A Critical Review of Piston Motion Noise in Combustion Engines

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Abstract: *Piston skirt contact is caused as the force in connecting rod changes its direction resulting in slapping noise. This is a major source of noise and also leads to wear of liners [1]. Major factors that affect piston slap include [1]:*
a) Cylinder Bore Temperature
b) Lubrication Oil Film Thickness
c) Oil Viscosity
d) Engine speed
e) Skirt Profile
f) Skirt Roughness
g) Skirt Waviness
i) Skirt Size
j) Wrist pin offset
k) Piston-Liner gap.
Piston slap takes occurs mainly near top dead center (TDC) positions.
Motion of crankshaft picks up the lubrication oil from sump. This oil is then transported along cylinder bore due to motion of piston skirt, piston rings and gravity. Oil is consumed either inside the combustion chamber or it returns back to sump, or is blown away by residual gases.
This work presents secondary motion of skirt considering tribological facts.

Keywords: *Piston secondary motion, liner system*

1. Introduction

According to Stribeck curve, various lubrication zones may be classified into following three major types: boundary zone, hydrodynamic zone and mixed zone [2]. In the boundary lubrication zone, the asperities in mating parts come into contact, whereas in hydrodynamic zone there is no direct contact and hence the film of lubricant separates mating surfaces. The function of piston rings of skirt assembly is to seal pressure inside the combustion chamber and prevent leakages of oil from crankcase into combustion chamber. Type of lubrication zone changes with variations in operational conditions of engine. As piston reaches towards dead center positions, its speed approaches zero and hence boundary lubrication zone dominates. At mid strokes, where piston speed is at its maximum values, the hydrodynamic zone dominates as seen in figure no 1.

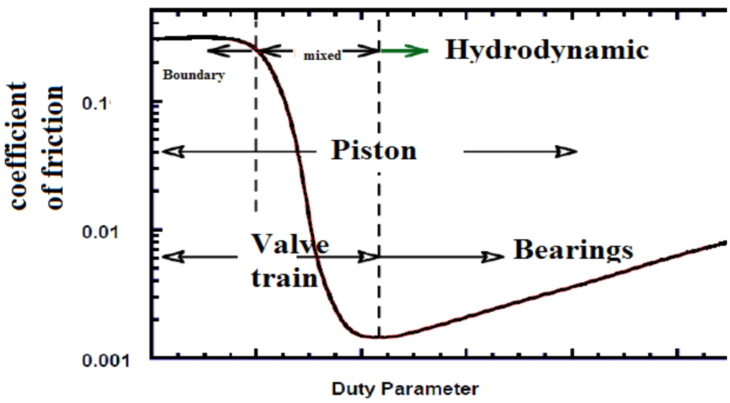

Fig. 1. Stribeck lubrication curve [1]

2. Background

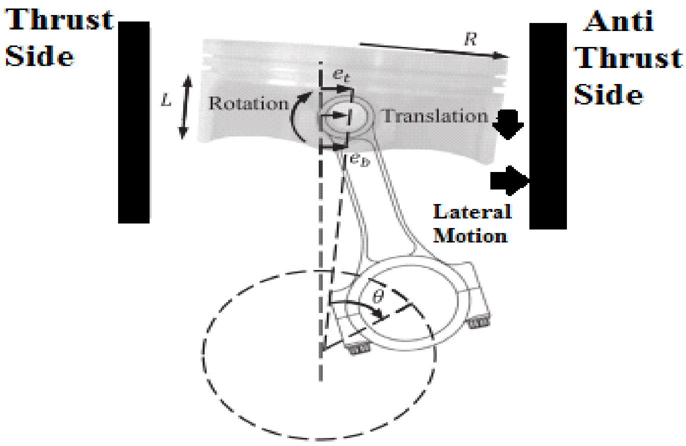

Fig. 2. Piston secondary motion [2]

Increasing demand for noise, vibration and harness comfort levels have led to detailed study of dynamic motion of skirt as skirt-liner contact plays an important role in various frictional losses occurring in engines [2]. Piston dynamics is a key concept to understand lateral motion of skirt as depicted in figure no 2 [3]. Piston may impact either on thrust side (TS) or anti-thrust side (ATS) of liner. These contacting motions cause vibrations in liner which are further transmitted from various surfaces of engine. In general, three major approaches have been identified to locate instances of skirt-liner contact during this motion [3]. First one includes study of piston secondary motion without taking into consideration lubrication effects of oil and rotatory motion of skirt. This approach is known as static method. In second one, piston side force is analyzed by taking into lubrication pressure distribution as represented by Reynolds equations [3].

