Shear-Heating Effects on Piston Skirts Lubrication in the Initial Engine Start Up

M. AFZAAL MALIK, S ADNAN QASIM, MUMTAZ ALI KHAN, RIAZ MUFTI

Abstract. **A 2-D hydrodynamic piston skirts lubrication model during initial engine start up is developed by incorporating shear heating effects due to lubricant flow between piston skirt and liner surfaces. Numerical analysis is presented based on 2D thermal energy equation having adiabatic conduction and convective heat transfer with no source term effects. Viscous dissipation coupled with piston motion, pressure field generation, temperature effects on oil viscosity and subsequent oil film profiles in the contact region are examined and influence of shear heating on hydrodynamic film thickness at the time of engine initial start up are investigated. This study suggests that oil film temperature rise due to shear heating adversely affects lubricant film thickness and hydrodynamic pressures, which in turn affect Newtonian lubricant performance during initial engine start up conditions.**

Key words: **Oil viscosity, Hydrodynamic lubrication, shear heating, piston skirt, initial engine start up.**

I. Introduction

Piston lubrication phenomenon in an automobile engine is significant under normal operating conditions but attains real importance during initial engine start up conditions. It is due to its impact on overall engine design, performance, durability and life. In normal engine operating conditions, out of total friction loss in an engine, sliding motion of lubricated piston accounts for almost 45% of losses. These losses are expected to enhance at the time of engine start up. Important factors affecting friction and piston lubrication characteristics are lubricant shear heating, film thickness, viscosity, nature of contact and engine operating conditions. At the time of cold engine start up, heat generated due to combustion in first two to three cycles may be assumed to be removed before it actually affects piston skirt surface and hence, could be neglected. An increase in temperature due to shear heating at the time of engine start up reduces lubricant viscosity, which decreases film thickness, thus adversely affecting load carrying capacity of lubricant.

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The effects of oil viscosity on oil film thickness are very significant. Viscosity degradation due to temperature variations may ultimately result in engine failure [1]. The effects of temperature variations within the oil film, pressure and shear rate are all needed if a realistic friction and wear model is required for detailed analysis of lubrication characteristics [2]. Studies have shown that thermal effects on oil film thickness due to viscosity changes in thermohydrodynamic (THD) lubrication and Thermoelastohydrodynamic lubrication (TEHL) are more influential as compared to non- Newtonian effects of lubricant [3].

Most of the early researchers either neglected variation of oil viscosity with temperature by taking it constant or estimated it by considering constant oil film and Hertzian pressure field at normal engine operating conditions [4]. Some others incorporated viscosity variation under normal engine operating conditions by using liner or wall temperature, local temperature or average temperature [5, 6, 7]. Highlighting the effects of viscosity changes due to temperature rise on oil film thickness during normal engine conditions, recent studies incorporated the effects of lubricant viscosity on temperature and shear rate and oil film thickness for piston ring pack but did not consider engine start up conditions [8]. To consider initial engine start up conditions TEHL modeling of steadily loaded journal bearings for rapid and slow engine start up by showing viscosity variations with temperature was done but piston skirt lubrication was not considered [9].

Review of previous research related to thermal effects in piston skirt lubrication clearly indicate that no generic and simplified piston skirt lubrication model at the time of engine start up exits to cater for oil viscosity variations due to temperature rise by viscous heating and corresponding effect on piston secondary eccentricities, oil film thickness and hydrodynamic pressures. This study develops a simplified numerical piston skirts lubrication model using 2D transient thermal energy equation with heat generated due to viscous heating and analysis of its effects on oil viscosity and film thickness during engine initial start up conditions. In this regard following assumptions are made:-

a. Lubricant is incompressible.
b. Side leakage, oil starvation and surface roughness factors are neglected.
c. Lubricant is Newtonian fluid.
d. Specific heat, density and heat conductivity coefficient of lubricant in a cycle are constant.
e. Oil viscosity is function of oil temperature and shear rate only.

f. Lubrication is pure hydrodynamic with fully flooded inlet and Reynolds exit conditions.
g. Thermal effects due to combustion are neglected.

