International Journal of Current Engineering and Technology E-ISSN 2277 – 4106, P-ISSN 2347 – 5161

Available at http://inpressco.com/category/ijcet

Research Article

Design of Valve System using Optimization Approach

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Accepted 15 April 2015, Available online 22 April 2015, **Vol.5, No.2 (April 2015)**

Abstract

The mathematical optimization techniques require extensive domain (physical understanding and accurate mathematical model) knowledge to design systems performing the desired functions optimally. However lack of domain knowledge makes the whole exercise futile as the mathematical optimization in such form does not account for all the possible means to accomplish the desired objectives. The situation becomes more complex due to the nature of objectives which are often contradictory. In such situations experimental optimization using Design of Experiments is applied which requires intensive experimentation on the prototype of the designed system. However, at the concept generation stage of the design process, the use of optimization approach which is based on the concept of increasing the ideality of the system is preferable. It ensures that the contradictions are resolved and the harmful functions are identified and reduced. In the present paper the characteristics of the optimization approach is demonstrated by considering the design of a mechanical system. As a case study design of valve system is carried out. The disadvantages of the conventional optimization approach are highlighted and a systematic and scientific procedure for design of valve system using optimization approach is presented.

Keywords: *Design, tribology, optimization, overhead camshaft, valve*

1. Introduction

The design of mechanical system that performs its desired functions satisfactorily, efficiently and economically is the key to survival in the competitive environment. The mechanical systems must be designed in such a way that it fulfils the desired objectives in the most efficient manner and with the use of minimum resources. The mathematical optimization methods provides the means to achieve these objectives by minimizing or maximizing some defined objective functions under a given set of constraints. The mathematical optimization techniques have been used extensively in the design of systems. However these methods requires comprehensive domain knowledge for its mathematical modelling because many of the desired system objectives are contradictory in nature. The mathematical optimization under these conditions is therefore difficult to achieve. Consider, for example, the multi-objective optimization of journal bearing [Hirani, 2005], wherein simultaneous minimization of power loss and oil flow was carried out subjected to the constraints on film thickness, film pressure and temperature rise between the bearing surfaces. The radial clearance, L/D ratio, oil groove location, feed pressure, and the oil viscosity were considered as the design variables. Based on this study Hirani [Hirani, 2005], however observed that the optimization cannot be performed using analytical approach, because it mostly predicts lower maximum pressure and temperature rise and excessive oil flow. Moreover the optimization process is unable to distinguish the sensitivity of the various journal bearing parameters. It also does not incorporate the operational aspects like shaft misalignment, effect of wear on the geometry and dimensions of the bearing which change the bearing clearance, change in the surface roughness parameters of the bearing and journal during the use etc.

These shortcomings can be easily overcomed by the use of experimental optimization techniques. The experimental optimization establishes relationship between different system parameters and suggests their robust combination. It is effectively utilized for the determination of sensitivity of the various design and operational parameters. However this method also have many limitations: it requires system prototype, more number of tests and not all parameters can be varied during the experiments. Mnay a times no definitive pattern or trend is observed for the various parameters. The experimental error also needs careful consideration. Consider, for example the case of a journal bearing for which a suitable lubricant needs to be identified [Muzakkir *et al*, 2012]. In this study several journal bearings were fabricated for testing on fully automated Journal Bearing Test Rig shown in

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Figure 1. The bearings were fabricated as per the drawing given in Figure 2.

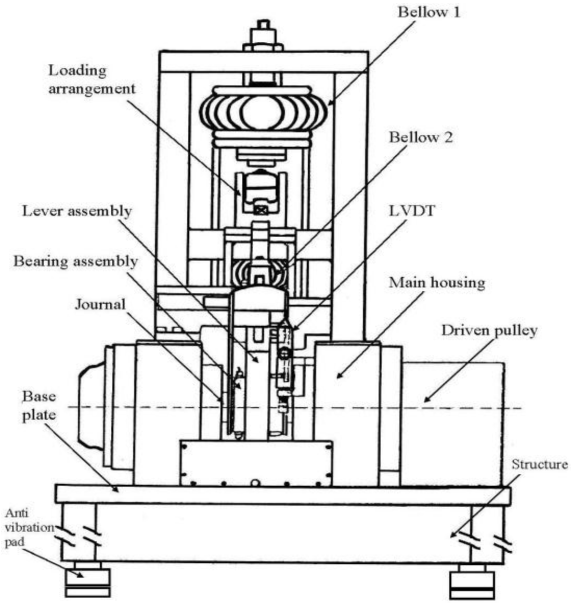


Figure 1 Journal bearing Test Rig [Muzakkir *et al*, 2012]