Oil film thickness is the third parameter to analyze the lateral motion of skirt. Skirt -liner contact may occur if this thickness is minimum which may occur either towards thrust or anti thrust side [4]. As the slope of film thickness changes, the squeezing action is initiated indicating a possible instance of piston slap. Other methods to study slapping motion includes energy transfer method and angular duration method [4]. Instances at which the maximum energy is transferred to liner wall can indicate a possible location of skirt-liner contact [4]. The angular duration method includes study of crank angle duration starting from initiation of squeezing action of oil film and terminating at occurrence of minimum oil thickness [5]. Up to 6-10 instances of actual skirt-liner contacts have been practically observed [5]. In order to validate various possible instances of slap, block vibration data from accelerometers mounted at various locations on engine block has been analyzed [6]. However, this data may include contributions due to other sources of noise such as combustion-based noise [6]. Pruvost has used spectro filters to separate the above-mentioned sources [7]. Liu and Randall used blind source separation (BSS) algorithm to achieve effective source separation [8]. Chen analyzed the concept of pseudo angular acceleration to study phase and frequency variations of slapping noise of skirt [9].

3. Reynolds equation for lubrication oil pressure distribution

The tribology of lubricating oil plays an important role in mechanical losses occurring in assembly of piston. About 3-5% of the total energy losses take place in the piston skirt assembly [10]. Figure no 3 shows a typical breakdown of various losses for a typical diesel engine, wherein it is clear from the figure that share of piston assembly accounts for about 20% -30% [11, 12].

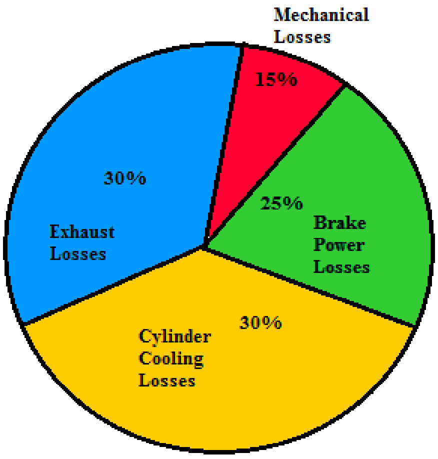

Fig. 3. Break up of total dissipation of fuel energy [10]

In 1886, Osborne Reynolds studied hydrodynamic pressure generated between two sliding surfaces. For an incompressible fluid with constant density, he proposed the Reynolds equation given by [13]:

$$\frac{\delta}{\delta x}\left(\frac{h^3}{12\eta}\frac{\delta P}{\delta x}\right)+\frac{\delta}{\delta z}\left(\frac{h^3}{12\eta}\frac{\delta P}{\delta z}\right)=\frac{1}{2}\frac{\delta(U_2-U_1)h}{\delta x}+(V_2-V_1)+\frac{1}{2}\frac{\delta(W_2-W_1)h}{\delta z} \quad (1)$$

In this relationship, the left-hand side terms are called pressure terms, whereas the right-hand side terms are known as source terms. The terms of $\frac{\delta U}{\delta x}\&\frac{\delta U}{\delta z}$ depict stretching action, whereas $\frac{\delta h}{\delta x},\frac{\delta h}{\delta z}$ depicts the wedging action. The velocity difference term (V_1 - V_2) is known as squeezing action as shown in figure no 4 [13].

