). Financial support was provided by Higher Education Commission, Pakistan.

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ISBN: 978-988-18210-7-2
ISSN: 2078-0958 (Print); ISSN: 2078-0966 (Online)

4. Piston secondary motion and ring twist are neglected during the normal engine operation.
5. Iso-viscous case, that is, viscosity is same in circumferential and sliding directions.
6. The lubrication is pure hydrodynamic with fully flooded inlet and Reynolds exit conditions.
7. The surfaces of the ring and the liner are perfectly smooth and rigid.
8. Thermal effects due to combustion are neglected.

II. MATHEMATICAL MODEL

A. Basic Equations of Piston Motion

The governing equation of piston motion is defined as function of crank angle [11].

$$U = r\omega sin\theta + r\omega(C_p + rsin\theta)cos\theta(l^2 - (C_p + rsin\theta)^2)^{0.5} \quad (1)$$

where $\omega = V/r$ (2)

B. Reynolds Equation and Hydrodynamic Pressures

According to our assumptions two dimensional Reynolds equation is used to calculate hydrodynamic pressures and lubricating film thickness as suggested by G.W. Stachowiak et al [5]. In non-dimensional form it is given by:

$$\frac{\partial}{\partial x*^2}\left(h^{*3}\frac{\partial p*}{\partial x*}\right) + \frac{b^2}{L^2}\left[\frac{\partial}{\partial y^*}(h^{*3}\frac{\partial p^*}{\partial y^*})\right] = \frac{\partial h^*}{\partial x^*} \quad (3)$$

The boundary conditions for Reynolds equation are [11]:

$$\frac{\partial p}{\partial x_{x=0}} = \frac{\partial p}{\partial x_{x=b}} = 0$$

p (0,y) = 0
p (b,y) = 0

The film thickness for a parabolic face profile and bore out of roundness can be expressed as

$$h = h_1(x) + h_2(y) + h_0 \quad (4)$$

where $h_1(x)$ is film thickness in sliding direction [12]

$$h_1(x) = \begin{cases} S_1(x^2 - 2a_1x + a_1^2) & -b/2 \le x \le a_1 \\ 0 & a_1 < x < a_2 \\ S_2((x^2 - 2a_1x + a_2^2) & a_2 \le x \le b/2 \end{cases} \quad (5)$$

$h_2(y)$ is film thickness in circumferential direction [9]

$$h(y) = R\left\{\frac{1}{\sqrt{1-\left[1-\left(\frac{R}{R+\Delta c}\right)\right]\sin\theta}} - 1\right\} \quad (6)$$

where
$\theta = \frac{y}{R}$; $0 < y < 2\pi R$

h_0 is important adjustable factor to iterate during computation to converge solution to desire criteria. Net resultant hydrodynamic force is assumed to be in equilibrium with applied Gas pressure force [3].

$$W = \int_0^L \int_0^b p(x,y)dx\,dy \quad (7)$$

Equilibrium condition in the radial direction is given by:

$$FG - W=0 \quad (8)$$

The profile of combustion gas as a function of crank angle rotation is shown in Figure 1.

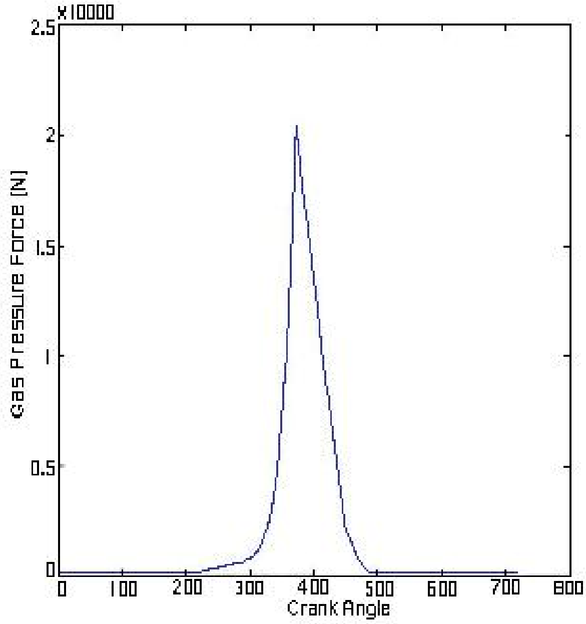

Figure:1 Gas Pressure Force Vs Crank Angle [11]