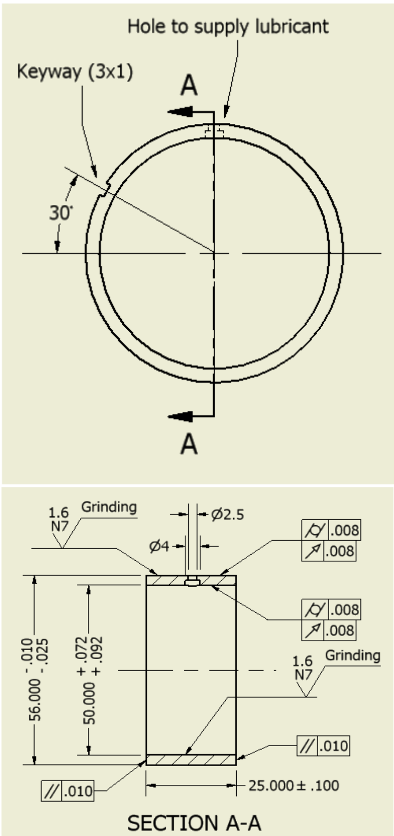


Figure 2 Details of the bearing fabricated for testing [Muzakkir *et al*, 2012]

The load, speed and lubricant temperature were taken as 5000N, 10 rpm and 75°C respectively. Each test was continued for 5 hours to obtain measureable wear. The Taguchi method was employed for the planning of experiments and analysis of test results. The factors and their levels selected for conducting the experiments is listed in table 1.

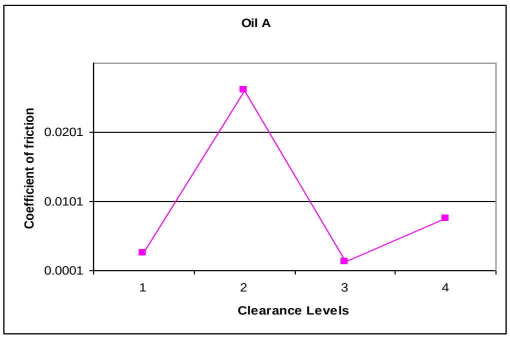
Table 1: Test factors and their levels [Muzakkir *et al*, 2012]

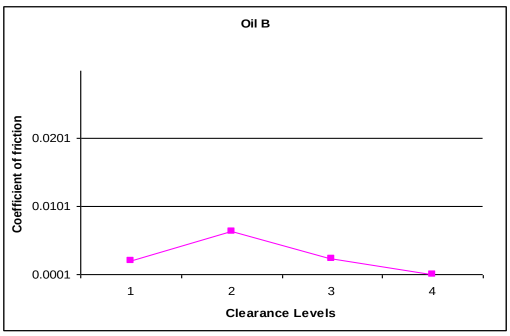
	Levels			
Factor	Level 1	Level 2	Level 3	Level 4
Clearance (micrometers)	41	42	43	44
Lubricating Oil	Oil type A	Oil type B	Oil type C	--

The Taguchi orthogonal array for determining the interaction between the different factors is given in table 2. The results of 12 experiments in terms of friction and wear are also given in table 2.

