Journal of Advanced Research in Fluid Mechanics and Thermal Sciences

Journal homepage:
https://semarakilmu.com.my/journals/index.php/fluid_mechanics_thermal_sciences/index
ISSN: 2289-7879

Influence of Macro-Scale Cylinder Liner Partial Surface Texturing on the Tribological Behavior of Two-Stroke Marine Diesel Engine Piston Ring

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ARTICLE INFO

Article history:
Received 8 June 2022
Received in revised form 10 November 2022
Accepted 22 November 2022
Available online 13 December 2022

Keywords:
Piston ring lubrication; partial surface texturing; effect of surface oil groove

ABSTRACT

In this paper, the influence of cylinder liner partial surface texturing in the top and bottom dead centers proximities was investigated. That influence was compared to full stroke surface texturing in order to evaluate the function of texturing on the surface of cylinder bore of low-speed marine diesel engine to improve the tribological performance of ring-liner pairs. Cylinder liner texturing has been designed in the form of macro-scale circumferential parabolic bottom shape oil grooves, with different depth-to-width ratios and area coverage density. Average Reynolds equation has been applied considering the oil flow factors through the micro asperities. Oil grooves has been numerically represented through the oil film configuration between piston ring and cylinder liner. The solution provides the cyclic average values of friction force of hydrodynamic lubrication, boundary lubrication, and total friction force. Results showed that the partial surface texturing has the superiority in improving tribological behavior of piston ring rather than that associated with full surface texturing.

1. Introduction

Surface texturing has been widely used in industrial areas to maintain the tribological behavior of the mechanical components that move relative to each other [1]. The micro asperities exist on the surface could be considered as the most modest type of surface texturing; however, it could act like a large number of hydrodynamic bearings [2]. It has been shown that the micro-texture has a vital effect generating additional supporting hydrodynamic pressure [3,4]. Partial surface texturing supremacy is obvious in micro-scale so that, most of works that discuss the advantages of partial surface texturing focus on the piston ring surface texturing. Thus Y. Kligerman *et al.,* [5,6] studied the reduction of the friction force between a partially surface textured piston ring and a cylinder liner. The ring surface texturing was represented in form of circular dimples. Authors developed a numerical model using Reynolds equation to study the effect of friction force between the running surfaces. According to their conclusions, increasing of dimple area density causes further decreasing

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https://doi.org/10.37934/arfmts.101.2.111120

in average friction force. Another important observation was that the optimum partial surface texturing reduces the average friction force by 30-55% compared to that of fully texturing. The former analytical study was investigated experimentally by Ryk *et al.,* [7,8]. Friction force between parallel flat surface and cylinder liner was measured using a reciprocating test rig. Comparing with a non-textured case, as well as with a full textured case, it was found that within the speed limitation of the test rig (1200 rpm), the friction could be reduced up to about 25%. However, the effect of texturing is not expected to be the same for different piston ring face profiles [9,10]. Furthermore, the retaining effect of the oil dimples was neglected, so the additional oil supply extracted from the dimples was not considered. Tomanik used a one-dimensional model to simulate surface texture effect on piston rings pack [10]. The model results provided the hydrodynamic pressure of the textured surface as well as the asperity contact pressure. One of his study observations was that the partial surface texturing is strongly recommended for flat piston rings. He agreed with Brizmer in referring that the Partial surface texture in micro scale tends to create an “equivalent Rayleigh-step” by the difference in geometry between the textured and non-textured regions [11]. That induces a significant increase in hydrodynamic pressure which was determined and the result showed that the dimensionless average pressure increases with the increase of the slender ratio that could be indication to the coverage are ratio.

It was stated also that the elliptical dimples orientation parallel to the sliding direction leads to the largest value of the average pressure. Finally, it was concluded that an optimum value of dimple depth could be done for any given set of operating parameters, while no optimum value for area density [12].