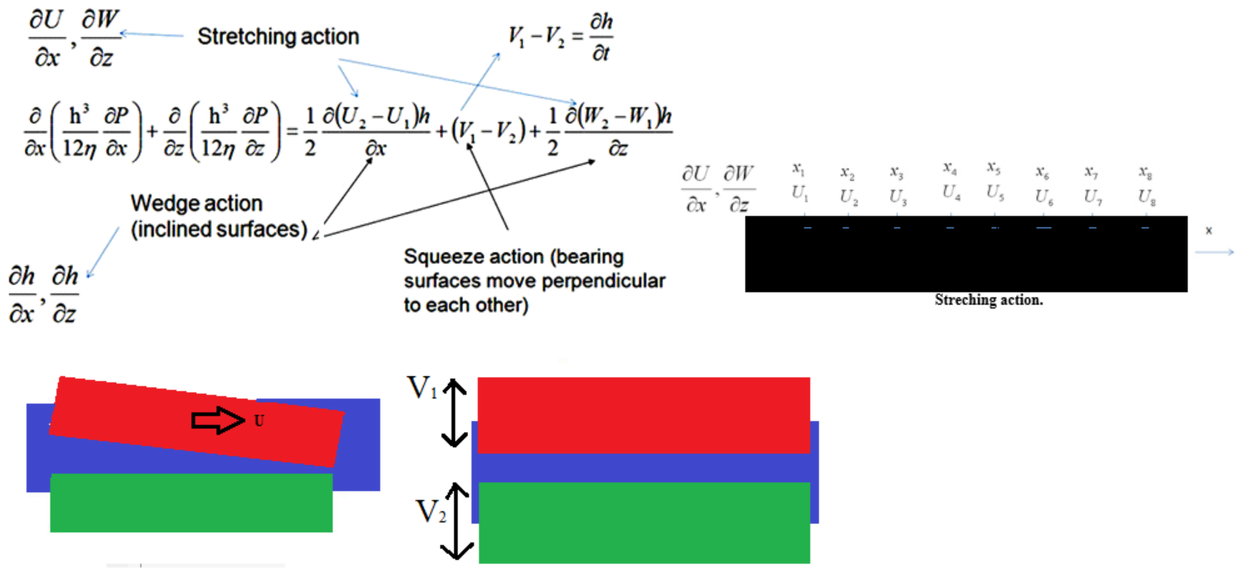

Fig. 4. Interpretation of terms in Reynolds equation [14]

Assuming that the lubrication oil used is a Newtonian fluid, flow is incompressible, value of viscosity is constant and neglecting various effects e.g., inertial forces, slip, angle of inclination, pressure gradient and stretching action, the Reynolds equation may be simplified as:

$$\frac{\delta}{\delta x}\left(h^3\frac{\delta p}{\delta x}\right)+\frac{\delta}{\delta z}\left(h^3\frac{\delta p}{\delta z}\right)=6\eta\,\frac{\delta(U_2-U_1)h}{\delta x}+12\eta\,\frac{\delta h}{\delta z} \quad (2)$$

In order to estimate the oil pressure distribution, this equation needs to be solved. One way to do this, is by considering pressure variations in one direction as shown in figure no 5. The piston may be assumed to be as a kind of short bearing and circumferential pressure gradient was neglected as compared to axial one. Using these assumptions, the Reynolds equation discussed above gets modified as:

$$[\frac{\delta}{\delta z}\left(h^3 \frac{\delta p}{\delta z}\right)] = 6\eta \frac{\delta h}{\delta x} \quad (3)$$

Using boundary conditions of P (θ,z= $\pm\frac{L}{2}$) = 0,the closed form of pressure distribution P can be expressed as:

$$P= \frac{-3\eta\omega}{c^2} \left(x^2 - \frac{L^2}{4}\right) \frac{\xi \sin\theta}{(1+\xi\ cos\theta\ )^3} \quad (4)$$

Where c is skirt-liner gap and ξ is skirt-liner eccentricity.
The density of lubricant is dependent on the generated oil pressure (P) and density at mean liner temperature (ρ_0) as expressed from following equation [6]:

$$\rho = \rho_0[1+\frac{0.6\times10^{-9}\times P}{1+1.7\times10^{-9}\times P}] \quad (5)$$

Table no F of presents values of various coefficients for different grades of SAE oils. The oil used for lubrication in present study was of SAE30W grade type for which viscosity may be expressed as [15]:

