Tribological Behaviour under Conformal and Non-Conformal Contact Condition of Piston Ring and Cylinder Liner Material in a Reciprocating Bench Test

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Abstract. Experiments on cylinder liner and piston ring were carried out on a reciprocating tester to measure the coefficient of friction. The tribological behaviour for conformal and non-conformal contacts was compared. Actual cylinder liner is and piston ring is used for conformal contact, which is cut as per the required dimension. After analyzing the composition of the liner material, a specimen is prepared for non-conformal contact. Tests were conducted for variable load, variable speed and variable temperature with various commercial lubricants to measure the friction coefficient. The result shows that conformal contact shows significant variation in friction coefficient as compared to non-conformal contact for all operating parameters. The geometry of contact, oil quantity, and viscosity of lubricant plays a vital role to characterize the behaviour of friction coefficient and wear.

Keywords: Friction, Wear, Cylinder liner, Piston ring, conformal and non-conformal contact.

1. Introduction

The engine is a major responsible component in vehicular design in the petrol and diesel engines where a great deal of energy loss at the interface of cylinder liner wall surface and piston ring[1]. The factors which are responsible for friction losses are gas pressure forces due to combustion, ring tension, hydrodynamic pressure, inertia force due to mass of the ring, speed and temperature of engine and asperity contact forces [2]. Due to the mechanical motion engine oil is used as a lubricant to reduce friction and wear. The properties of the lubricants used in the engine plays an important role in to extend the life and improve the performance of an engine [3]. The operating conditions and performance of the contact also depend on the shape of the bodies in contact[4].

The physic-chemical and tribological properties of commercial engine oil plays a vital role and it is observed that the coefficient of friction and load have an inverse correlation [5]. Keller [6] has understood the topography on the tribology cast iron and observes that friction coefficient mainly influenced by surface asperities and an initial presence of these asperities produces the high friction coefficient. Sutaria [7] developed an algorithm for calculation of the cylinder pressure and piston ring assembly friction with crank angle under unfired single cylinder four-stroke gasoline engines. The results indicate that piston speed, piston ring tension and viscosity of lube oil are playing an important role in the friction forces. Riyadh [8] has performed an experiment on a test rig and observed that coefficient of friction decreases as the load increases due to the temperature rise between the two sliding surfaces and its effect on the viscosity of lubrication. Truhan [9] has investigated that coefficient of friction is dependent on the load and temperature. Mixed or boundary lubrication condition exists at room temperature. Peggy [10] has conducted experiments and analyzes the effect of low phosphorous engine oil that after surface break-in all the oil containing friction modifiers observed reduces a coefficient of friction. To understand the interactions between the surface tension, the surface energy of the solid phase and their influence on the coefficient of friction, it is observed that wth the increasing oil temperature, the stribeck curve minimum is shifted to a higher sliding speed due to lower viscosity [11]. In this work, tribological behaviour for conformal and non-conformal contact for cylinder liner and piston ring were compared. Various commercial lubricants were used for the experimental work to measure the friction coefficient.

2. Experimental setup

2.1 Specimen preparation

The material used for liner-specimen was taken from the market. Test samples were cut from real engine parts to keep original curved surface finishing. For a non-conformal contact, specimens were prepared as per the dimension of test rig holder. After analyzing the composition and microstructure of liner material, cast iron plates were prepared by casting and subsequently turned and grounded. The hardness of the plates was 79.8HRC and the surface roughness was 0.365±0.2µm. The piston ring segments were taken from the market which is actually used in Honda engine with a capacity of 100CC. Piston ring was cut into two halves to fix on the groove of the ring attachment holder. The surface roughness of piston ring was 2.75±0.05 µm.