Adding controlled macro-scale surface texturing to the cylinder liner could be more beneficial as it lasts for the life of engine. However, the mathematical modeling of oil film thickness between textured cylinder liner and the piston ring texturing requires a fine resolution computational mesh. The continuous super-positioning between piston ring face and cylinder liner oil grooves requires more accurate as well as complicated computational efforts for a suitable integration of the solution domain. Considering those computational efforts, a numerical model has been developed and introduced in this paper to study and compare between the effects of using full and partial texturing of cylinder liner in order to improve the tribological behavior of the first compression piston ring of 2-stroke marine diesel engine.

2. Mathematical Representation

Calculating oil film thickness and hydrodynamic pressure between piston ring and cylinder liner simultaneously was carried out by applying Reynolds equation [13,14].

$$\frac{\partial}{\partial \mathrm{x}}\left(\varnothing_{\mathrm{x}}\frac{\mathrm{h}^3}{\mu}\frac{\partial \mathrm{P}}{\partial \mathrm{x}}\right) = 6\mathrm{U}\varnothing_{\mathrm{c}}\frac{\partial \mathrm{h}}{\partial \mathrm{x}} + 6\mathrm{U}\sigma\frac{\partial \varnothing_{\mathrm{s}}}{\partial \mathrm{x}} + 2\varnothing_{\mathrm{c}}\frac{\partial \mathrm{h}}{\partial \mathrm{t}} \tag{1}$$

where U is the linear velocity of piston ring, h is the oil film thickness, and P is the oil film pressure, μ is the dynamic viscosity of lubricating oil.$\phi_x, \phi_C, and\ \phi_s$are correction factors could be referenced to Patir’s study [13]

The nominal oil film thickness between piston ring and cylinder liner was modeled as

$$h_n(t) = h_m(t) + h_s(x) \tag{2}$$

where h_m is the minimum oil film thickness (MOFT), and $\mathrm{h_s}(x)$ represents the profile of the ring surface as depicted in Figure 1, and is shown as follows

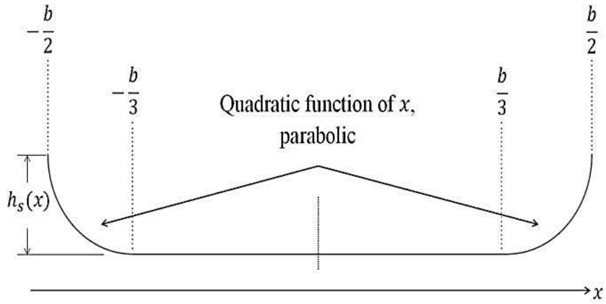


Fig. 1. Details of piston ring face profile textured

$$h_s(x) = \left(\frac{3x+b}{1.25-b}\right)^2 \quad for \quad -b/2 \le x < -b/3$$

$$h_s(x) = 0 \qquad for \quad -b/3 \le x \le b/3 \tag{3}$$

$$h_s(x) = \left(\frac{3x-b}{1.25-b}\right)^2 \quad for \quad b/3 < x \le b/2$$

The oil film configuration between the ring face and the liner surface has been modeled as following **Error! Reference source not found.**

$$\mathrm{h} = \mathrm{h_n(t)} + \sqrt{\left[\frac{\mathrm{h_p^2+r_p^2}}{\mathrm{2h_p}}\right]^2 - \zeta^2} - \left[\frac{\mathrm{r_p^2-h_p^2}}{\mathrm{2h_p}}\right] \quad \text{if } |\zeta| < \mathrm{r_p} \tag{4}$$

$$\mathrm{h} = \mathrm{h_n(t)} \qquad \text{if } |\zeta| \ge \mathrm{r_p}$$

For partial surface textured cylinder liner, it has been considered that the textured areas are twenty crank angle degrees after and before top dead center (TDC) and bottom dead center (BDC) respectively as shown in Figure 2. While the full textured cylinder has the grooves all along the stroke area. Piston ring stroke was divided into circumferential cells with constant widths of ζ (ζ = 4 mm in this work), each cell contains one oil groove of width $2r_p$, and depth h_p as shown in Figure 3.