Table 2 Orthogonal Array [Muzakkir *et al*, 2012]

S. No.	Clearance [C]	Oil [O]	[C][O]	[C]² [O]	[C][O]²	Coefficient of friction	Wear (gms)
1	0	0	0	0	0	0.002030	0.01910
2	0	1	2	0	0	0.002530	0.00200
3	0	2	1	0	0	0.001090	0.00180
4	1	0	0	0	0	0.006400	0.00390
5	1	1	1	2	1	0.026090	0.00200
6	1	2	0	2	1	0.001460	0.00180
7	2	0	2	2	1	0.002470	0.02200
8	2	1	1	2	1	0.001320	0.00020
9	2	2	2	1	2	0.002060	0.00310
10	3	0	1	1	2	0.000170	0.00510
11	3	1	0	1	2	0.007620	0.02000
12	3	2	2	1	2	0.001150	0.00320





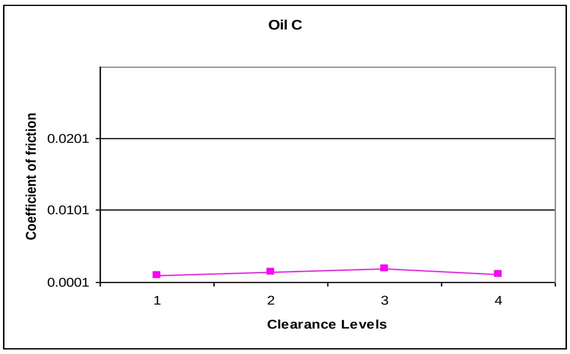


Figure 3 Variation of coefficient of friction at four clearance levels [Muzakkir *et al*, 2012]

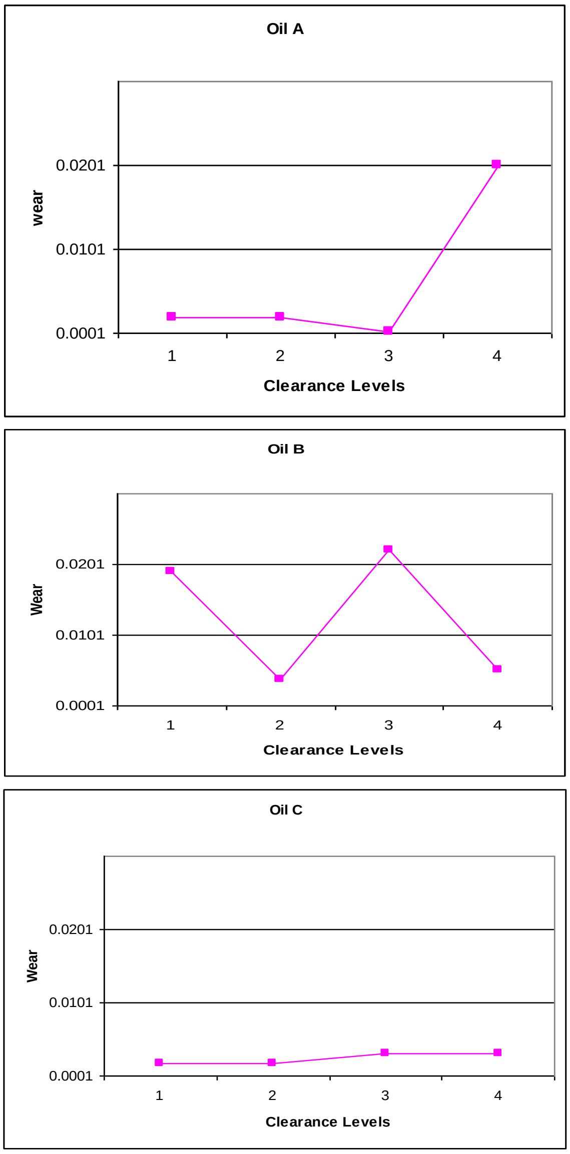


Figure 4 Variation of wear at four clearance levels [Muzakkir *et al*, 2012]

The results obtained were analyzed for determination of the direct and combined effect of the factors and the contribution of each factor on the coefficient of friction and wear was determined using the Taguchi Orthogonal array. The results are tabulated in Table 3.

Table 3 Contribution of factors on coefficient of friction and wear in% [Muzakkir *et al*, 2012]

	Clearance	Oil Type	Combined effect (Clearance * oil)	Combined effect (Clearance^2 * oil)	Combined effect (Clearance * oil^2)
Effect on Coefficient of friction (%)	41.78	42.47	10.21	2.76	2.76
Effect on Wear (%)	50.29	5.14	1.00	21.78	21.78

Based on these experimental observation and analysis it was concluded that the clearance has a negligible effect on the wear and coefficient of friction of the journal bearing when oil C is used. The coefficient of friction slightly increases and then decreases with the increase in the clearance when Oil B is used. The combined effect of clearance and oil type is significant on the wear however the combined effect of clearance and oil type is not significant on the coefficient of friction. These observations do not suggest a definitive solution to the given problem. Therefore the experimental optimization does not effectively provides an optimum solution.