2.2 Experimental Apparatus and Procedure

Tests were carried out with a reciprocating friction and wear monitor test rig (TR-281M-M6). A piston ring segment reciprocates against the counter cylinder liner segment. In this work, tests were performed with a range of load, speed, temperature and lubricants for conformal and non-conformal contact. Prior to each test, the ring and liner were cleaned by acetone to remove metal fragments and oil from the surface. Before start, the test, 2drops of oil is added to the contact surface of liner and piston ring. New specimens were used for each test. The Win Ducom software was used for data acquisition and display of results. An electronic weighing balance (accuracy of 0.0001mg) was used to measure the weight loss of the specimens. The above procedure is repeated for all tests.

Figure 1. Reciprocating test rig

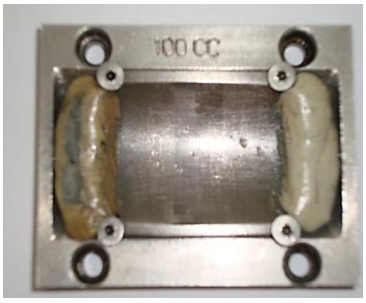

Figure 2. Holder and liner with conformal contact

Figure 3 Holder and liner with non – conformal contact

3. Results and Discussion

A series of experiments were carried out for conformal and non-conformal contacts to measure the friction coefficient and he results are graphically illustrated

Table 1 Oil Properties

Lubricants	**Viscosity(cSt)**		**Flash Point (°C min)**	**Viscosity Index**
	40°C	**100°C**		
SAE10W30	119.75	11.51	228	102
SAE20W40	124	14.3	240	114
SAE20W50	156.3	17.87	256	126

Table 2 Material composition of prepared cylinder liner on X-met 5000-portable Analyzer

Elements	Fe	C	Si	Mn	Cr	Cu	P	S
Weight (%)	Rest	3.16	2.05	0.67	0.3	0.27	0.21	0.062

3.1 Frictional behaviour

3.1.1 Effect of speed

Figure 4 and figure 5 summarizes the effect of speed for various lubricants on the friction behaviour of sliding pair of the cylinder liner and piston ring for conformal and non-conformal contacts. With the increase of speed, for conformal contact, the friction coefficient range from 0.178 to 0.04

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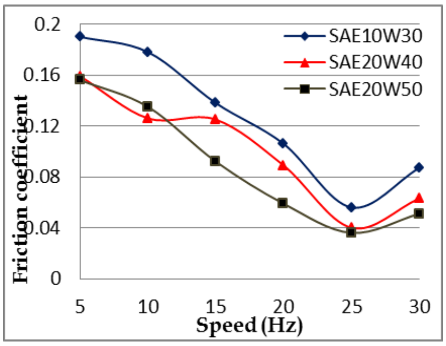

Figure 4. Effect of speed on friction coefficient for CC

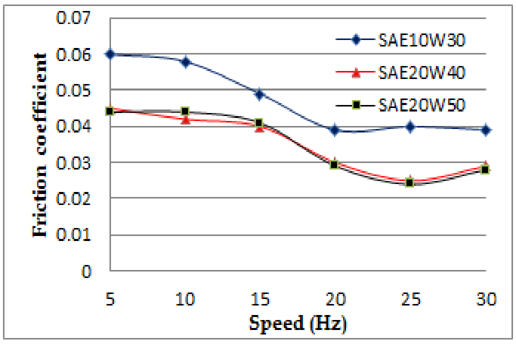

Figure 5. Effect of speed on friction coefficient for NCC

while for non-conformal contact, the range is 0.06 to 0.03. It shows that contact area and the quantity of lubricant play a vital role. From the graph, it is observed that at lower operating speed, it offers highest friction coefficient. At lower operating speed, a hydrodynamic action is not strong, so the film thickness may be small [12]. Smaller film thickness leads to an increase in the asperity contact between the contact surfaces, results in high friction coefficient. For non-conformal contact, elastohydrodynamic influence is more evident [13]. This effect is visible for all sliding speeds. With the further increase of speed, the hydrodynamic pressure developed between the contact surfaces and shearing between the contact surfaces becomes smaller. This may be due to the mixed film developed between the contact surfaces. Afterwards, a large amount of hydrodynamic pressure generated and negligible shear can occur between the surfaces. Due to that effect, contact surfaces completely separated by the fluid film and the friction is generated due to the shearing of the lubricant fluid film which is dependent on the rheological properties of the lubricant. The result is friction coefficient again increases.