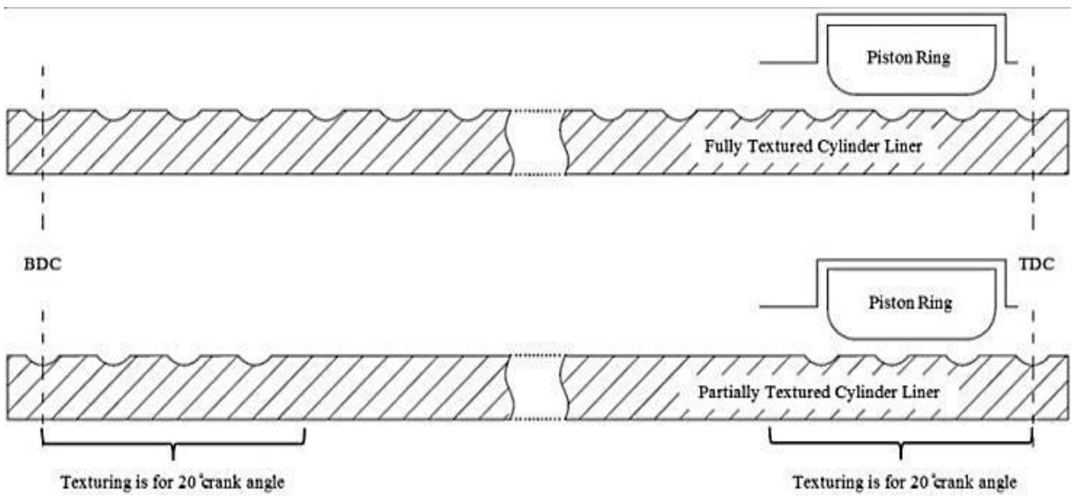


Fig. 2. Schematic cross-section of fully and partially cylinder liner

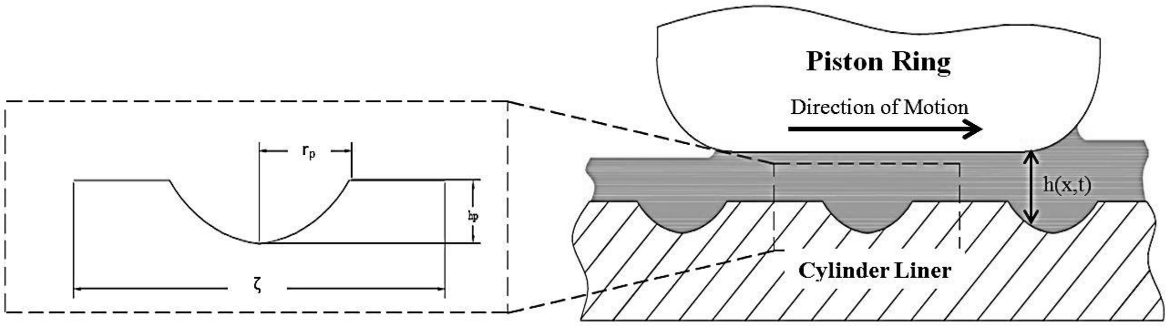


Fig. 3. Schematic drawing of a single surface cell containing one oil groove

According to literature, the area coverage of grooves (S_p) and the groove aspect ratio (e) were assumed to be the two major design parameters of the groove dimension [15], and they are mathematically shown as;

$S_p =(2r_p / \zeta) * 100$, $e = hp / 2r_p$

3. Asperity Contact Model

A stochastic model developed by Greenwood and Tripp [18] was employed to calculate the asperity contact pressure between surfaces of cylinder liner and piston ring

$$\mathrm{P_{asp}} = \left(\frac{16\sqrt{2}}{15}\right)\pi(\eta\beta\sigma)^2\acute{\mathrm{E}}\sqrt{\frac{\sigma}{\beta}}\mathrm{F_{2.5}(H)} \tag{7}$$

where β is the radius of peak, η the peak density. $F_{2.5}(H)$ is the asperity height probability distribution function. Gaussian distribution asperities contact force as a function of asperities pressure calculated by the following equation

$$\mathrm{F_{asp}} = 2\pi\mathrm{R}\int_0^b \mathrm{P_{asp}}.\,\mathrm{dx} \tag{8}$$