II. Mathematical Model

A. Basic Equations of Piston Motion

To define basic equations of piston motion its position, velocity and acceleration along axis of cylinder are determined by making these as function of crank angle. Small secondary piston skirt eccentricities (perpendicular to the axis of cylinder liner) are incorporated and calculated to define equations of motion. We calculate piston inertia, hydrodynamic force, hydrodynamic friction force and moments in the way similar to that defined by Zhu et al [10]:

$$\begin{bmatrix} m_{pis}\left(1-\frac{a}{L}\right)+m_{pis}\left(1-\frac{b}{L}\right) & m_{pis}\frac{a}{L}+m_{pis}\frac{b}{L} \\ \frac{I_{pis}}{L}+m_{pis}(a-b)\left(1-\frac{b}{L}\right) & m_{pis}(a-b)\frac{b}{L}-\frac{I_{pis}}{L} \end{bmatrix}\begin{bmatrix}\ddot{e}_t \\ \ddot{e}_b\end{bmatrix} = \begin{bmatrix} F+F_s+F_f\tan\emptyset \\ M+M_s+M_f \end{bmatrix}$$

(1)

B. Reynolds Equation and Hydrodynamic Pressures

In our mathematical model 2-D Reynolds equation is used to calculate hydrodynamic pressures and forces describing fluid flow in a wedge (gap) as suggested by G.W. Stachowiak et al [11]. In non-dimensional form it is given by:

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

(2)

Boundary conditions for Reynolds equation are [3]:

$$\frac{\partial p}{\partial x}_{x=0} = \frac{\partial p}{\partial x}_{x=\pi} = 0 \;;\; p = 0 \text{ when } x_1 \le x \le x_2$$

$$p(x,0) = p(x, L) = 0$$

(3)

C. Hydrodynamic Forces and Shear Stress

Hydrodynamic pressure, shear stress and hydrodynamic friction forces are determined using:-

$$F_h = R\iint_A p(x,y)\cos x\,dx\,dy \quad (4)$$

$$F_{fh} = R\iint_A \tau(x,y)\,dx\,dy$$

(5)

$$F_{fh} = R\iint_A\left(\eta\frac{U}{h}+\frac{h}{2}\frac{dp}{dy}\right)dx\,dy$$

(6)

The total normal force acting on piston skirt is given by

$$F_s = \tan\varphi(F_G + F_{IP} + F_{IC})$$

(7)

D. Oil Film Thickness Equation

Considering piston eccentricities, hydrodynamic film thickness is approximated by following expression

$$h = C + e_t(t)\cos x + [e_b(t) - e_t(t)]y \cos x$$

(8)

E. Thermal Energy Equation

The temperature rise in oil film is determined by using two dimensional transient thermal energy equation with heat generated from viscous dissipation(shear heating) as under[8]:-

$$\rho_f C_f\left(\frac{\partial T}{\partial t}+u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}\right) = K\left(\frac{\partial^2 T}{\partial^2 x}+\frac{\partial^2 T}{\partial^2 y}\right)+\emptyset \quad (9)$$

Where $\emptyset$ is viscous dissipation term given by:

$$\emptyset = \tau\frac{\partial u}{\partial y}$$

(10)

Thermal boundary conditions are as under:

$T = T_o$ at inflow on inlet side (x=0)

$\frac{\partial T}{\partial x} = 0$ at exit on inlet side ($x=\pi$)

$T = T_o$ at $y = 0$

$\frac{\partial T}{\partial y} = 0$ at $y = L$

F. Oil Film Mean Temperature.

Oil film mean temperature is estimated by the following relationship [8]:-

$$T_m = \frac{1}{hB}\int_0^h\int_0^B T(x,y)\,dx\,dy \quad (11)$$

G. Viscosity- Oil Film Temperature Relation

The temperature dependency of oil viscosity is estimated using Vogel equation [8] as under:-

$$\mu = a_0 \exp\left(\frac{T_1}{T_2+T_m}\right) \quad (12)$$

Table-1(Piston input parameters)

