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Development of Methodology for Selection of Best Tribological Parameters for Hermetic Reciprocating Compressor

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Abstract: It is important to decide whether compressor is capable to take load or not. So for proper working of compressor lubrication system must be consistent in performance. To achieve such type of lubrication system we have to optimize lubrication parameters and factors contributing in lubrication. In hermetic compressor the selection of oil type, oil quantity, fit between bearing and shaft, material pair (shaft material), oil viscosity etc. are very important. The problem is occurred while selecting these parameters because these parameters are selected on the basis of trial and error method. There is no exact method to know which parameter has to be selected in order to get optimized performance of compressor. That's why we have to develop such methodology which will overcome above problem. In particular the methodology will be developed for best selection of tribological parameter for hermetic reciprocating compressor .This will overcome trial and error approach & it will give optimum tribological parameters.

Keywords: Reciprocating Compressor, Tribological Parameters, Tare Test, Best Selection, Methodology.

I. INTRODUCTION

A hermetic reciprocating compressor having major contribution in energy consumption of the refrigerator. Therefore it is main area for improvement of compressor performance which results into less overall power consumption of the refrigerator. There are different losses in compressor like thermodynamic, mechanical, electromotor etc. which affect the compressor performance. In this paper attempt is made to develop a methodology for selection of tribological parameters which directly related to above losses. Basically more focus was done on frictional (mechanical) losses. The main factors affecting the performance of the compressor are bearing design, oil type, frictional losses, material type, and geometrical parameters involving in compressor. Developing methodology for selection of best tribological parameters is the main objective of this study.

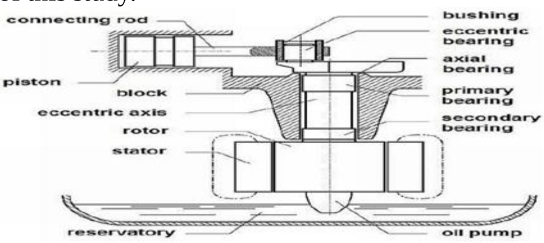

Fig 1

In hermetic reciprocating compressor, oil supply system is usually composed of an oil pick up tube, crankshaft holes and oil grooves. The oil pick up tube is immersed into the oil sump, at the bottom of the shell. During operating process, the oil is pumped into the oil pick up tube and delivered to the shaft bearings, connecting rod and piston. In hermetic compressor the selection of oil type, oil quantity, fit between bearing and shaft, material pair (shaft material), oil viscosity etc. are very important. The main problem is that these parameters are selected by trial and error method. Today there is no formula to calculate exact quantity of oil required in the sump. There is no particular formula or method to choose above tribological parameters. In addition with theoretical analysis, engineering software & practical analysis is also important to validate the results. The main objective of this study is to find out solution for above mention problems. For that it is very necessary to understand the findings of previous researchers and some notification regarding them.

A. RefikÖzdemir, ErhanKasapoğlu, BilginHacıoğlu, Mustafa Duyar [1]studied the effect of crankshaft geometry, bearing clearance, lubricant viscosity on the frictional losses. They did the detailed parametric simulation of compressor by modifying the existing design with new selected parameters. The result of numerical analysis has shown

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that the numerically calculated mechanical loss level is similar to the performance results measured in the calorimeter test system. S. Boyde et al [2]based on the performance of a hermetic reciprocating compressor using HFC-134a and linear-type POEs oil of ISO VG 22, carried out the experiments on a calorimeter to investigate the performance comparisons between this compressors using different viscosity of linear-type of POEs oil. Kobayashi M., Hayashi A. and Yoshimura T. [3] compared the effects of bearing material, chemically treated shafts, bearing stiffness and refrigerant solubility in the oil to the characteristics of journal bearings lubricated with a mixture of HFC-134a and lubricating oil.Stachowiak and Batchelor [4]the bearing friction loss was estimated using a regression equation, which was derived from a sample of several hundred theoretically, evaluated journal bearings. The regression equation takes the geometrical data, bearing operation data, and the viscosity data as arguments and delivers an estimation of the bearing power loss. The regression can be written as:

H = 3.9307 *103*V1-0.706 *V21.577*L0.477* D2.240* N1.287* c0.249*T-0.204*(1+ lnW*) 1.324

II. PREPARATION OF CHARACTERISTIC MATRIX FOR DIFFERENT TRIBOLOGICAL PARAMETERS.

Different parameters are like oil type, oil quantity, oil viscosity, clearance between shaft and bearing are varied to check its effect on power loss. Three bearings are considered during calculation such as main journal bearing, sub journal bearing, and upper journal bearing respectively for Compressor Model 1 and Compressor Model 2.