III . NUMERICAL SOLUTION PROCEDURE

Finite difference approach is adopted to solve the governing equations. While computing, the initial value of 'h_0' is assumed and hydrodynamic pressures are calculated accordingly. Then, hydrodynamic pressures are integrated over ring face area to determine the net resultant load 'W'. If equilibrium condition is satisfied, crank angle gets new increment and the whole procedure is repeated for updated crank angle. If equilibrium condition is not satisfied, then the value of 'h_0' is updated according to the following expressions [3]:

$$h_{0,\ approx}^{(k)} = (\frac{FG}{W})^{\gamma} h_{0,old}^{(k)} \quad (9)$$

$$h_{0,new}^{(k)} = h_{0,old}^{(k)} + \lambda_1(h_{0,approx}^{(k)} - h_{0,old}^{(k)}) \quad (10)$$

where γ is empirical coefficient ranging from 0.1 to 0.2, λ_1 is an under-relaxation factor equal to 0.15-0.5 and *k* denotes the crank angle position [3]. If gas pressure force and the net resultant force differ by 0.5% it is assumed that convergence criteria is satisfied, the minimum film thickness and the hydrodynamic pressures are calculated for complete cycle by iterative process.
Flow chart for the aforementioned procedure is given in Fig. 2

ISBN: 978-988-18210-7-2
ISSN: 2078-0958 (Print); ISSN: 2078-0966 (Online)

