MODELLING A ROTA-DYNAMIC LUBRICANT TEST SYSTEM FOR DETERMINING ENGINE SERVICE-LIFE PERFORMANCE

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ABSTRACT. *Lubricant plays an important role in determining the performance of engine and their service life. A simple test rig simulating cam follower contact mechanism was designed and developed to test and evaluate performance of lubricant by wear scar. In the design, stainless steel ring was simulating as a cam, and a mild steel pin simulating as a flat face follower. Five groups of tests were conducted to evaluate the performance of the test rig using new and used 20W-50 motorcycle engine oil. Wear tests were carried out at 2.94N, 5.89N, 8.83N, 11.77N and 14.72N for 3 minutes at room temperature. The extent of wear was measured by wear volume using a geometric model. Performance of new and used lubricant is compared by using average wear volume. The results had shown that the magnitude of wear volume for wear tests conducted using contaminated lubricant is in the range of 1.34 – 1.77 times with respect to wear tests using pure lubricant. This had highlighted the importance of good lubricant in order to provide better performance and longer life to the engine.*

KEYWORDS. *Lubricant, Performance, and Wear Preventive Properties.*

INTRODUCTION

The performance of lubricant can be evaluated through the relative wear preventive properties (Priest, *et al.*, 1999). In a good performance lubricant (Truhan, *et al.*, 2005), a high level of wear preventive capability can be provided. The lubricant testing system is designed for the purpose to assess the performance of lubricant through wear and tear properties. The system has a simple mechanism consisting of a rotating shaft can be loaded by a cantilever to simulate a cam follower mechanism in the engine (Kano & Tanimoto, 1991). In the engine, lube pump is used to deliver a sufficient quantity of lubricant to lubricate moving engine parts. Since it takes time to deliver lubricant to all the moving parts, the wear of cam follower is most significant during staring of engine. This wear can be reduced by the thin surface films that stay in between the contact of cam and follower each time after we stop the engine. To reduce this incident, a good quality of lubricant should be used.

In Davis and Eyret research (Davis & Eyret, 1990), the friction and wear of a piston ring/cylinder bore material combination was studied using a pin-on-plate tribometer to reproduce the wear mechanisms encountered in an internal combustion engine with different types of lubricant. A similar test was done by Childs and Sabbagh (1989) to study the wear mechanisms responsible, particularly the relative importance of high cycle metal fatigue and chemical reaction film wear. Falex pin and V-blocks machine had been developed to investigate the wear behavior. Baldwin and Lee (1983) modifier this machine to simulate the valve train wears of an automobile engine. While in the case of Ramamohana Rao and Mohanram (1994), they established a bush and journal system to study the wear characteristics of journal bearings under mixed-lubrication conditions.

Observation under actual conditions is the best way to determine the lubricant performance, but it is costly. Furthermore, it is inconvenient and time consuming to disassemble a complex machine system for observing the mechanism performance. Therefore, shorter and less costly tests are developed for laboratory usage. Generally, the results of laboratory extreme pressure and wear prevention tests do not necessarily correlate with the actual conditions. However, test presently available in laboratory is the only means to evaluate these properties at a reasonable cost.

METHODOLOGY

The rota-dynamic lubricant test system is designed and developed systematically through standard stages as shown in the Figure 1. Preliminary design concepts are developed for the model to function as a lubricant test system. Theoretical model base on two dimensional analysis of basic load, materials and constraints are verified for the system. Design modelling by using CAD and CAE application tools are then generated to provide detail blue-prints, simulating functionality and design refinements of the conceptual design model. The CAE analysis provides a numerical verification before the prototyping is carried out for the best model concept. Construction development of the lubricant test system is completed with a prototype, which is tested and verified again for its physical functionality. Physical modification and improvement is always necessary before the system could used for the purpose of experimentation and data collection of wear properties of the lubricant test system.