S/No	Parameter	Value	S/No	Parameter	Value
1	Piston Mass(m_{pis})	0.295 kg	6	a	0.0125m
2	Radius (r)	0.0415m	7	b	0.0015m
3	Length (L)	0.0338m	8	Cg	0.0002m
4	Pin Mass(m_{pin})	0.09kg	9	Poisson Ratio	0.3
5	Piston-pin off-set	0.001m	10	Young's modulus	200 GPa
6	Clearance(C)	0.00001 m			

III. Numerical Solution Procedure

Due to complex nature of problem, we solved it by using a numerical method. The Reynolds equation, oil film thickness equation and thermal energy equation were solved simultaneously using iterative numerical scheme. To maintain numerical stability central, backward/forward differencing were used. Flow chart to illustrate the computational procedure is as under:-

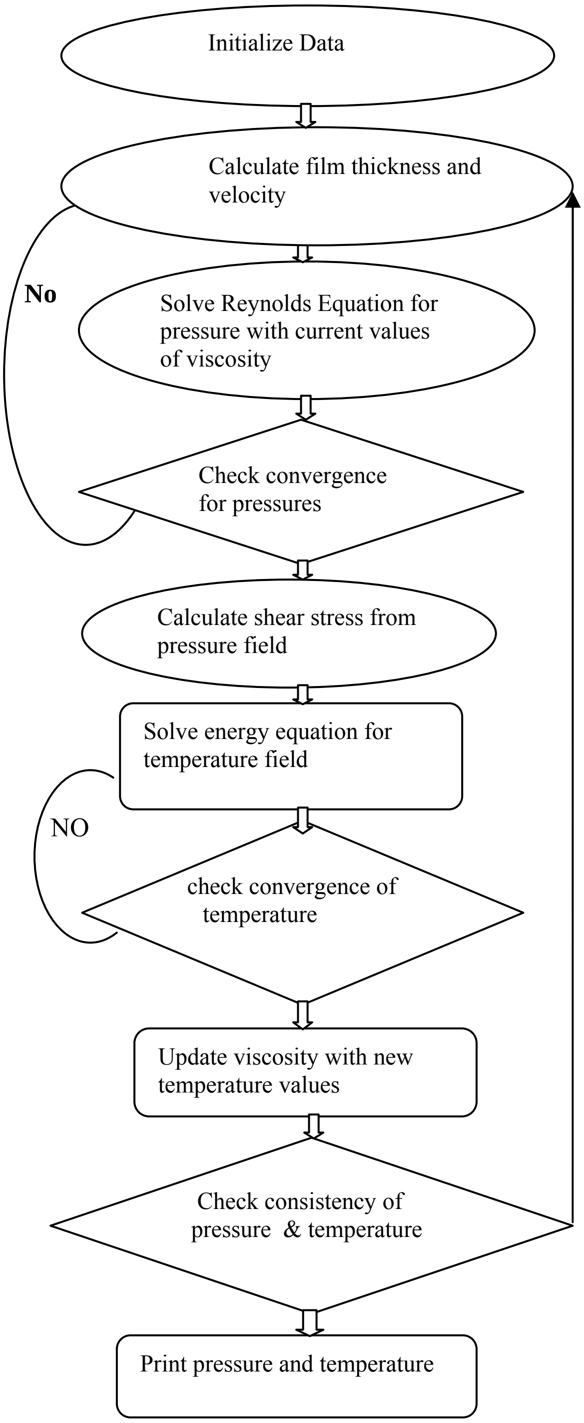


Fig:1 Flowchart of Computational Scheme

IV. NUMERICAL RESULTS AND DISCUSSION

A. *Piston Eccentricities*

Figure 2 shows dimensionless piston eccentricities (E_t & E_b) and crank angle with three horizontal lines that is, lower line is the touching line on the major thrust side, upper line is the touching line on minor thrust side and mid-line (zero line) indicates piston eccentricity to be zero. $E_t(t)$ represents piston skirt eccentricity at the top position and $E_b(t)$ is the eccentricity of piston skirt at the bottom. If the eccentricity curve goes beyond either upper line or the lower line then solid-to-solid contact establishes. It can be seen that 'E_b ' curve remains well away from both upper and lower lines throughout the entire four strokes of engine cycle. In contrast, 'E_t' curve has very significant variations in its amplitude as just after power stroke there is very significant phase shift such that during power stroke and beyond it goes very close to the lower line. Despite getting so close, it still does not touch the line, which means that for an ultra fine surface and neglecting surface friction effects actual physical contact can still be avoided. This finding is very significant as it implies that despite shear heating effects adversely affecting viscosity of the lubricant thereby reducing its film thickness and load carrying capability, possibility of adhesive wear between piston skirt and cylinder liner can still be avoided under assumed conditions. This profile is disturbed piston eccentricities profile as compared to the isothermal case of modeling piston skirts lubrication in initial engine start up conditions [12].