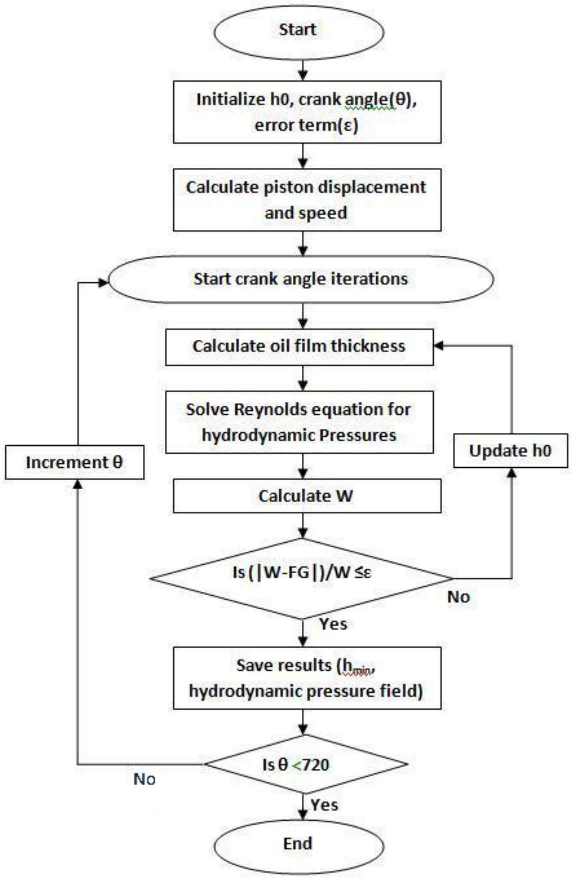

Fig:2 Flowchart of Computational Scheme

IV. NUMERICAL RESULTS AND DISCUSSION

A. Effect of Minimum Film Thickness

Minimum lubricant film thickness between the opposing surfaces of piston top ring and cylinder liner as a function of 720 degree crank rotation is determined as shown in figure 3. Intake stroke has film thickness profile in the shape of bell shape curve, which shows film thickness rise to attain peak value when piston is at the mid of induction stroke. It implies that during the first half of the induction stroke, there is continuous oil film build up between the opposing ring and the liner surfaces to cover extra space created by distorted cylinder liner due to wear. This corresponds to an increase in piston velocity and shearing of lubricant till the mid of stroke. After midpoint there is a gradual drop in piston speed till it reaches the end of the induction stroke at BDC. For the entire duration of the induction stroke, the hydrodynamic pressures are expected to remain in the minimal range and hence there is substantial oil film thickness but the generated hydrodynamic pressures are not enough to provide sufficient lift to fully separate opposing surfaces in relative motion. In the compression stroke, the piston travel from BDC to TDC involves continuous reduction of minimum film thickness till it is reduced to near minimum value. The film profile curve shows a gentle reduction from BDC to the mid of stroke when there is corresponding rise in piston speed and there is some build up of hydrodynamic pressures compared to earlier pressure values. After the mid of compression stroke the descending slope gets steep due to the fact that there is significant build up of hydrodynamic pressures due to the corresponding sharp rise in piston chamber compression pressures prior to actual combustion. Nearing TDC, the top ring is expected to have low intake of lubricant and there is oil scarcity, which may affect its lubrication and in case of an unexpected impulse or sudden shock, there are chances of lubricant side leakages due to squeezing action. Just after TDC and at the beginning of the expansion stroke, the actual combustion takes place, which is transferred to the top ring in radial direction. At that instant, in reality there is partial starvation of lubricant and hence thinnest lubricant film is witnessed. The piston travel towards the midpoint after combustion gas force effects subside results in rebuild up of the minimum film thickness due to further lubricant supply and corresponding sudden rise in hydrodynamic pressures. For the entire duration of the expansion stroke, another bell shaped curve of lesser magnitude is generated. The difference between the earlier similar shape curve (Induction stroke) and this curve is that there was oil flooding and lubricant flow without substantial rise in hydrodynamic pressures in induction stroke but here although there is less flooding but a substantial rise in hydrodynamic pressures due to combustion gas force effect, which means enhanced load carrying capacity of lubricant hydrodynamic film to conveniently separate opposing top ring and cylinder liner surfaces and substantial reduction of chances of possible wear of ring or liner. At the end of expansion stroke, the lubricant film thickness attains lower value but it is still more than the one noticed at the end of compression stroke. There is another sharp rise in film thickness magnitude starting from BDC till piston has travelled two-third of the distance in the exhaust stroke but the peak value is far less than other peak values. It is due to the fact that on one hand there is lubricant supply but on the other there is sudden drop in magnitude of the hydrodynamic pressures due to visibly reduced impact of combustion gas force. At around 700 degree crank angle and after attaining peak value, magnitude of lubricant film thickness sharply reduces to the minimum value. This exponential drop is indicative of oil starvation, minimal piston speed and significantly reduced hydrodynamic pressures till piston reaches at the end of the exhaust stroke.

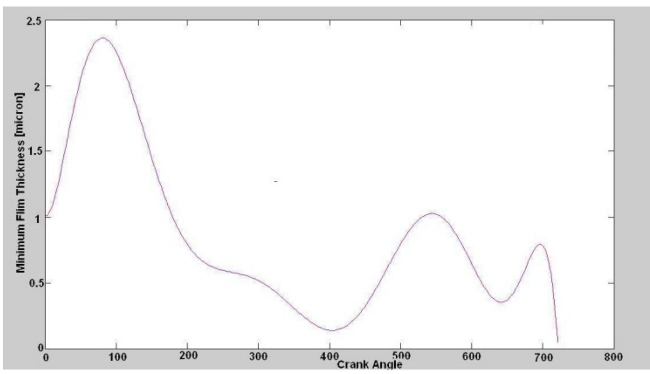

Fig:3 Minimum Film Thickness (Microns) Vs Crank Angle

B. Discussion on Coefficient of Friction

Top compression ring friction significantly contributes towards overall engine friction losses. Modeling initial engine start up conditions by focusing on hydrodynamic friction force and applied load involve coefficient of friction and viscous shear flow calculations. To recognize its significance, the coefficient of friction is calculated and

ISBN: 978-988-18210-7-2
ISSN: 2078-0958 (Print); ISSN: 2078-0966 (Online)

plotted against 720 degree crank rotation for top ring lubrication in out of round worn cylinder bore as shown in Fig:4. Coefficient of friction gets instantly reduced at the beginning of the induction stroke and generally maintains a constant value for the entire duration of induction stroke. It corresponds to small magnitude of applied load and smaller values of hydrodynamic friction force. A sudden rise and immediate fall in the form of a spike at the beginning of compression stroke is followed by a constant value of the coefficient of friction covering the entire compression stroke, actual combustion period and till piston crosses the midpoint in the expansion stroke. Instantaneous reduction and increase in the coefficient of friction at slow piston speed at 490 degree crank angle is followed by slightly further rise in its value at the end of the expansion stroke. For the entire duration of the exhaust stroke, the value of coefficient of friction remains constant. It is quite obvious that there is no appreciable change in the coefficient of friction related to top piston ring lubrication except that at the time of initiation of the 4-stroke engine cycle there is significant instantaneous drop in its value. It implies that in the initial engine start up conditions, the top ring friction is substantial and does not appreciably decrease with the change in piston cyclic speed or magnitude variations of the applied load.