$$\eta = 0.1531*9.8 * 10^4 * a * e^{\left(\frac{b}{c}\right)} \quad (6)$$

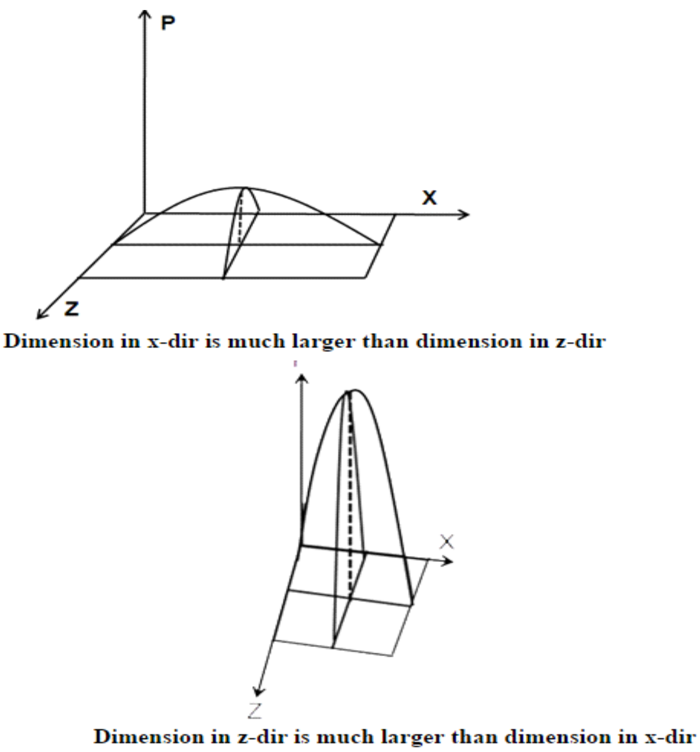

Fig. 5. Variation of pressure along one direction [14]

Since pressure terms have magnitude of order MPa and the oil film thickness are in microns, there may be some inconsistencies while solving the Reynolds equation.

This may be sorted out by considering the Nondimensionalization in which space coordinates may be written as:

$$\left.\begin{array}{l} \bar{x} = \frac{x}{X} \\ \bar{y} = \frac{y}{Y} \\ \bar{z} = \frac{z}{Z} \\ \bar{h} = \frac{h}{C} (5.7) \\ \bar{p} = \frac{pC^3}{6\eta U X^2} \\ \bar{t} = \frac{tU}{C} \end{array}\right\}$$

Substituting these non-dimensional values, the new equation gets modified as:

$$\frac{\delta}{\delta \bar{x}}\left(\bar{h}^{3}\cdot\frac{\delta \bar{p}}{\delta \bar{x}}\right) + \frac{X^2}{Z^2}\frac{\delta}{\delta \bar{z}}\left(\bar{h}^{3}\cdot\frac{\delta \bar{p}}{\delta \bar{z}}\right) = \frac{C}{X}\frac{\delta \bar{h}}{\delta \bar{x}} \tag{8}$$

In order to solve this equation, finite element analysis (FEA) method was used for which the mating surfaces were analyzed into number of nodes as shown in figure no 6 [14]. A mesh was made so that nodes on the lubrication zone of skirt correlates with those used in finite element analysis (FEA) in order to analyze the pressure distribution of lubricating oil.

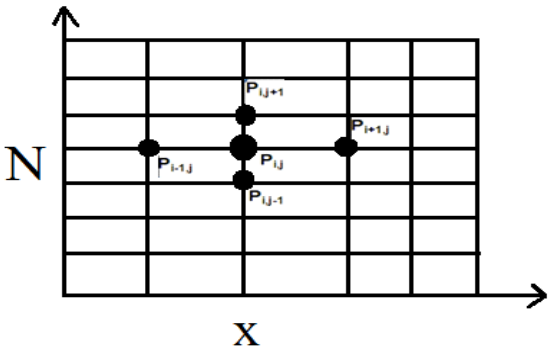

Fig. 6. Nodal representation of surface [14]

Various gradient terms of Reynolds relationship can be solved using Taylors approximation which yields following results [14]:

$$\bar{h}^{3}\cdot\frac{\delta \bar{p}}{\delta \bar{x}} = \frac{\bar{h}_{i,j+0.5}^{\ 3}\ \bar{p}_{i,j+1} + \bar{h}_{i,j-0.5}^{\ 3}\ \bar{p}_{i-1,j} - (\bar{h}_{i,j+0.5}^{\ 3} + \bar{h}_{i,j-0.5}^{\ 3})\bar{p}_{i,j}}{\Delta \bar{x}^{2}} \tag{9}$$

$$\bar{h}^{3}\cdot\frac{\delta \bar{p}}{\delta \bar{z}} = \frac{\bar{h}_{i,j+0.5}^{\ 3}\ \bar{p}_{i,j+1} + \bar{h}_{i,j-0.5}^{\ 3}\ \bar{p}_{i-1,j} - (\bar{h}_{i,j+0.5}^{\ 3} + \bar{h}_{i,j-0.5}^{\ 3})\bar{p}_{i,j}}{\Delta \bar{z}^{2}} \tag{10}$$

$$\frac{\delta \bar{h}}{\delta \bar{x}} = \frac{\bar{h}_{i+1,j} - \bar{h}_{i-1,j}}{2\,\Delta \bar{x}} \tag{11}$$