Considering these limitations of the conventional optimization methods, a better optimization approach is proposed in this paper.

2. The Optimization Approach

The proposed optimization approach is based on the concept of ideality. It has emerged as a very useful tool in the optimum design of mechanical systems that generates feasible solutions. Ideality is defined as the benefit to cost ratio and expressed as:

$$\text{Ideality} = \frac{\sum \text{Useful Functions}}{\sum \text{Harmful Functions} + \sum \text{cost}} \quad (1)$$

Useful functions are desired functions from a product, system or process. The harmful functions include undesirable functions such as: weight, friction, misalignment, noise, vibration, wear, etc. As every harmful function costs the society, therefore it is necessary to quantify it. The main thrust of a designer must be in increasing the useful functions, but it requires more concentrated efforts and is generally time consuming. Therefore the attention is focussed in minimizing the harmful functions by the suitable design of the system and by incorporating the best available technologies.

If the focus is on minimizing the initial cost then it may so happen that the running cost/maintenance cost may increase rendering the whole exercise futile. As a case study, design of valve system is considered in the present work. The valve system comprises of a valve, cam, follower, spring and cam shaft. It is much more complex as compared to the journal bearing. The optimization of each element by the process of reducing their harmful effects and increasing the ideality would result into an optimum valve system.

3. Case study: Design of the Valve System

The main considerations in the design of the DOHC system is the minimization of friction and wear of direct acting overhead camshaft valve train. This problem is restated in the form of Functional Requirement (FR):

FR_1 = Minimize friction between cam-follower
FR_2 = Minimize wear between valve and valve guide
FR_3 = Minimize wear between valve and valve seat

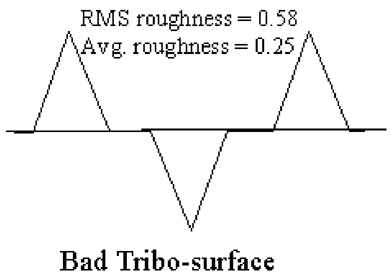
All three FRs are negative functions and need to be minimized without impairing cost and any of useful functions. To quantify coefficient of friction between cam-follower, a mixed lubrication approach [Katoh & Yasuda, 1994] is used, such as:

$$f_{cam-follower} = f_{liquid} . X_{liquid} + f_{boundary} X_{boundary} \quad (1)$$

The coefficient of friction due to fluid film (hydrodynamic/elastohydrodynamic) f_{liquid}, is generally lesser than one percent of the coefficient of friction due to boundary lubrication, $f_{boundary}$. Therefore always main emphasize is to avoid/minimize boundary contact. This contact can be reduced by increasing the dimensionless film parameter Λ (often referred as specific film thickness), defined as:

$$\Lambda = \frac{h_{\min}}{\sqrt{R_{rms,c}^2 + R_{rms,f}^2}} \quad (2)$$

$R_{rms,c}$ is root mean square (rms) surface roughness of cam-surface, and $R_{rms,f}$ is rms surface roughness of follower surface. Interestingly here rms value is used, while arithmetic average roughness is more common in manufacturing industry. To clarify this, let us examine Fig. 7.