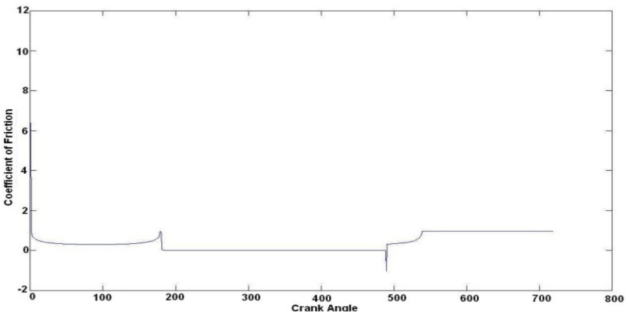

Fig:4 Coefficient of Friction Vs Crank Angle

C. Discussion on Hydrodynamic Pressure Fields

In the top piston ring lubrication model, eight pressure fields are selected at different crank angles to highlight appreciable instantaneous changes in their magnitudes and their correlation with the corresponding minimum film thickness profiles at engine start up speed of 600 revolutions per minute (RPM).

At 90 degree crank angle when piston reaches at the mid of intake stroke, positive hydrodynamic pressures rise over the surface plane of piston top ring axial and circumferential directions as shown in Fig: 5. The slope rises gently and peak hydrodynamic pressure values are found to be very less, which further clarifies that such high values of minimum oil film thickness in the mid of induction stroke are due to low hydrodynamic pressures (Refer to Fig: 3).

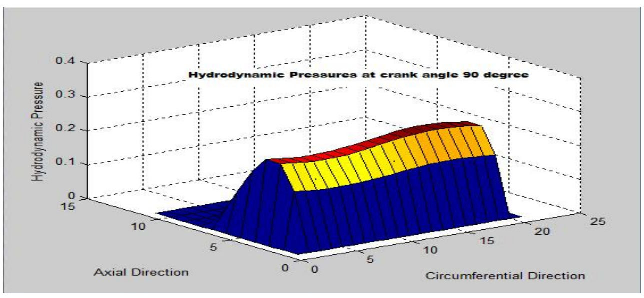

Fig:5 Hydrodynamic Pressure Field at 90º Crank Angle

Piston travel from mid point to BDC completes the intake stroke at 180 degree crank rotation. However, there is no appreciable change in the magnitudes of the hydrodynamic pressure fields. Even at 200 degree crank rotation when piston has already traveled some distance in the compression stroke and corresponding film thickness value is at the end point of first bell shaped curve (Refer to Fig:3), there is slight increase in the magnitude of hydrodynamic pressure fields, although profile shapes do not change, as shown in Fig:6.

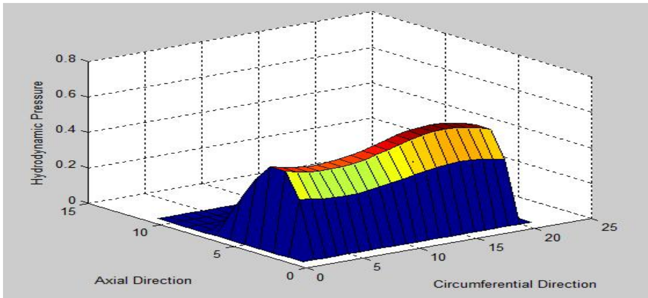

Fig:6 Hydrodynamic Pressure Field at 200º Crank Angle

In the compression stroke, the piston assembly pushes the air-fuel mixture upwards resulting in a swift reduction in the mixture volume due to the corresponding increase in the compression pressure. In the entire compression stroke, there are very significant changes in profile and the magnitude of pressure fields as well in their slopes. Just after piston's initial travel in the power and expansion stroke and slightly prior to actual combustion, at 370 degree of crank angle, the peak hydrodynamic pressures rise exponentially from a few Pascal to hundreds of thousands of Pascal and the slopes of pressure fields become very steep as shown in Fig: 7. There is a visible shift of pressures from most of the surface close to the origin. At this point, the minimum film thickness value is inching towards its lowest value (Refer to Fig:3).