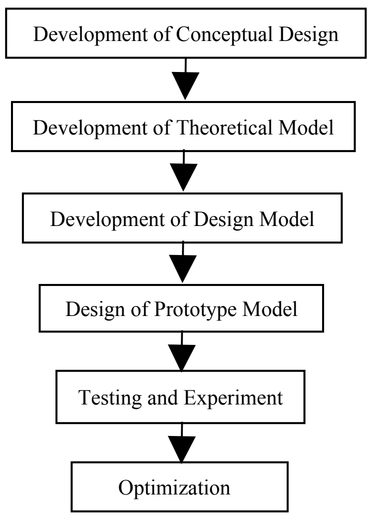
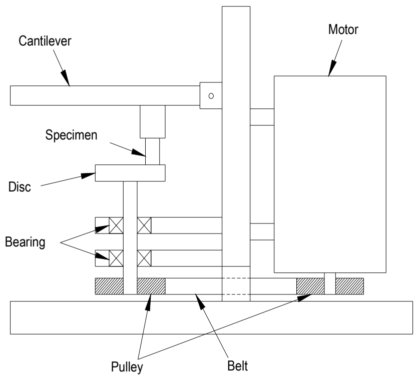


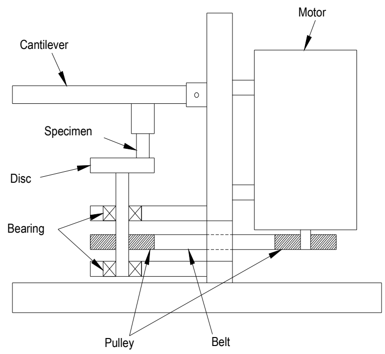
Figure 1: Work flow of design and development.

CONCEPTUAL DESIGN

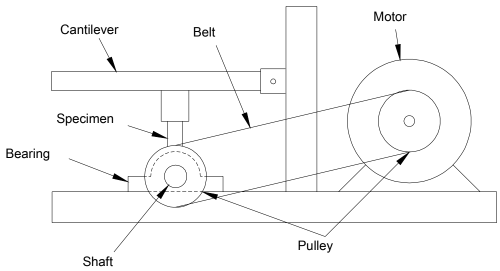
Based on the cam follower mechanism, three conceptual designs for the simulation of contact condition and mechanism of a lubricant test system are generated. Mainly, the system requires a torque output from an electric motor, which is then transmitted to the driven shaft by pulleys and belts. A cantilever concept is then utilized to amplify the load that acts on the specimen. Design selection chart is used to aid in the decision making of selecting the most appropriate conceptual model among the three proposed designs, Figure 2. A set of criteria based on manufacturability, materials, and design complexity are utilised as weightings for the evaluation of a suitable system.



Concept-1



Concept-2



Concept-3

Figure 2. Development of three conceptual designs (1, 2 & 3).

The results from the weighting of the design selection chart, Table 1, concept-3 is identified for development since it carries the best total weighting marks. An eventual three dimensional model is established as shown in the Figure 3.

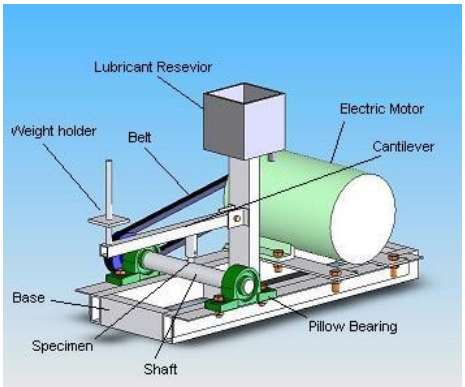


Figure 3. 3-D model of selected conceptual design.

Table 1: Design selection chart.

Criteria	Weighting	Concept			
		1	2	3	Reference
Ease of fabrication	7	S	S	+	D
Less material consumed	8	-	-	+	A
Ease of installation	7	+	+	S	T
Size	5	S	S	S	U
Machining	8	+	+	-	M
Number of components	6	S	S	+	
	Sum of +	2	2	3	0
	Sum of -	1	1	1	0
	Total	1	1	2	0
	Total of weight	8	8	12	0

THEORETICAL MODEL

Mathematic calculation software Mathcad 12 is utilsed for the theoretical calculation. The ability to recalculate is an added advantage when dealing with design alteration. For the system, the transmission shaft will be subjected to a combination of twisting and bending moment. Therefore, the minimum shaft diameter is determined by

$$d = \sqrt[3]{\frac{16 \cdot T_e}{\pi \cdot \tau_{all\_shaft}}} \qquad \text{(Due to twisting moment)} \qquad (1)$$

$$d = \sqrt[3]{\frac{32 \cdot M_e}{\pi \cdot \sigma_{all\_shaft}}} \qquad \text{(Due to bending moment)} \qquad (2)$$

where T_e is the equivalent twisting moment, M_e is the equivalent bending moment, τ_{all_shaft} is the allowable shear stress and σ_{all_shaft} is the allowable bending stress.