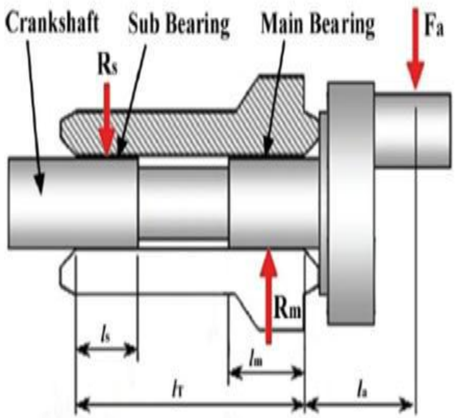

Fig 2.Crankshaft of Compressor

The characteristic matrix is prepared by using above mention tribological parameters. In this matrix dimensions of bearings and shafts are varied according to trial. Diameter of bearing 1&2 is same while bearing 3 having different diameter .Similar case for clearance between shaft and bearing. In this study the performance parameters are frictional power loss, max. temperature ,and total oil flow . Different trials are taken by varying the Oil Viscosity, diameter of journal, clearance between shaft and bearing to see the effect on frictional power loss, max. temperature in bearing and total oil flow. Five different trials are taken in this study with five different viscosities. Below is the trial with viscosity of 22 cst likewise different trials are taken by varying the viscosity accordingly 32cst, 42 cst, 10 cst, 15 cst.

					Max Temp.(Tmax) °C			Total oil flow(Q) m^3/s			Total oil flow
d For1-2 JB(m)	d For U JB (m)	Clearance for 1-2 JB (m)	Clearance for U. JB (m)	Power loss(H) Watt	T1	T2	T3	Q1	Q2	Q3	Q=Q1+Q2+Q3
0.015984	0.013972	0.00002286	0.0002794	49.5	96.124	90.6863	94.1539	2.56E-08	3.99E-08	2.92E-08	9.47E-08
0.014	0.012	0.00002286	0.0002794	32.9	94.342	89.9899	91.65	2.16E-08	3.29E-08	2.66E-08	8.11E-08
0.0145	0.0125	0.00002286	0.00002794	36.7	94.092	90.358	92.246	2.38E-08	3.38E-08	2.73E-08	8.49E-08
0.015	0.013	0.00002286	0.00002794	40.6	94.753	89.999	92.869	2.44E-08	3.80E-08	2.79E-08	9.03E-08
0.0155	0.0135	0.00002286	0.00002794	45	95.438	90.342	93.519	2.50E-08	3.90E-08	2.86E-08	9.26E-08
0.0165	0.0145	0.00002286	0.00002794	54.8	95.4378	91.064	94.897	2.88E-08	4.10E-08	2.99E-08	9.97E-08
0.017	0.015	0.00002286	0.00002794	60.2	96.098	91.442	95.627	2.95E-08	4.20E-08	3.06E-08	1.02E-07
0.015984	0.013972	1.65E-05	2.67E-05	56.7	95.1386	89.9275	94.56	1.74E-08	2.73E-08	2.72E-08	7.19E-08
0.015984	0.013972	1.78E-05	2.79E-05	54.7	95.393	90.104	94.511	1.89E-08	2.97E-08	2.92E-08	7.78E-08
0.015984	0.013972	1.91E-05	2.92E-05	52.8	95.617	90.266	94.461	2.04E-08	3.22E-08	3.13E-08	8.39E-08
0.015984	0.013972	2.03E-05	3.05E-05	51.1	95.816	90.418	94.421	2.20E-08	3.48E-08	3.35E-08	9.03E-08
0.015984	0.013972	2.16E-05	3.18E-05	49.5	95.982	90.555	94.379	2.38E-08	3.75E-08	3.59E-08	9.72E-08
0.015984	0.013972	2.29E-05	3.30E-05	47.9	96.12	90.686	94.344	2.56E-08	3.99E-08	3.83E-08	1.04E-07
0.015984	0.013972	2.41E-05	3.43E-05	46.5	96.249	90.793	94.304	2.75E-08	4.34E-08	4.10E-08	1.12E-07

Table I Characteristic Matrix

The above trial is for no load conation. In no load condition we didn’t consider load acting on bearing. All the calculations are done by referring ESDU book.