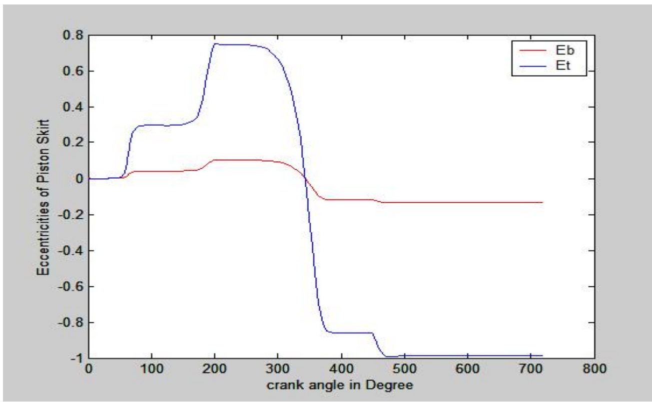


Fig:2 Dimensionless Piston Skirt Eccentricities

B. *Effect of Minimum Film Thickness*

Time based maximum & minimum lubricant film thickness established between opposing surfaces of piston skirts and cylinder wall is determined as shown in figure 3. Intake stroke has negligible film thickness. This time period becomes very critical as it corresponds to the piston eccentricities towards minor thrust side. In the compression stroke oil film thickness rises and attains its vertex near the end of piston stroke. This means that more the film thickness value increases, more are the chances that piston eccentricities will get reduced. Hence, even lesser chances for any solid-to-solid contacts to take place. At the start of piston expansion stroke, exponential rise in pressures due to combustion causes combustion gas force to deliver real thrust to the piston thereby severely affecting oil film thickness between piston skirts and cylinder liner.

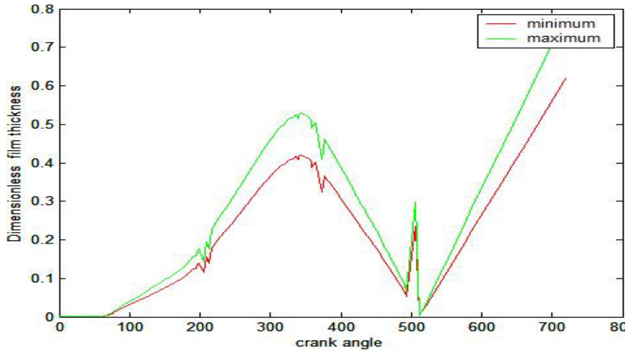


Fig: 3 Maximum & Minimum Film Thickness Profiles

C. Discussion on Temperature Variations

Modeling initial engine start up conditions by incorporating shear heating effects involve adiabatic heat generation and flow. This results in temperature rise from inlet temperature, which, in our case is assumed to be same as ambient temperature, that is 40^0C (40+273=313K). Figure 4 shows instantaneous temperature rise from initial value to clearly indicate variations at every node of the mesh. This nearly unify initial temperatures rise to values around 316K, followed by spread out temperature profiles across the oil film such that minimum and maximum temperature variations range between 315K and 319K. This temperature variation is quite significant especially in the context of initial engine start up conditions as it is assumed that at the time of engine start up heat generated due to combustion is instantaneously removed from the system before it actually reaches the piston skirts and enhances internal film temperatures. The temperature rise due to shear heating, even by few degrees K is undesirable as it adversely affects lubricant viscosity and reduces it to an extent that oil film thickness gets reduced compared to the case when these shear heating effects are neglected [12]. Reduction in oil film thickness decreases load carrying capacity of lubricant.