In order to ensure that the design structure of the test rig is safe to use, AISC codes below is used to check the critical parts for safety. Such that, for

SHEAR $\tau_{all} = 0.40 S_y$ (3)

BENDIN $0.60 S_y \le \sigma_{all} \le 0.75 S_y$ (4)

BEARING $\sigma_{all} = 0.90 S_y$ (5)

where τ_{all} is the allowable shear stress, σ_{all} is the allowable normal stress and S_y is the yields strength.
The maximum and minimum hole for the hub is also determined by

$$D_{\max} = D + \Delta D \tag{6}$$
$$D_{\min} = D \tag{7}$$

where D is the basic size of the hole and ΔD is the tolerance grade for hole. For shaft with clearance fit, the maximum and minimum diameter is determined by

$$d_{\max} = d + \Delta d \tag{8}$$
$$d_{\min} = d + \delta_F - \Delta d \tag{9}$$

While for shaft with interference fit, the maximum and minimum diameter is determined by

$$d_{\max} = d + \delta_F + \Delta d \tag{10}$$
$$d_{\min} = d + \delta_F \tag{11}$$

where d is the basic size of the shaft, Δd is the tolerance grade for the shaft and δ_F is the fundamental deviation. For the ease to mount and dismounting the indenter ring from the shaft, snug fit is designated. At the same time, fit between pulley and shaft is also designated with medium drive fit.

FINAL DESIGN MODEL

From the theoretical calculations obtained, the appropriate dimensions for the lubricant test rig are consequently determined. By using SolidWorks 2004, the real scale of design model is established, Figure 4. To ensure that the prototype works within the safely limit, COSMOS analysis is also carried out for verification on the critical parts of the model, such as the pivot lug, the pivot pin, the cantilever and the shaft.

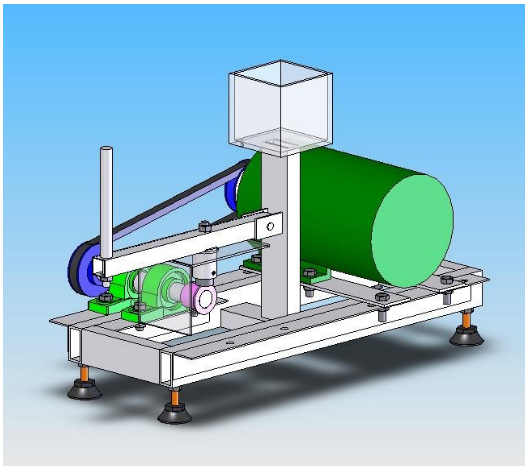


Figure 4. Test rig 3-D model created by SolidWorks 2004.

CONSTRUCTING PROTOTYPE MODEL

Verification of the final model by CAE analysis is preceded by the fabrication of the prototype test rig. A complete master blue-print of the design model, raw materials, and accessories are identified as shown in the Table 2.

Table 2: List of material for prototype fabrication.

Raw material	Dimension (mm)	Quantity	Remarks / Application
Structural steel-angles (W x H x T x L)	38 x 38 x 2 x 575	2	Base
	38 x 38 x 2 x 30	2	Tension lug
	38 x 38 x 2 x 220	1	Base
Plate (W x T x L)	50 x 4 x 280	2	Motor support plate
	50 x 4 x 30	2	Pivot lug
Rectangular channel (W x H x T x L)	25 x 50 x 3 x 150	3	Base
	25 x 50 x 3 x 250	1	Column
	25 x 50 x 3 x 600	2	Base
Square channel (W x H x T x L)	25 x 25 x 1 x 330	1	Cantilever
Solid cylinder (D x L)	50 x 250	1	Shaft
	50 x 60	1	Specimen holder
	16 x 220	1	Weight holder
	9 x 50	1	Pivot pin
	50 x 50	1	Ring (indenter-stainless steel)
Key (W x H x L)	6 x 6 x 25	2	
Pulley	Diameter: 75	2	
Belt	A-45	1	
Electric motor		1	Single phase 0.5hp – 1420rpm
Clevis pin		2	
Pillow bearing		2	To support shaft
Bolts	M10	10	
	M8	2	
Nuts	M10	10	
	M8	2	
Washers	-	14	
Stand	-	4	