Continuous change in oil film configuration between piston ring and cylinder liner creates a drop in pressure in piston ring trailing edge region causes discontinuity of the oil film. According to Reynolds boundary conditions, the discontinuity boundary condition can be assumed as

$$\frac{\mathrm{dP}}{\mathrm{dx}} = 0 \tag{9}$$

$P = P_{Cavity}$

$\mathrm{P_{cavity}}$ is assumed to be atmospheric pressure. If there is no cavity at the ring exit. The boundary condition assumed to be as follows

$$P(b/2) = \mathrm{P}_T \tag{10}$$

where P_T is the exit pressure. For a fully flooded lubrication conditions at the front of piston, the leading edge pressure is

$$P(-b/2) = P_L \tag{10′}$$

where P_L is the pressure before the ring.

4. Friction Force Calculation

In mixed lubrication regime, the total friction force between cylinder liner and piston ring surface consists of two components; the shear hydrodynamic friction in the oil film layers, and the asperity contact force. The two components could be compound into the following expression to calculate the total friction force.

$$F_{fric} = -2\pi R \int_0^b \left\{ \frac{\mu U}{h}\left(\emptyset_f + \emptyset_{fs}\right) + \emptyset_{fp}\frac{h}{2}\frac{\partial P}{\partial x} + C_f P_{asp} \right\} dx \tag{11}$$

where ϕ_f, ϕ_{fs}, and ϕ_{fp} are surface roughness correction factors referred to Patir's model **Error! Reference source not found.**. C_f is the coefficient of asperities friction, R is ring radius.

Lubricating oil flow rate per unit length through each crank angle has been calculated to consider the oil consumption because of the grooves existence. Oil flowrate could be calculated according to Eq. (12)

$$q = -\frac{h^3}{12\mu}\frac{\partial P}{\partial x} + \frac{Uh}{2} \tag{12}$$

5. Scheme of Numerical Simulation

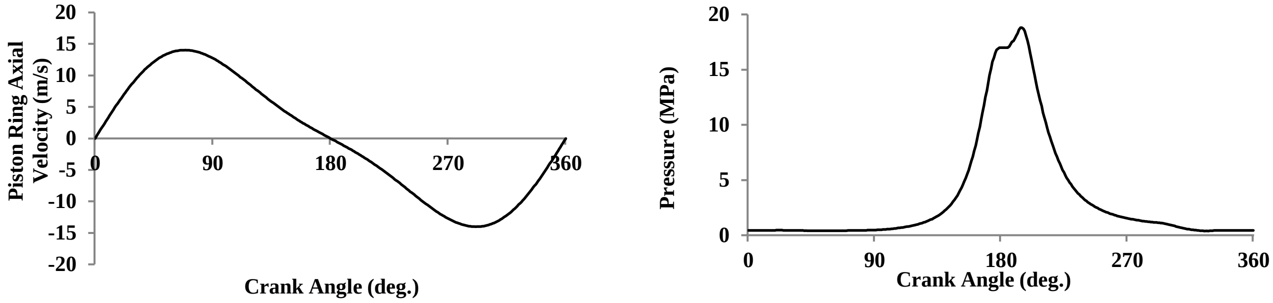
A mathematical simulation was developed using Matlab to monitor the tribological interaction between the cylinder liner and the top compression piston ring of 2-stroke marine diesel engine. Cylinder liner surface enhanced with perimetric oil grooves. In case of full textured cylinder liner the grooves are distributed uniformly all along the engine stroke. However, it was important to examine the piston ring-cylinder liner interaction in case of partially textured cylinder liner (*i.e.,* existence of grooves at the TDC and BDC proximities). Governing equations have been solved using Finite Difference Method, using half degree crank angle as a step increment. Crank angle of BDC was assumed to be zero at which the simulation start point. All calculations have been averaged for one complete engine cycle. Numerical scheme starts with assuming an initial MOFT (h_m) to calculate the oil pressure along the ring face. The MOFT and oil film pressure were calculated simultaneously to satisfy the force balance criterion at each crank angle step according to Eq. (12)

$$\frac{F_W - F_Z}{F_Z} \leq 0.0001 \tag{12}$$

where F_W is the summation of forces act radially to the cylinder liner, and F_Z is the summation of oil film pressure and asperity contact forces. Oil film thickness at each crank angel was determined taking into account the superposition between the ring face and the according oil groove. The mathematical model results have been verified experimentally using reciprocating tribometer described with further details in previous work for the author **Error! Reference source not found.**.

