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Study on the Clearance Design for Low Side Impacts of Engine Piston

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Clearance movements of engine piston are regarded very important because they cause impact vibrations as well as many tribological problems. Some of the major parameters that influence these kinds of performances are piston profiles, piston offsets and clearance magnitudes. In our study, computational investigation is performed about the piston movements in the clearance between piston and cylinder liner by changing the skirt profiles and piston offsets. Our results show that curved profile and more offset to thrust side have better performance with low side impact during the engine cycle.
Keywords: clearance movement, skirt profile, impact vibration

1. INTRODUCTION

Clearance design is very important to many mechanical elements of moving parts in many aspects. Among the many moving mechanical components in the clearance, the performance and endurance of piston depend strongly on the skirt profile. In general, tight clearance design makes tribological problems with low noise and vibration, while loose clearance design have less friction and wear with higher side impact of piston to cylinder block. Because piston movement is under high pressure and sudden change of velocity, the clearance design of skirt profile influences the tribological problems and side impacts on the engine.

In order to investigate the movement of piston in the clearance between piston and cylinder block, it is necessary to verify the fluid film behaviors of hydrodynamic lubrication. Piston movements are traced in the clearance by equating the applied forces from combustion pressure and many other inertia forces with fluid film pressure. In our work, the fluid film pressures are computed with three kinds of piston movements such as sliding, rolling and translation with Reynold's boundary condition during the engine cycle.

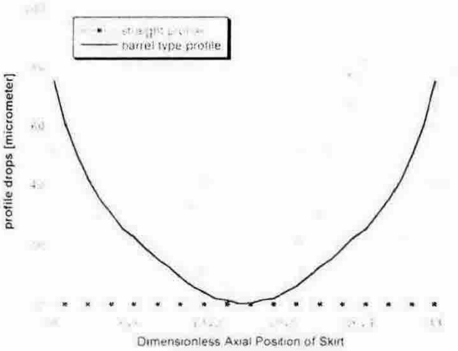


Figure 1 Profiles of straight and barrel types

2. PISTON SKIRT PROFILE OF FLAT AND BARELL TYPES

Better lubricant film of hydrodynamic lubrication exists when the sliding direction is to the narrower space of gap. Straight profile (flat type) can have lubrication film on the narrower gap of sliding direction (*i.e.* thrust side). However, on the other side piston have wider gap of sliding direction (*i.e.* anti-thrust side), so the piston is not supported by the fluid film pressure and sometimes direct impact on the cylinder block occurs with large vibration, so called piston slap. When the skirt profile has curved feature of barrel type, it can have supports on both sides of piston skirts such as thrust and anti thrust sides. Figure 1 shows the profile difference between straight and barrel shapes at 90 degree of piston pin assembly position and Figure 2 shows surface shapes of barrel type skirt profile.

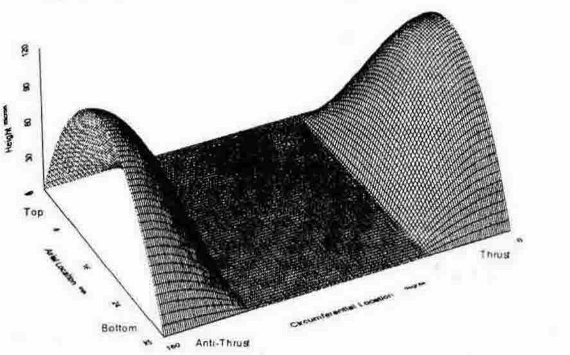


Figure 2 Skirt shape of barrel type

3. PISTON MOVEMENTS IN THE CLEARANCE

The force equilibrium between side force of piston and fluid film pressure is computed at each time step (720 steps) during engine cycle by tracing the upper and lower positions of piston as it is done by Dursunkaya [1]. The forces on the piston come form combustion pressure, acceleration of piston components, fluid film pressures, (Eq.1) by three kinds of movements such as translation, sliding and rotation. Many other researches have performed the computation of hydrodynamic lubrication pressure with half Sommerfeld boundary condition for three kinds of movements which has some physical simplications. In our study, we impose Reynolds' boundary condition for the computation of fluid film pressure, which meets more physical conditions of piston movements as it is shown in Figure 3. In the region of cavitation, none of the movements can make hydrodynamic lubrication pressures.

$$\frac{\partial}{\partial y^*}\left(h^{*3}\frac{\partial P^*}{\partial y^*}\right)+\frac{\partial}{\partial \theta}\left(h^{*3}\frac{\partial P^*}{\partial \theta}\right)=-U^*\left(\frac{(\varepsilon_h-\varepsilon_t)\cos\theta}{L^*}+\frac{\partial f(y^*,\theta)}{\partial y^*}\right)+\beta\left(\dot{\varepsilon}_t\cos\theta+\left(\dot{\varepsilon}_h-\dot{\varepsilon}_t\right)\frac{y^*}{L^*}\cos\theta\right) \quad (1)$$

Total pressure p^* is obtained by the summation of three kinds of movements as Eq. (2).