Substituting these relationships and rearranging them we have:

$$P_{i,j} = A_{i,j}\ P_{i,j+1} + B_{i,j}\ P_{i,j-1} + C_{I,j}\ \ P_{i+1,j} + D_{i,j}\ p_{i-1,j} + E_{i,j} \tag{12}$$

Most of values of nodal pressure ($P_{i,j}$) are unknown so an iterative loop must be used with suitable convergence limits (£) to get values of pressure. i.e.

$$\frac{(\sum_{i=1}^{n}\sum_{j=1}^{m} \bar{p}_{i,j})_{iteration\ k} - (\sum_{i=1}^{n}\sum_{j=1}^{m} \bar{p}_{i,j})_{iteration\ k-1}}{(\sum_{i=1}^{n}\sum_{j=1}^{m} \bar{p}_{i,j})_{iteration\ k}} \leq £ \tag{13}$$

4. Results and discussions

A MATLAB code was used to analyze the lubrication behavior of oil between piston skirt and liner considering its motion of skirt analogous to that of a journal inside the bearing [14].
Plots of oil pressure distribution on the piston skirt plane were next analyzed for each 90° crank angle rotation of skirt in case of a single cylinder of engine for a compete cycle as shown in figure no 7-10. Rotation operating speed of engine was 2000 RPM with nominal skirt-liner gap of 0.05mm.

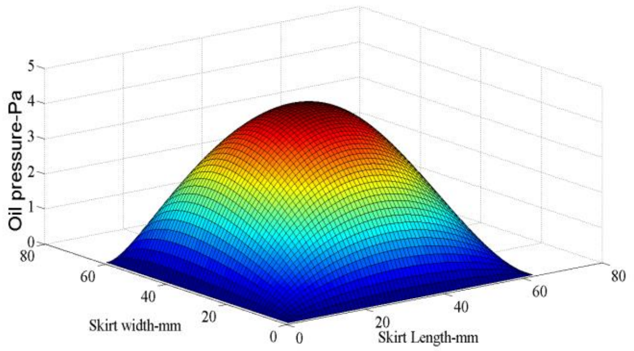

Fig. 7. Oil pressure distribution (180° crank angle)

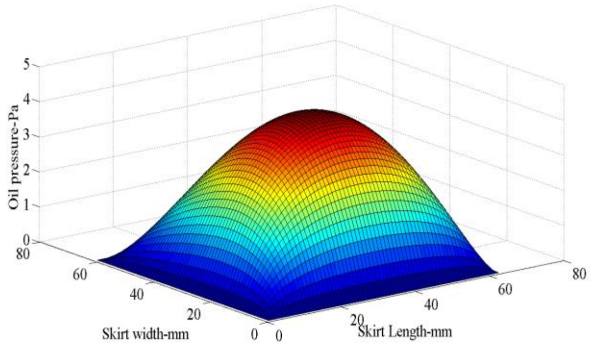

Fig. 8. Oil pressure distribution (360° crank angle)

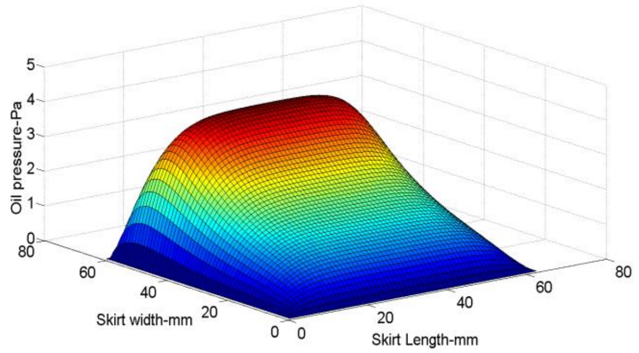

Fig. 9. Oil pressure distribution (540°crank angle)

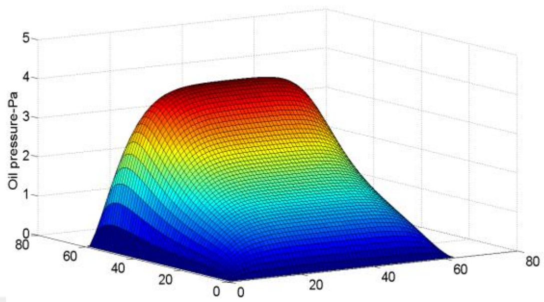

Fig. 10. Oil pressure distribution (720°crank angle)

At 180° crank angle towards the end of intake stroke, peak pressures were observed towards the midpoint of skirt. At 360° crank angle position, as the end of compression stroke approaches, the slopes of pressure curves starts shifting slightly towards right hand side. At 540° crank angle, the peak values of oil pressures were seen to shift towards the bottom part of skirt. During the exhaust stroke peak of slopes shifted slightly towards right side. There is a lag between positions corresponding to peak cylinder pressure developed and peak hydro dynamic oil pressure developed.