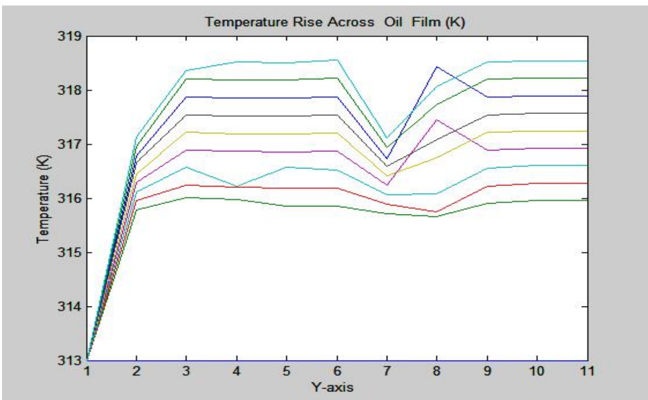


Fig: 4 Temperature Rise from Ambient Conditions

The negative effects of reduced load carrying capacity of engine lubricant include simultaneous increase in magnitude of piston eccentricities (Refer to Figure 2) and increased probability of adhesive wear of piston skirts due to establishment of solid-to-solid contact between opposing surfaces in relative motion.

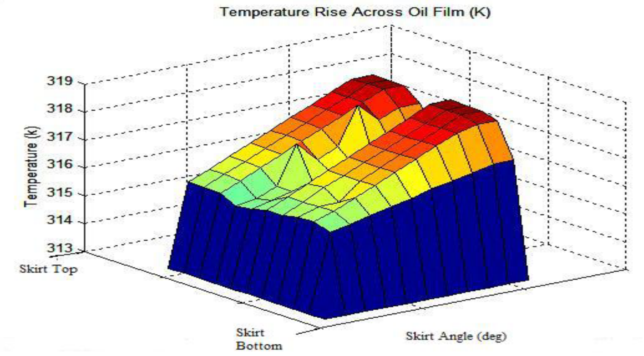


Fig 5: Temperature Variations over Skirt Surface

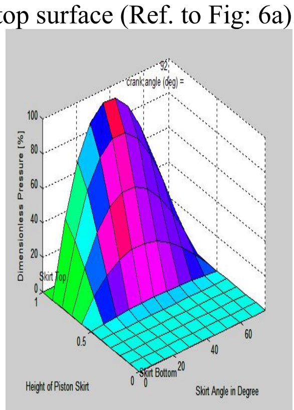
Figure 5 shows temperature field over the entire skirt surface area. Had it been an isothermal case then there would have been no temperature rise over the piston skirts surface. The temperature profiles clearly indicate surface points adjacent to extreme left corners of skirt top and bottom experience least temperature variations. In comparison, points (skirt top and near bottom) from at the mid surface till at three quarters from left experience significant temperature rise, which means that these portions of piston skirts are more vulnerable due to reduced viscosities and load carrying capacity of lubricant. Hence, more chances of skirts wear at those places.

D. Discussion on Hydrodynamic Pressure Fields

For our basic model, results of eight pressure fields at engine start up speed of 600 revolutions per minute (RPM) are shown & discussed below.

In the intake stroke as piston travels down hydrodynamic pressures start developing over skirt surface. Peak hydrodynamic pressures when fully developed have their values near piston skirt top surface showing gentle slope of instantaneous pressure fields.

At 90 degree crank angle when piston reaches at the mid of intake stroke, positive hydrodynamic pressures rise from mid point to the skirt top surface. The slope rises gently and peak pressure values are found at the centre point of skirt top surface (Ref. to Fig: 6a).