6. Results and Discussions

The dynamic parameters of piston ring, such as linear speed and gas pressure, have been input to the calculation algorithm as practical numerical values as shown in Figure 4. Cylinder liner oil grooves assumed to have a parabolic bottom shape, with different area densities and different aspect ratios.



(a) (b)

Fig. 4. Dynamic parameters of piston ring (a) Piston ring linear velocity (b) Combustion chamber gas pressure

Figure 5 depicts the non-dimensional average oil flow rate through the gap between the piston ring face and cylinder wall, using full and partial surface texturing with different grooves dimensions. It could be seen that grooves caused a noticeable increase in oil flow rate, with a steeply increasing at largest grooves area density (S_p=50%). The rate of increase is larger in case of fully textured cylinder liner. However, the rate of increasing of oil flow is limited for small groove area density (10%).

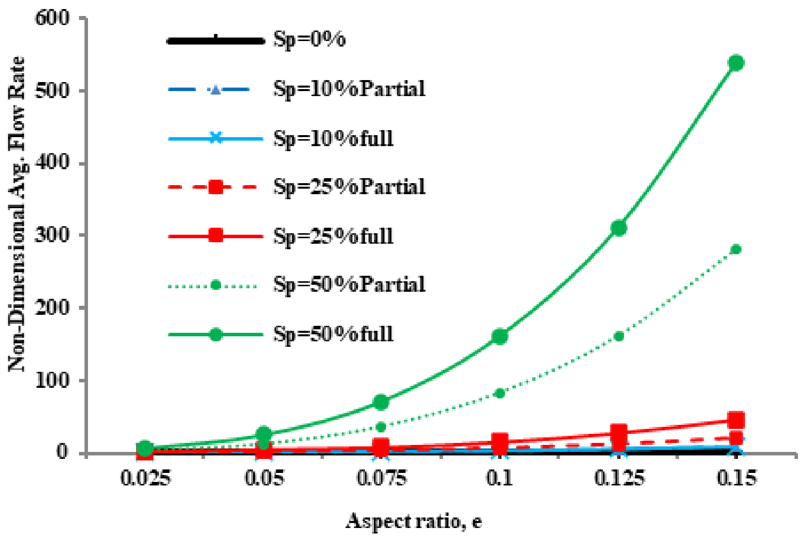


Fig. 5. Non-dimensional cyclic average oil flow rate between piston ring and cylinder liner

Oil flow rate is increased by increasing aspect ratios for higher area densities and reach its maximum at the largest grooves dimensions. As it well expected, oil flow rate affects the oil consumption as well as affect the hydrodynamic friction forces as shown in Figure 6.

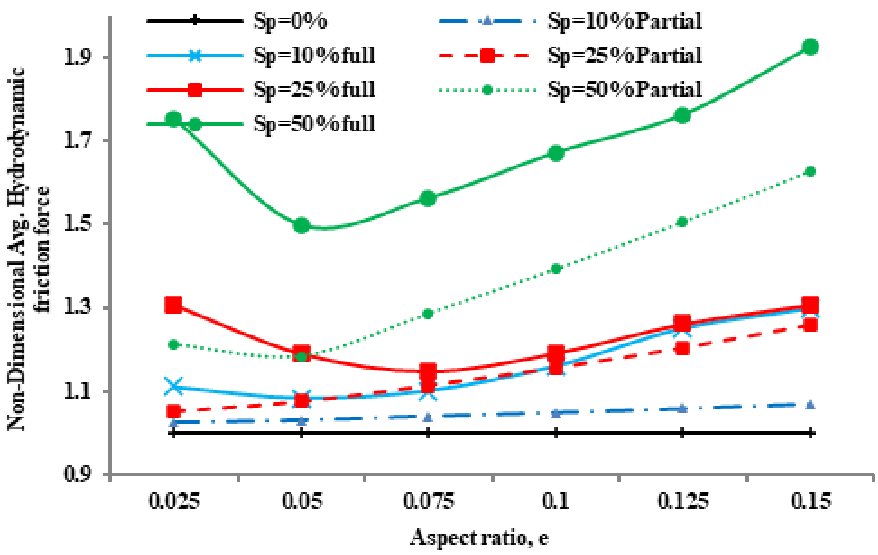


Fig. 6. Effect of grooves aspect ratio on non-dimensional cyclic average hydrodynamic friction force