3.1.2 Effect of load

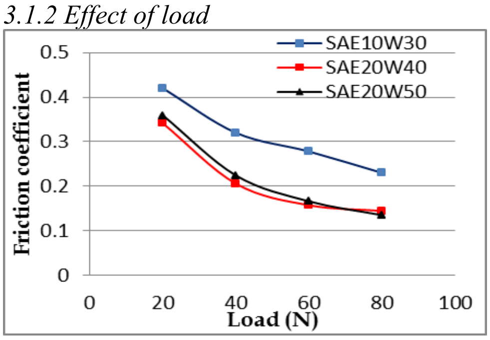

Fig.6. Effect of load on friction coefficient for CC

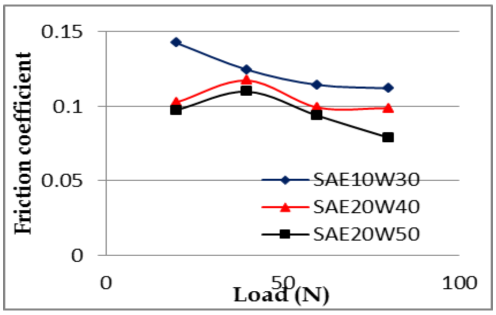

Figure 7. Effect of load on friction coefficient for NCC

Above figure shows the effect of load on the coefficient of friction at 600 rpm and ambient temperature for various lubricants. The nature of curve for all lubricants is similar and follows a negative trend for conformal contact while for non-conformal contact only little reduction takes place. For conformal contact, the friction coefficient range from 0.420 to 0.135 while for non-conformal contact, the range is 0.143 to 0.079. During the initial stage, surface irregularities play a major role for friction coefficient. With the effect of small load, no more contact area generated between the specimens. Boundary lubrication regime is developed between the surfaces which may be responsible for increasing the friction coefficient. With the increase of load, surface irregularities in the form of asperities increase the contact area between the sliding surfaces. The surface becomes smooth due to the high wear rate hence mixed lubrication is developed. Lubricant formed film which may be stable due to the smooth geometry between surfaces. In non-conformal contact, full film lubrication condition exists between the mating surfaces which are responsible to developed elastohydrodynamic lubrication. The result is little reduction in friction coefficient.

3.1.3 Effect of temperature

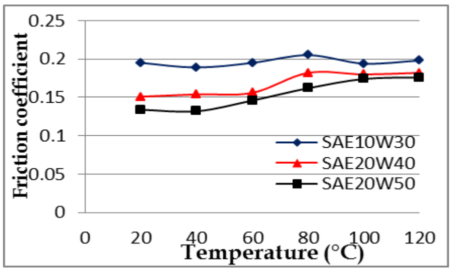

Figure 8. Effect of temperature on friction coefficient for CC

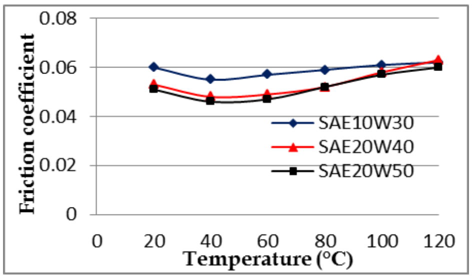

Figure 9. Effect of temperature on friction coefficient for NCC

It is observed from the figure that the friction coefficient changes with change of temperature for conformal and non-conformal contact both. SAE10W30 offers stable performance compare with SAE20W40 and SAE20W50. This may be due to SAE20W40 and SAE20W50 are a more viscous lubricant, so they might be more affected with temperature as compare to SAE10W30. With the increase of temperature, the viscosity of the lubricant reduced and it became thinner so it generates asperity contact between the mating surfaces. This change the lubrication regime from mixed to boundary which may be responsible for increasing the friction coefficient.