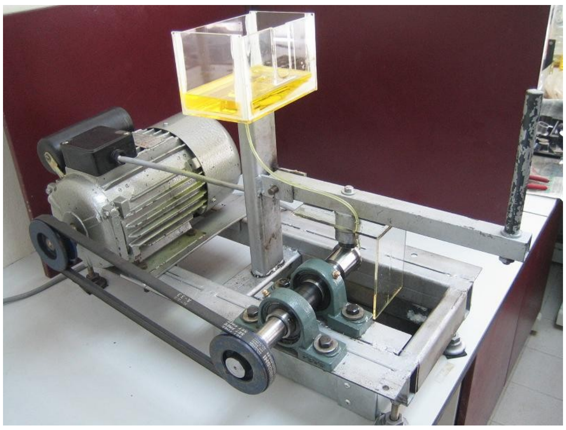


Figure 5. Fabricated test rig prototype.

TESTING AND EXPERIMENT

The ultimate objective of developing the lubricant test system is to facilitate the simulation and the evaluation on the performance of wear and tear behaviour in the engine system. Performance of pure and contaminated lubricant is compared by measuring the degree of wear on the mild steel specimen. Five groups of tests have been carried out for each type of lubricant to check the reliability of this design. The conditions of the tests carried out are summarized as shown below and in the Table 3:

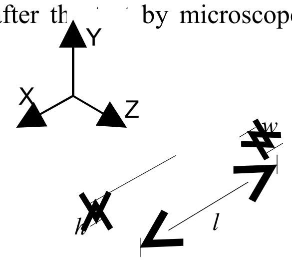
Sliding speed: 3.32 m/s
Test duration: 3 minutes
Temperature: Room temperature
Lubricant type: 20W-50 (new)
20W-50 (used 1000 km)
Lubricant flow rate: 0.0637 liter / hour

Table 3: Mechanical properties of specimen and indenter.

	Material	Poisson's ratio, ν	Modulus of Elasticity, E (N/mm ^{2})	Rockwell B-Hardness
Speci men	Mild steel	0.29	$200(10^3)$	48.1
Indent er	Stainless steel	0.25	$190(10^3)$	76.0

MEASUREMENT OF WEAR

A schematic drawing of a hypothetical wear volume on pin can be illustrated as shown in Fig. 6. The wear volume is a function of three parameters, which are width, length and height, *f (w, l, h)*. The width is equivalent to the thickness of indenter ring, which is known as 3 mm thick. While the two unknown parameter length and height are to be determine after th[illegible] by microscope and mathematical relation, respectively.



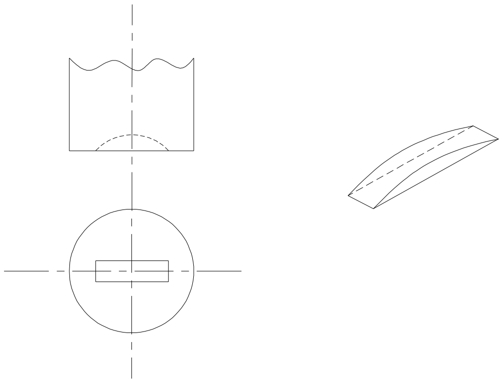


Figure 6: Assumed wear volume.

At X-Y plane, the area of the wear volume is governed by $x^2 + y^2 = R^2$, where R is the radius of the indenter ring. The Fig. 7 illustrates the shape profile and the governing relationship.

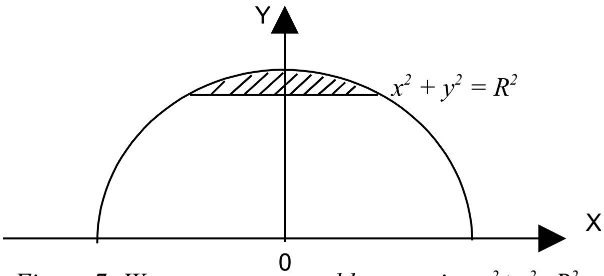


Figure 7: Wear area governed by equation $x^2+y^2=R^2$.