The energy transferred to liner due to impacts may be analyzed by taking various grid points on the liner. Energy at each point ($W_{I,J}$) can be calculated using average of local force ($\tilde{F}_{I,J}$) between two time steps as [12]:

$$W_{I,J} = \tilde{F}_{I,J} \left(h_{2I,J} - h_{1I,J} \right) \quad (14)$$

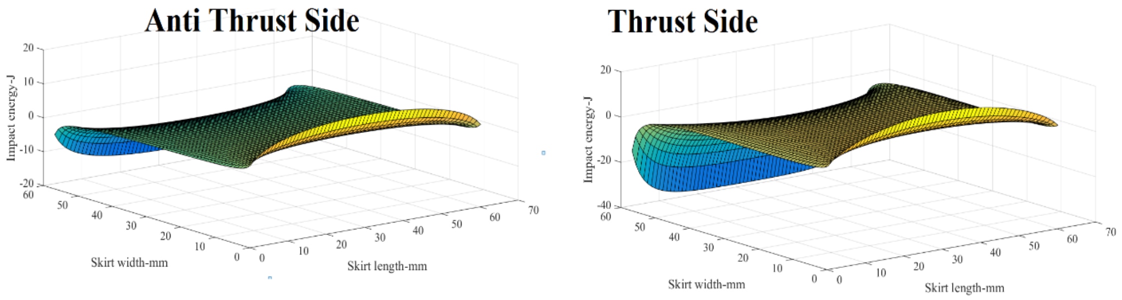

Fig. 11. Impact Energy (720°crank angle)

Figure no 11 shows variations in the impact energy transferred to liner walls at 2000 RPM at 720˚ crank angle position. It is clear that energy is transferred both on thrust as well as anti -thrust sides at the same time. When value of $W_{I,J}$ is positive, lubricant is squeezed and oil film absorbs the impact energy. Otherwise, this energy is utilized in skirt deformation.
The acoustic power available at surface of engine (P_a) may be expressed in terms of radiation efficiency (σ), density of air (ρ_a), wave speed(c_a), area of noise radiating surface (A_r)and surface velocity(v_r) as [3]:

$$P_a = \sigma^*\rho_a{}^*c_a{}^*v_r{}^2\, A_r \tag{15}$$

The impact power (P_v) can be expressed in terms of impedance (Z) and impact velocity V_v by the following relationship:

$$P_v = ZV_v{}^2 \tag{16}$$

Hence overall transmission efficiency (η_t) can be written in terms of these responses and expressed as:

$$\eta_t = \frac{\mathrm{Pa}}{\mathrm{Pv}} = \frac{\sigma V^2{}_r \rho_a C_a A_r}{Z\, V_v{}^2} \tag{17}$$

Using law of conservation of energy, it may be assumed that impacting energy is transmitted without any loss to outer surface of skirt and may be expressed in terms of thickness of structure (h), area (A), velocity (V) and density (ρ). i.e. [12]

$$\frac{1}{2}\rho_v h_v A_v V_v{}^2 = \frac{1}{2}\rho_R h_R A_R V_R{}^2 \tag{18}$$

Where subscript v denotes liner structure and R denotes engine block.

Hence the transmission efficiency at each node (i,j) at both thrust and anti -thrust side may be expressed by [12]:

$$\eta_t(i,j) = \frac{\sigma\rho_v\rho_a h_V C_a A_v(i,j)}{Z_{i,j}\rho_R h_R} \tag{19}$$

Where $Z(i,j) = \frac{F_v(i,j)}{V_v(i,j)}$

The SPL at each node can be estimated taking reference pressure level P_{ref} ($2*10^{-6}$ Pa) and expressed in terms of distance of microphone from engine block (R) as [12-20]:

$$SPL_{\,i,j} = L_{\,i,j} + 10\log\left(\frac{S^-}{4\pi R^2}\right) \tag{20}$$

Where $L_{\,i,j} = 10\log\left(\frac{P_a\, i,j}{P_{ref}}\right)$

10. Conclusions

Piston slapping motion is a major cause of noise and vibrations in engines. The dynamic equations of piston secondary motion may be solved to see effects of various skirt design issues.

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