From the experiments, it was found that conformal contact shows more friction coefficient as compare to non-conformal contact. In non-conformal contact, the graph shows almost stable or little change in their performance. Here, the same amount of oil is added for both types of contacts. The contact area covered by the specimen is quite more in conformal contact which offers high friction coefficient. This may be due to the large covered area; lubricant is pushed aside during its one stroke and does not flow back in time for the return stroke. It causes starvation which reduced oil film thickness due to the inadequate volume of the lubricant present at inlet [14] [15].

With non-conformal contact, sufficient amount of lubricant is placed between the liner and piston ring. The full film can generate which reduce the metal to metal contact and moves under elastohydrodynamic regime result are lower friction coefficient. It means that friction and wear are more affected with conformal contact only. In a real engine, there are conformal contacts generating between the cylinder liner and piston rings. So to analyse the wear behaviour, conformal contact is taken for the further experiments.

3.2 Weight loss of cylinder liner and piston ring

The experiment is performed for a constant load 80N and duration is 100 min to calculate the weight loss of the cylinder liner and piston ring for various lubricants.

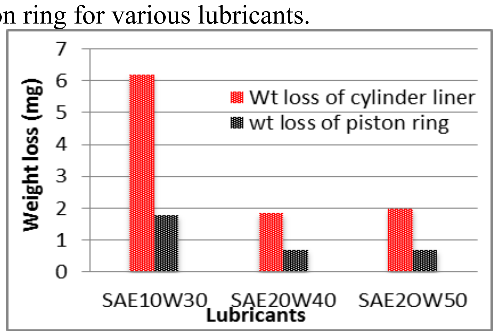

Figure 10. Weight loss of the piston ring and cylinder liner for CC

Figure 10 shows the comparisons of the weight loss of cylinder liner and piston ring. Cylinder liner offers more weight loss as compare to piston ring and SAE10W30 have a major weight reduction as compare to SAE20W40 and SAE20W50. The cylinder liner shows 2 to 5times greater weight loss as compare to the piston ring. This relationship is reversed in an actual operating engine [16, 17].

Here, cylinder liner offers 70.96%, 61.08% and 65% higher weight loss as compare to the piston ring. The wear loss with SAE10W30 lubricant for piston ring is 43.64% and 34.55% and for cylinder liner is 63.80 and 62.86% high as compare to cylinder liner and piston ring. Viscosity plays a vital role in this phenomenon. SAE10W30 is a low viscous lubricant so it may be not capable to form a fluid film between the sliding surfaces result is more wear occurs.

In an actual engine, the piston ring experiences more wear as compare to cylinder liner but at laboratory scale, it's vice versa. In a combustion chamber, variations in pressure, temperature and gas composition occurs which are difficult to reproduce in a laboratory test [17]. There are dissimilarities in geometry between the ring and liner and instead of cycling loading, it perform constant load throughout the operation which results high wear generates in a test rig [18].

4. Conclusion

Based on the presented experimental outcomes, following conclusions have been drawn.

- Conformal contact offers high friction coefficient for all operating condition. In this case, the same amount of oil is added for both contacts but the contact area covered by the conformal contact is quite more. So starvation occurs due to that thin film formed which caused boundary regime and increases friction coefficient. The operating conditions and performance of the contact are strongly depends on the contact geometry of the specimen.
- Mixed lubrication plays the significant role in decreasing the friction coefficient for conformal contact while elastohydrodynamic influence is more evident with non-conformal contact.
- SAE10W30 offers higher friction coefficient as compare to SAE20W40 and SAE2W50 lubricants for all operating parameters. It shows that the quality of the lubricants plays a vital role on tribological behaviour.
- There are dissimilarities in geometry and working condition between the test ring and actual engine. So during the wear analysis of the specimen, the cylinder liner shows 2 to 5times greater weight loss as compare to the piston ring. This relationship is reversed in an actual operating engine.

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