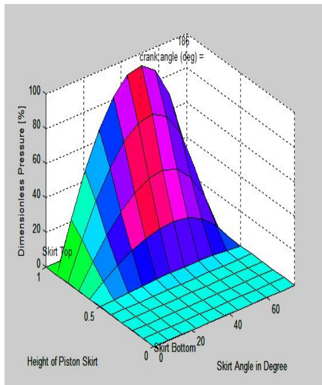


Fig : 6a Crank angle 90^0 Fig: 6b Crank angle 180^0

Piston travel from mid point to bottom dead centre in the intake stroke clearly indicate shift in peak pressures from midpoint of skirt top surface to the right. At 180 degree crank angle when piston is at the end of intake stroke, positive hydrodynamic pressures are seen rising from mid point to the skirt top surface but their slope is steep and peak

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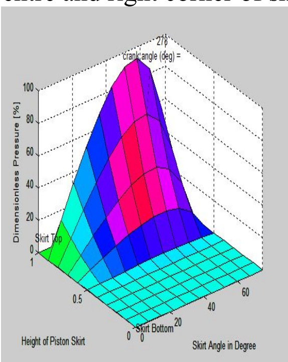
pressure values are seen to shift away from centre point of skirt top surface and towards right side (Refer to Fig: 6b).

In the compression stroke, piston pushes the air-fuel mixture upwards resulting in swift reduction in mixture volume due to corresponding increase in compression pressure. With the piston stroke travel initially pressures rise sharply than before resulting in steeper slopes before peak pressure values are attained. Further steepness in slope results in shifting of peak pressures from skirt top to skirt bottom surface. Peak hydrodynamic pressures initially generate at the piston skirt top surface but later shift towards skirt bottom surface showing less gentle slope of instantaneous pressure fields.

Piston reaches the mid of compression stroke at 270 degree crank angle. Positive hydrodynamic pressures rise very sharply generating far steep slopes than before. There is very visible shift of peak hydrodynamic pressures from the centre of skirt top surface to the right but the slope remains steep and peak pressure values are found at the mid of centre and right corner of skirt top surface (Refer to Fig: 6c).



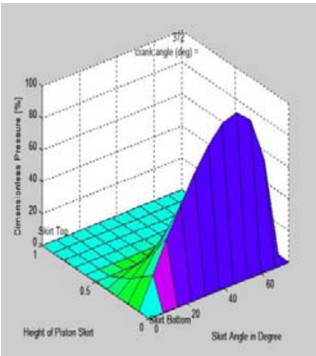
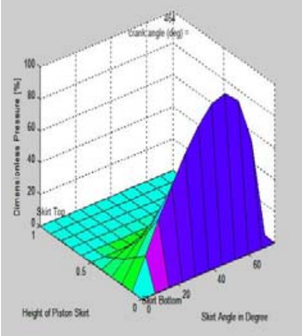


Fig: 6c Crank Angle 270^0 Fig: 6d Crank Angle 360^0

By the time piston reaches at the end of compression stroke that is, at 360 degree crank angle there is clear shift of positive hydrodynamic pressures to the skirt bottom surface such that these rise from mid point to the skirt bottom surface, the slope is steep but higher peak pressure values are close to the centre point of skirt bottom surface (Refer to Fig:6d). In the power stroke peak hydrodynamic pressures keep generating at the piston skirt bottom surface throughout the entire length of stroke showing less gentle slopes of instantaneous pressure fields. This shift is understandable as there is drastic directional shift of piston secondary motion coupled with exponential rise and subsequent fall of combustion gas pressures along with simultaneous increase in swept volume.

At 450 degree crank angle when piston is at the mid of power/expansion stroke, positive hydrodynamic pressures rise from mid point to the skirt bottom surface but the slope gets gentle and higher peak pressure values are found close to the centre point of skirt bottom surface (Refer to Fig: 6e).