$$P^* = -U^* P_1^* + \dot{\varepsilon}_t P_2^* + \left(\dot{\varepsilon}_h - \dot{\varepsilon}_t\right) P_3^* \quad (2)$$

for the pressure P_1^* of sliding:

$$\frac{\partial}{\partial y^*}\left(h^{*3}\frac{\partial P^*}{\partial y^*}\right)+\frac{\partial}{\partial \theta}\left(h^{*3}\frac{\partial P^*}{\partial \theta}\right)=\left(\frac{(\varepsilon_h-\varepsilon_t)\cos\theta}{L^*}+\frac{\partial f(y^*,\theta)}{\partial y^*}\right) \quad (3)$$

for the pressure P_2^* of translation:

$$\frac{\partial}{\partial y^*}\left(h^{*3}\frac{\partial P^*}{\partial y^*}\right)+\frac{\partial}{\partial \theta}\left(h^{*3}\frac{\partial P^*}{\partial \theta}\right)=\beta\cos\theta \quad (4)$$

for the pressure P_3^* of rotation:

$$\frac{\partial}{\partial y^*}\left(h^{*3}\frac{\partial P^*}{\partial y^*}\right)+\frac{\partial}{\partial \theta}\left(h^{*3}\frac{\partial P^*}{\partial \theta}\right)=\frac{y^*}{L^*}\beta\cos\theta \quad (5)$$

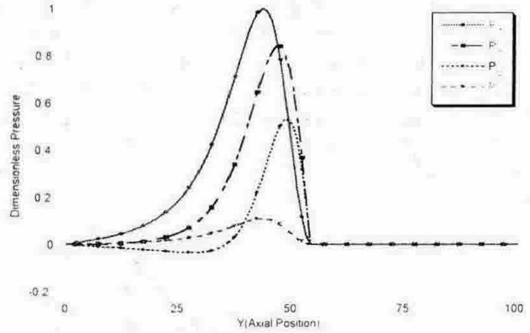


Figure 3 Pressure profile of each component of movement using Reynolds boundary condition and total pressure depending on the constants U^*, $d\varepsilon^*_b/dt$ and $d\varepsilon^*_t/dt$

4. RESULTS

Regarding the effects of clearance size on the piston movements, it is found that large clearance between piston and liner can cause side impact to the cylinder block as it is shown in Figure 4. The upper and lower locations of piston agitate in wider range at c=60μm than at c=20μm.

Skirt profile of barrel type can have lower side impact than the flat type as shown in Figure 5 at the same amount of clearance size, especially during the explosion period (0-180 crank angle degree).

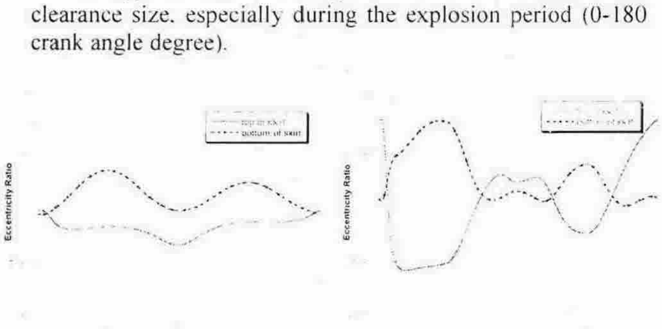
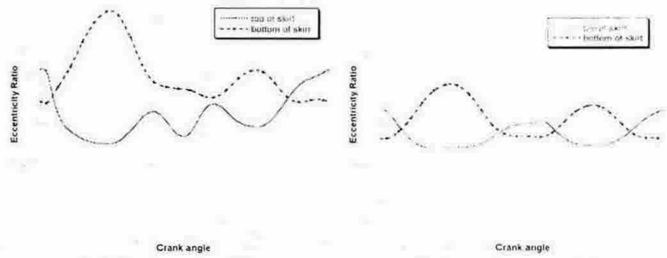
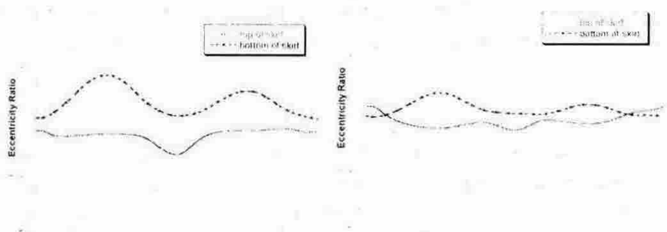


Figure 4 Piston movements according to the clearance sizes at 0.4*mm* offset and 4000 *rpm*



(a) flat type c=20μm (b) barrel type c=20μm

Figure 5 Piston movements according to skirt profiles at 0.4*mm* offset and 4000 *rpm*



(a) offset =0.2*mm* (b) offset=0.4*mm*

Figure 6 Piston movements according to piston pin offset at c=20μm and 4000 *rpm*

In Figure 6, it is shown that more piston pin offset to the thrust side (minus direction of eccentricity) makes very well aligning movement. This is also very useful design information for the piston assembly because it can reduce side impact (slapping) and noise and vibration.

5. CONCLUSION

In our work, we performed computational work of piston movement in the clearance for reducing side impact to the cylinder liner. It is found that barrel type of skirt profile and small clearance and large piston offset reduce side impact.

6. REFERENCES

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7. ACKNOLEGEMENT

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