Generally speaking, the hydrodynamic friction in all combinations of grooves dimensions were increases in comparison to the cylinder liner without grooves (S_P=0). The effect of area density attributed to higher oil influx resistance that is caused by the oil grooves on the cylinder liner, which in turns, and due to no-slip condition, leads to higher shear stress in the lubricant layers between the ring surface and the cylinder liner. Eventually, the possibility of hydrodynamic lubrication regime increases. That could be vital to protect the engine parts from direct metal-to-metal contact. However, that contributes for increase the hydrodynamic friction losses as well [20].

Generally, the hydrodynamic friction increasing rate is relatively smaller for ranges of groove dimensions in case of partial texturing due to smaller textured area, and especially with small groove aspect ratios. Moreover, the difference between partial and full texturing influence become more insignificant with decreasing either the grooves coverage area or aspect ratio. Friction force due to asperity contact pressure tightly correlates to lubrication regime. Figure 7 depicts the average non-dimensional asperity contact friction force between piston ring and cylinder liner containing parabolic oil grooves. For a fully textured surface, it could be claimed that as the groove aspect ratio increases (e>0.05), or groove area density decreases (S_p≤25%), there is a decrease in the asperity contact friction force. While partially textured surface showed a different behavior, as the minimum asperity contact force has been achieved by larger oil groove density. It could be further noticed that grooves aspect ratio has a limited effect in asperity contact friction force, especially for case of partial surface texture. The average total friction forces have been calculated and presented in Figure 8.

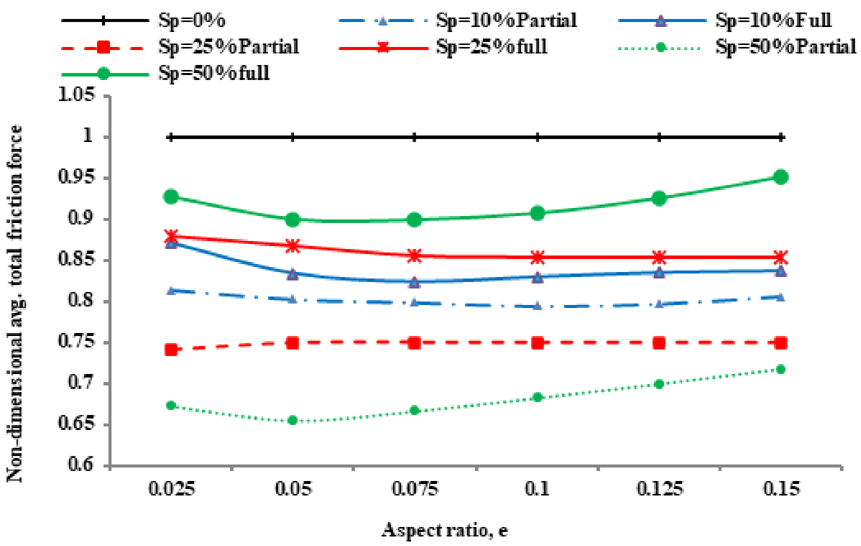


Fig. 7. Effect of grooves aspect ratio on non-dimensional cyclic average asperity contact friction force