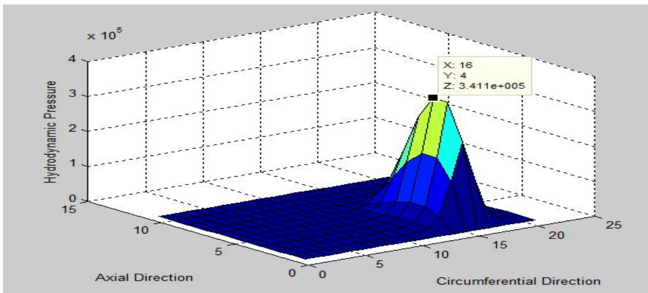

Fig: 7 Hydrodynamic Pressure Field at 370º Crank Angle

At 400 degree crank angle, when piston reaches close to one fourth of power/expansion stroke, there is slight reduction in magnitude of hydrodynamic pressures but peak pressure values remain in the range of hundreds of thousands of Pascal. As shown in Fig: 8, the pressure profiles and slopes follow the same trend as is witnessed at 370 degree crank angle. At this point, the hydrodynamic film thickness is minimum as shown in Fig:3, which in turn clearly highlights the relationship between hydrodynamic pressure rise and corresponding decrease in minimum film thickness. It implies that such sharp pressure rise just before and after combustion could cause damage to the top

ISBN: 978-988-18210-7-2
ISSN: 2078-0958 (Print); ISSN: 2078-0966 (Online)

compression ring and may increase the liner out of roundness as already minimized hydrodynamic film might fail to protect the top compression against adhesive wear.

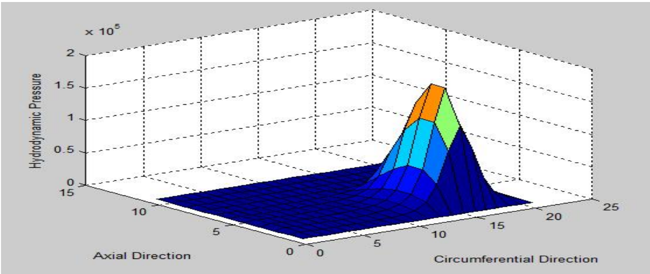

Fig: 8 Hydrodynamic Pressure Field at 400º Crank Angle

At 530 degree crank angle, when piston reaches near the end of expansion stroke, the hydrodynamic pressure fields return to their original profiles and their magnitudes reduce from hundreds of thousands of Pascal values to a fraction of the same as shown in Fig: 9. This corresponds to the vertex of the minimum film thickness in the second bell shaped curve (Refer to Fig: 3).

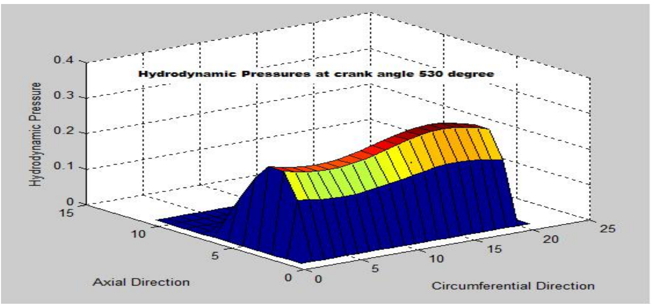

Fig: 9 Hydrodynamic Pressure Field at 530º Crank Angle

At 640 degree crank angle, when piston has crossed the midpoint of exhaust stroke, there is slight rise in magnitudes of hydrodynamic pressures as could be seen in Fig: 10. The pressures profile shape remains the same as is seen at 530 degree crank rotation. This corresponds to another dip of minimum film thickness at the terminal point of the second bell shape curve (Refer to Fig: 3).

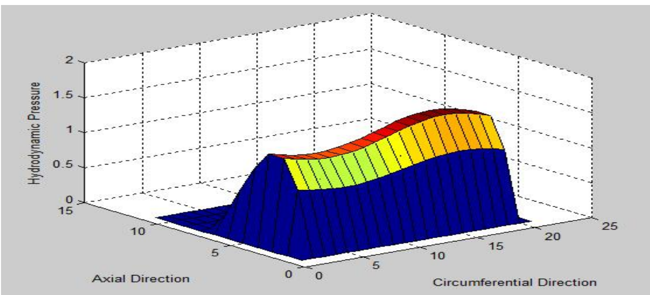

Fig: 10 Hydrodynamic Pressure Field at 640º Crank Angle

At 690 degree of crank rotation, film thickness value corresponds to the peak point of third curve, which is smaller than the other two bell shaped curves (Refer to Fig: 3). An increase in hydrodynamic film thickness results in simultaneous reduction of hydrodynamic pressures, as shown in Fig: 11. Although hydrodynamic pressure drop is witnessed but the overall magnitudes are far less and insufficient to be considered as real hydrodynamic pressures. So, in this region of piston stroke despite availability of lubricant film and flow sufficient pressures are not available to the protect top compression ring from damage or wear in the initial engine start up conditions.