By rearranging the governing equation yields

$$y = \left(R^2 - x^2\right)^{1/2} \tag{12}$$

Thus, the wear depth is simply the term

$$\begin{aligned} h &= R - y \\ &= R - \left(R^2 - x^2\right)^{1/2} \end{aligned} \tag{13}$$

At the same time, the wear area that lay below the curve can be easily determine by integration of

$$A_1 = \int_{-\infty}^{\infty} y \; dx$$

$$A_1 = \int_{-\infty}^{\infty} \left(R^2 - x^2\right)^{1/2} dx \tag{14}$$

where $\infty + |-\infty| = l$ would be the length of the wear scar measured under 10X microscope. Since the shaded area is the actual wear area, the imaginary area, A_2, has to be deducted from the integrated area to yield

$$A_2 = l \times y \tag{15}$$

Therefore, the actual wear volume can be found as

$$V = w \cdot (A_1 - A_2)$$

$$V = w \cdot \left[\int_{-\infty}^{\infty} \left(R^2 - x^2\right)^{1/2} dx - l \cdot y \right] \tag{16}$$

RESULTS AND DISCUSSION

The average wear volume obtained was observed, identified and tabulated as in Table 4. The results of the wear volume under pure and contaminated lubricant are obtained and compared with respect to the Hertzian pressure applied in the tests.

Table 4. Average wear volume.

Max. Hertzian pressure, P_{max} (MPa)	Average wear volume for pure lubricant, V (mm^3)	Average wear volume for contaminated lubricant, V (mm^3)	Order magnitude of wear
39.25	0.2130	0.2992	1.40
55.51	0.3200	0.5046	1.58
67.99	0.6014	0.8039	1.34
78.50	0.7694	1.0816	1.41
87.77	1.1750	2.0831	1.77

The characteristic behaviour of the wear test under both pure and contaminated lubricant oil has shown that wear volume increases with respect to the increasing applied load, Figure 8. However, in contaminated lubricant condition, the wear rate is progressively higher compare to using of pure lubricant in the tests. The order magnitude of wear caused by contaminated lubricant with respect to pure lubricant is in the range of 1.34 – 1.77 times. In the other words, the level of wear preventive for contaminated lubricant is found to be at least 1.34 times lower than pure lubricant. Wear debris in contaminated lubricant has been a cause for the additional abrasive wear.

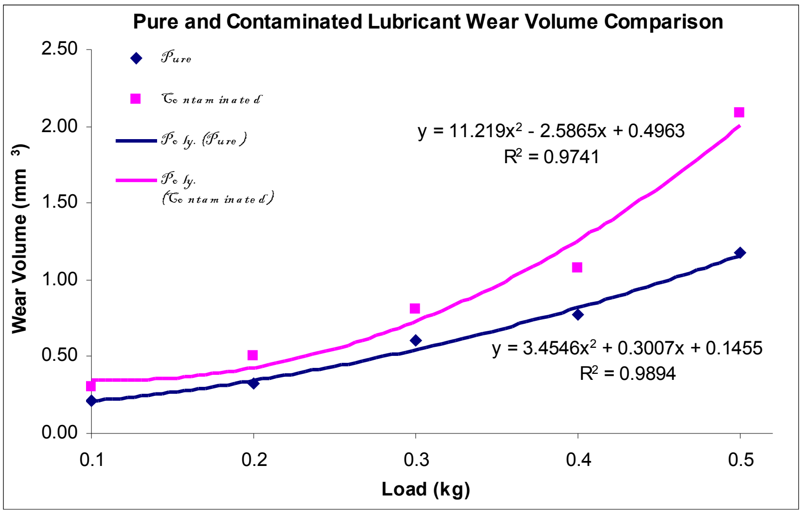


Figure 8. Variation of wear volume under pure and contaminated lubricant conditions

CONCLUSIONS

Tests have been carried out to compare the wear preventives level of 20W-50 pure lubricant and 20W-50 contaminated lubricant (used 1000km). It is found that wear rates are proportional to the load that pressed on mating parts. As expected, the wear volume is more severe when contaminated lubricant condition is introduced. With the presence of wear debris, the specimen experienced additional abrasive wear. Results concluded that the level of wear preventive for 20W-50 pure lubricant is at least 1.34 times higher than 20W-50 contaminated lubricant. Therefore, it is important to replace the engine oil at recommended period of time in order to prolong engine life and retain its maximum performance.

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