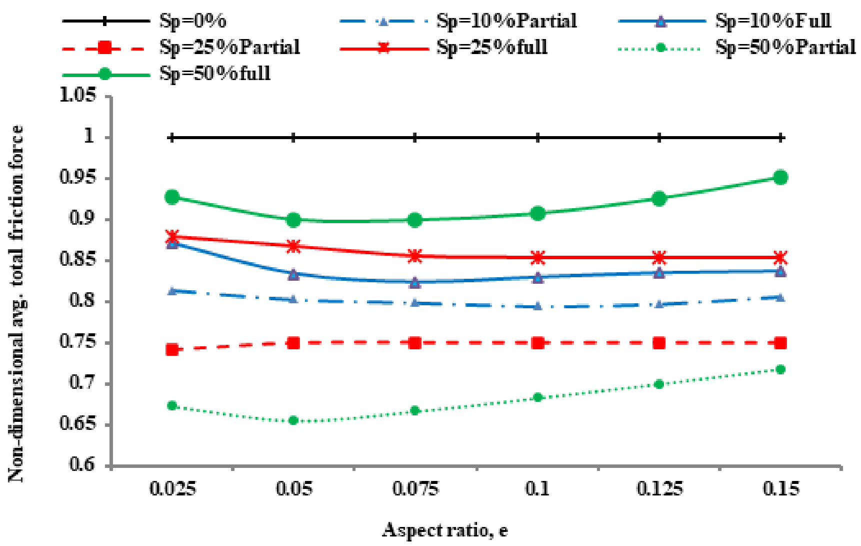


Fig. 8. Effect of grooves aspect ratio on non-dimensional cyclic average total friction force

Although the increase in hydrodynamic friction, the oil grooves cause a significant decrease in total friction force, an evidence to the effective decrease in asperity contact friction. In case of fully textured cylinder liner, the total friction forces decreased (up to about 15%) with decreasing the grooves area densities. On the other hand, friction force decreasing rate is increased with partially textures cylinder liner (up to about 35% decrease in total friction), and the influence of area density is reversed, i.e. the total friction is decreased with increasing the groove density.

Oil grooves generate a significant resistance fluid influx, and the hydrodynamic pressure oscillations exist over the textured zone **Error! Reference source not found.**. In macro-scale texturing, where the width of the groove is comparable to that of the piston ring, increasing the area density leads to further increase in the oil flow, which in turns leads to further increase in hydrodynamic friction force. At mid-stroke position, where the maximum velocity of piston ring occurs, the oil film thickness is maximum and the hydrodynamic lubrication dominates the lubrication regime in this case. Further increase in the oil flow by oil grooves exist in these sites leads to further hydrodynamic power losses without any additional benefits to the lubricating conditions furthermore; it contributes to increase oil consumption. From other perspective, mixed lubrication dominates the TDC and BDC proximities due to the low speed of the piston ring, and hence the oil film thickness is minimum at those sites, at which the opportunities of metal-to-metal contact is increased, and the boundary friction forces are maximized. The partial surface texturing in TDC and BDC proximities provides an effective hydrodynamic lubrication, and decreases the boundary friction force. The effect of the larger grooves area densities is the best at the sites of severe lubrication conditions however; its negative effect of increasing viscous friction in other stroke sites dominates the results of full textured cylinder liner [21]. While the smaller area densities grooves has overall acceptable effect all over the piston stroke. The superiority of partial surface texturing in this study agrees with literature perspectives, however, to the best of authors knowledge, most of the previous studies were focusing on the micro texturing on the piston ring and a significant hydrodynamic pressure step created by the difference in geometry between the textured and non-textured region.

7. Conclusion

A numerical model has been developed, using average flow theory, to study the effect of fully and partially macro-scale texturing on the cylinder liner tribological behavior. The result revealed that generally, full surface texturing could improve the lubricating conditions of the marine engine piston ring, but large area density grooves (Sp>25%) contributes to increase the oil consumption. Generally, Partial surface texturing in the vicinity of TDC and BDC could improve the lubricating conditions compared with that of full texturing. Partial surface texturing with large groove area densities revealed the best results in decreasing the total friction force. The grooves with larger aspect ratio (e>0.75), and smaller area densities (Sp<50%) has the best results for full stroke surface texturing, while effect of groove aspect ratio is insignificant for partial surface texturing. This research needs to be expanded with a future work about calculating the oil hydrodynamic pressure and cavitation.

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