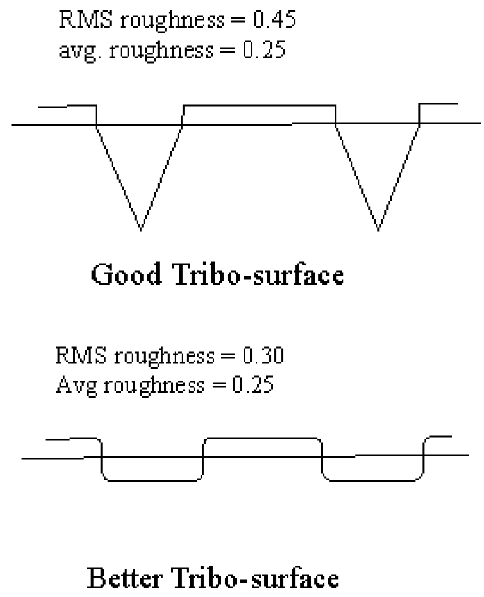


Figure 7 Comparison between arithmetic mean and root mean square roughness

The measurement of average roughness imposes a linear penalty on all points whether a point is too close to nominal line or too far. However, rms roughness parameter uses square term. If there are three points: A one unit, B two units and C three units away from nominal line. RMS roughness parameter put penalty of one, four and nine on points A, B, C respectively. Therefore rms value is a better roughness parameter compared to average roughness. From tribological point of view, a surface without any asperity but with a number of depressions is always preferred.

To maximize specific film parameter, Λ, designer needs:

- To reduce the rms surface roughness of both the surfaces. Katoh and Yasuda [1994] reduced the friction level of direct acting valve train by improving the surface roughness of cam and follower.
- To increase minimum film thickness. Increase in minimum film thickness can be achieved by lowering the imposed load. Imposed load consists of inertia load of valve, follower assembly and spring; spring load and friction at various surfaces. Reduction in spring load (preload and spring rate) and solid friction lowers the load on the cam surface. However, inertia loading due to acceleration has dual role. As shown in Fig. 8, inertia loading changes it sign on reversal of acceleration. During positive acceleration inertia loading impose load on the cam, while during negative acceleration it reduces the cam load.

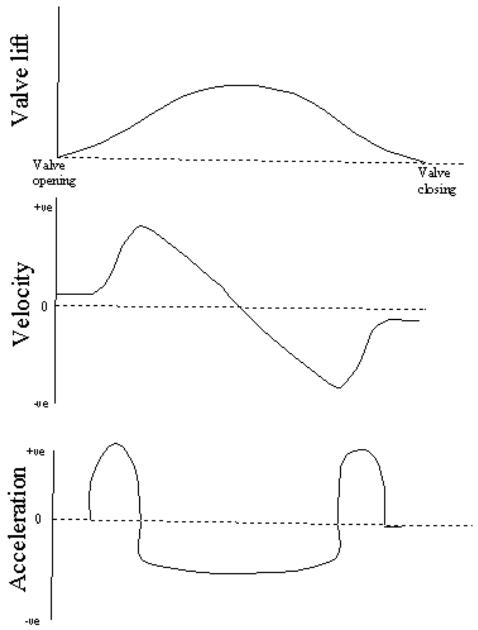


Figure 8 Displacement, velocity and acceleration of cam

Ball *et al* [1989] performed parametric study and conclude an increase in reciprocating mass causes increase in the film thickness, therefore chances of getting full film thickness will be higher at high load. To clarify this, once again let us consider Fig. 8. This figure shows zero velocity at maximum lift position. As velocity is essential to develop hydrodynamic or elastohydrodynamic effect, film thickness between cam and follower will be zero and boundary lubrication will dominate. Some relief is provided by reduction in cam load due to negative inertia of large reciprocating mass. This brings a contradiction: designer needs high reciprocating mass to increase the minimum film thickness at nose radius, while low reciprocating mass is required to cut down the power loss at cam flank-follower interface.

Careful study indicates that cam-follower interface is subjected to dynamic load. Under dynamic load, an additional lubrication mechanism, named as squeeze film dominates particularly when two surfaces are approaching each other. Under dynamic load condition, it is probable that squeezing out of lubricant takes more time, the load will reverse before the separating film breaks down and surfaces actually touch. This regime of squeeze film resolves the designer's contradiction of low and high inertia loading.

To summarize abovementioned paragraphs, friction loss can be reduce by providing better surface roughness (lower rms value and that allows retaining lubricant) for cam and follower, lower spring preload and spring stiffness. Further, if it is assured (accounting squeeze film effects) that minimum film thickness will be 2 to 3 times higher than composite surface roughness than lower inertia loading will help to reduce the friction between cam and follower interface.

Fukuoka *et al*. [1997] developed a lighter valve DOHC train using an aluminium tappet, an aluminium spring retainer and a thin sintered shim to reduce the inertia loading on cam. They concluded a 40% reduction in friction. Recently Gebauer and Gavrilescu [2004] manufactured a hollow valve for automotive engine to reduce the inertia loading on cam.