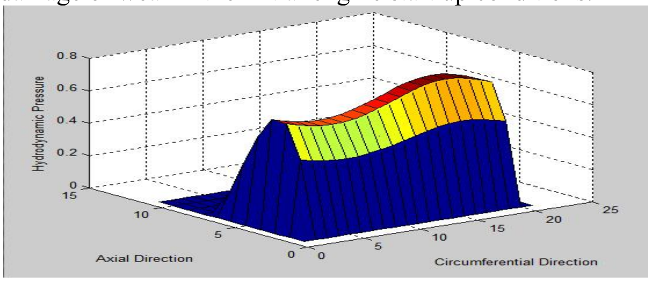

Fig: 11 Hydrodynamic Pressure Field at 690º Crank Angle

At the end of the exhaust stroke at 720 degree crank rotation, the engine cycle is complete. The piston is at TDC and has minimal velocity approaching zero. Due to inertia effect of flywheel, the piston is about to commence its new cycle. At this point, the hydrodynamic lubricating film thickness goes down to approach zero value due to minimal or no piston displacement. The hydrodynamic pressure profiles change shape as those witnessed at 370º and 400º, although magnitude of such pressure fields remain in the range as seen in the exhaust stroke.

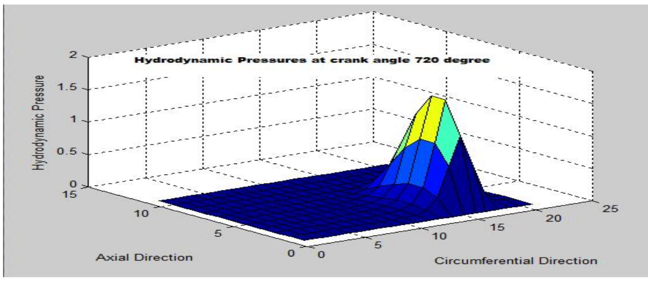

Fig: 12 Hydrodynamic Pressure Field at 720ºCrank Angle

V. CONCLUSION

In this paper, we have tried to address the question of the effect of bore out of roundness and how does it affect lubricating hydrodynamic film thickness and coefficient of friction in the initial engine startup. Numerical analysis was presented based on 2D Reynolds equation having fully flooded Reynolds exit condition. Despite the issues of stability and convergence, the finite differencing approach demonstrates a reasonable way to model elliptical bore effects on lubricating film thickness and friction force coefficient in engine initial startup. Numerical results show that hydrodynamic film and pressure profiles get affected due to non-circular bore. Furthermore, numerical simulations indicate that the minimum film thickness increases with non-circular bore. In conclusion, we have addressed and assessed some of the issues concerning two-dimensional hydrodynamic lubrication in initial engine start up, although further work is needed to extend this method to conduct parametric studies.

NOMENCLATURE

a1, a2 = geometrical parameters of ring face profile = 0.5 for perfectly parabolic face profile
b = ring axial width = 0.0015m
Cp = radial clearance = 0.00001m
FG = combustion gas force

h^* = dimensionless hydrodynamic film thickness
L = ring length
l = connecting rod length = 0.15m
p = dimensional hydrodynamic pressure
p^* = dimensionless hydrodynamic pressure
R = ring radius = 0.0415m
S1, S2 = geometrical parameters of ring face profile = 2e-6
r = crank radius = 0.0418m
U = piston velocity in axial direction.
V = engine speed [rpm] = 600 rpm
W = net resultant radial force
x^* = dimensionless axial coordinate
y^* = dimensionless circumferential coordinate
θ = crank angle
Δc = maximum bore distortion
ε = error term = 0.005

ACKNOWLEDGEMENTS

The authors are indebted to College of Electrical and Mechanical Engineering, School of Mechanical and Manufacturing Engineering, National University of Sciences and Technology, Pakistan for the completion of this research work. We also gratefully acknowledge the help provided by Higher Education Commission, Pakistan for this purpose.

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