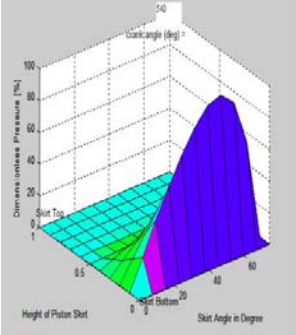
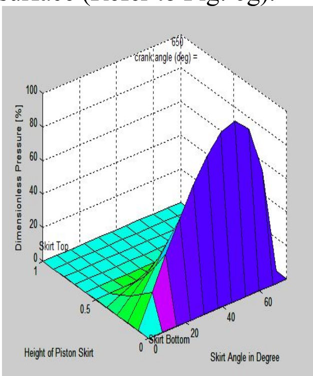


Fig:6e Crank Angle 450^0 Fig:6f Crank Angle 540^0

At 540 degree crank angle when piston is at the end of power/expansion stroke, positive hydrodynamic pressure profiles at the skirt bottom surface have gentler slope but higher peak pressure values again shift from the centre point towards the right of skirt bottom surface (Refer to Fig: 6f).

In the exhaust stroke peak hydrodynamic pressures are generated at the piston skirt bottom surface throughout the entire length of stroke showing gentle slopes of reduced instantaneous peak pressures.

At 630 degree crank angle when piston is at the mid of exhaust stroke, positive hydrodynamic pressures rise from midpoint of skirt to the skirt bottom surface but the slope becomes gentle, peak pressure values are found to have reduced and shift close to the centre point of skirt bottom surface (Refer to Fig: 6g).



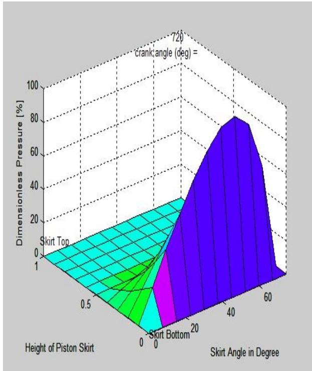


Fig:6g Crank Angle 630^0 Fig:6h Crank Angle 720^0

At 720 degree crank angle when piston reaches at the end of exhaust stroke, positive hydrodynamic pressure profiles at the skirt bottom surface maintain their gentler slope but at visibly reduced peak pressure values, which again shift slightly from the centre point towards the right of skirt bottom surface (Refer to Fig: 6h).

V. Conclusion

In this paper, we have addressed the question of shear heating of a Newtonian fluid and how does it affect lubricant characteristics of Newtonian oil in hydrodynamic thin film flows in initial engine startup. Numerical analysis was presented based on 2D thermal energy equation having adiabatic conduction and convective heat transfer with no source term effects. Despite the issues of stability and convergence, finite differencing approach demonstrates a reasonable way to model shear heating effects of lubricant in engine initial startup. Numerical results show that piston eccentricities and hydrodynamic pressure profiles get affected due to shear heating. Furthermore, numerical simulations indicate that the minimum film thickness gets sufficiently small with rise in temperature due to shear heating. In conclusion, we have addressed and assessed some of the issues concerning shear heating effects on the lubricant in engine initial startup, although further work is needed to extend this method to conduct parametric studies.

Nomenclature

a= Vertical distance from top piston skirt to piston pin.
b= Vertical distance from top piston skirt to piston centre of gravity
C= Radial clearance between piston and liner
C_f= Specific heat of oil

P= Hydrodynamic pressure
T = Oil film temperature
T_o = Ambient(inlet) oil temperature (40 K)
T_m = Mean oil film temperature
u= Piston entraining (sliding) velocity along x-direction
v= Piston velocity across oil film along y-direction
$\frac{\partial U}{\partial y}$ = Velocity gradient across oil film
μ = Oil viscosity (0.03187 Pas)
k = Oil thermal conductivity (0.13 W/mK)
ρ_f = Oil density (870 kg/m^3)
a_0 = Vogel equation parameter (0.06782mPas)
T_1 = Vogel equation parameter (1153 K)
T_2 = Vogel equation parameter (376 K)

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