In summary, FR_1 (Minimize friction between cam-follower) has two sub-functions. First sub-function, $FR_{1.1}$, requires improvement in surface roughness of cam-follower interface, while second sub-function, $FR_{1.2}$, demands reduction in cam loading. This second sub-function has two sub-functions: $FR_{1.2.1}$ states reduction in inertia loading, while $FR_{1.2.2}$ emphasizes on reduction in spring load. $FR_{1.2.1}$ has five sub-functions: $FR_{1.2.1.1}$ = Reduction in weight of poppet valve, $FR_{1.2.1.2}$ = Reduction in weight of spring retainer, $FR_{1.2.1.3}$ = Reduction in weight of shims. Reduction in weight of spring is not accounted as a sub-function of $FR_{1.2.1}$ as reduction in spring contributes very little (only 30%) and further spring designed for lower stiffness and preloaded can handle this weight reduction.

Minimize wear between valve and valve guide, which is the second FR of the present problem, can be achieved successfully by changing the mechanical field to magnetic field. Nowadays magnetic materials are available which show good strength even at 200°C. One such magnetic valve-guide is shown in Fig. 9. Cylindrical permanent magnetic generally align the valve stem at the center. Axial motion of valve is controlled by cam and follower arrangement.

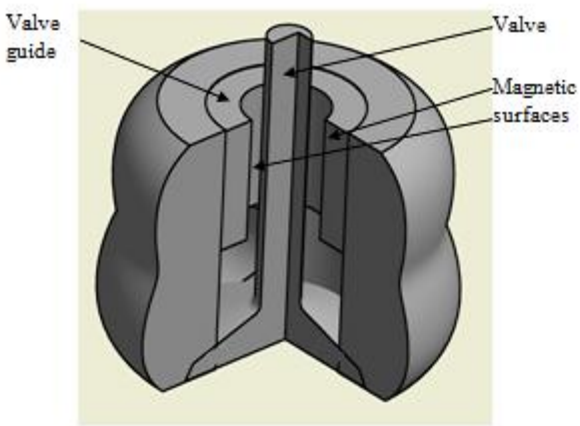


Figure 9 Magnetic valve-guide

Constraint of using unleaded petrol has aggravated problem of Minimizing wear between valve and valve seat. Function of lead to acts an EP additive and reduce the wear between valve and valve seat. This wear is caused [Lewis & Joyce, 2001] by valve closing velocity (Impact wear), combustion load (sliding wear) and valve misalignment. Misalignment can be avoided using self centering magnetic valve guide. Sources of impact wear are impact velocity and valve mass. Sliding (due to high combustion pressure) wear can be reduced by increasing the stiffness of valve head, using wear resistant material and improving lubrication. Generally increasing stiffness of valve head increases the mass of valve. However increase in valve mass increase the chances of impact wear. One obvious choice is use material which is light and stiff, or in other word chooses a material that has high Young's modulus and low density. High value of valve-stiffness is required during closed valve is condition when combustion pressure is high. During valve-opening to valve-closing, a low weight valve is desirable. This can be achieved using a variable stiffness structure, such as hollow valve filled with magnetorheological fluid, which is light and show high stiffness at combustion time [Muzakkir and Hirani, 2015]. MR fluids show a transition from a liquid behavior to a solid one, upon application of a magnetic field. The applied magnetic field intensity is controlled by the electric current supplied to electromagnet(s).

Conclusions

The optimization approach involving the concept of ideality is presented. This approach was applied to the valve system. Out of the several valve systems the direct acting overhead camshaft (DOHC) valve syatem was identified to be best. The conceptual design of DOHC is expressed as a problem statement to reduce friction and wear of direct acting overhead camshaft valve trains. The application of the optimization approach to the DOHC by following tribological and other conventional design considerations have resulted into the following:

- Better design of cam that enhance the squeeze film action at the cam-nose and follower interface.
- Reduce reciprocating weights.
- Provide better rms surface roughness to cam and follower to ensure retainability of lubricant by surfaces.
- Reduce spring preload and stiffness.
- Provide permanent magnetic bearing support that acts as a valve guide.

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