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If we have to find out power loss in bearing in running condition with load acting on it then we can make use of phenomenon of temperature rise in bearing. Because when load is acting on bearing then its temperature going to be rise .if we take the viscosity at the temperature in our calculation its nothing but the calculation of power loss of bearing with load acting on it. The viscosity at that temperature can be found by using Daniel chart. Similar calculations are done by taking viscosity at working temperature. After finding the characteristic matrix for compressor model 1 similar procedure is applied for compressor model 2.

III. ESTABLISHING INTER-RELATIONSHIP AMONG VARIOUS PARAMETERS

For finding the inter-relationship between various parameters like oil type, oil viscosity, fit between crankshaft and bearing, material first off all we have to decide best pair from the above matrix. For that we have to apply some criterion on the basis of which we can easily choose the pair.

3.1Material Pair Selection- The most important parameter while selecting material is the coefficient of friction and wear.

3.2 Minimum oil film thickness (ho) - In order to achieve efficient operation of compressor there should be sufficient oil film thickness in between journal and bearing. As load goes on increasing oil film thickness goes on decreasing. So it is necessary to maintain minimum oil thickness at high load.

3.3 Selection of Best Pair of Compressor Model 1& Model 2

From above table it is found that if we increase diameter of shaft then frictional power loss in bearing also increases. That is H is directly proportional to Power loss. If we increase the clearance between shaft and bearing then frictional power loss in bearing goes on decreasing. We have to select Pair which has less frictional power loss.

From calculation it is cleared that the blue coloured pair has less power loss than existing pair. We can't go for less diameter because again the question of shaft strength and can't go for higher clearance it will create shaft deflection problem. So that within range for same diameter if we vary the clearance we get less frictional power loss than existing model.

So we select pair for
Model 1-diameter for lower bearing 0.6293 inch (0.015984 m) and for eccentric bearing 0.5501 inch (0.013972 m) and clearances 0.9 thou (0.00002286 m) for lower bearing and 1.3 thou (0.00003302 m) for eccentric bearing.

Model 2 – diameter for lower bearing 0.7511 inch (0.019077 m) and for eccentric bearing 3 is o.7143 (0.01814 m) inch and clearances 0.6 thou (0.00001524 m) for bearing 1,2 and 0.9(0.00002286 m) thou for eccentric bearing.

IV. EXPERIMENTAL TESTING OF PAIRS OF COMPRESSOR MODEL 1 AND MODEL 2

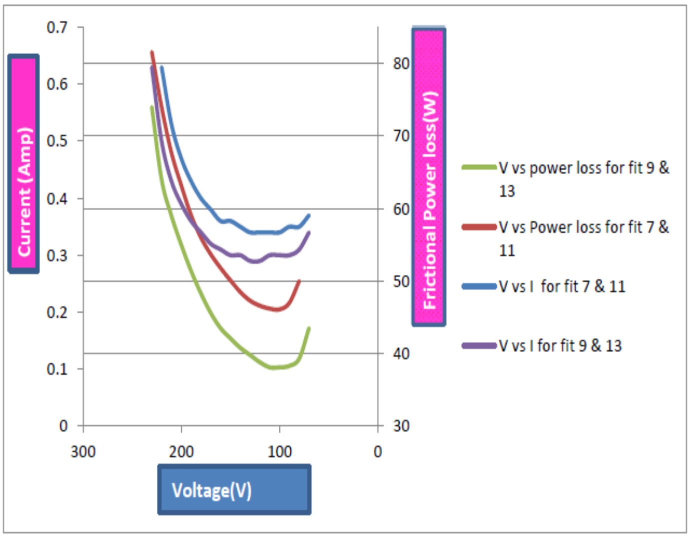


Fig. 4 Graphical Representation of Testing Result

The above selected pairs are tested experimentally to check actual frictional power loss. For that tare test is taken to find out the frictional power loss for selected pairs. These pairs are tested at no load condition.

Dia. of JB 1&2 (m)	Dia. of JB 3(m)	Clearance of JB 1 & 2 (m)	Clearance of JB 3 (m)	Pract. Power loss(W)
0.015984	0.013972	0.0000178	0.00002794	81.6
0.015984	0.013972	0.0000226	0.00003302	74

Table II Model 1 Testing Result

Dia. of JB 1&2 (m)	Dia. of JB 3(m)	Clearance of JB 1 & 2 (m)	Clearance of JB 3 (m)	Pract. Power loss(W)
0.019077	0.018 14	0.0000127	0.0000229	103
0.019077	0.018 14	0.0000152	0.0000229	100
0.019077	0.018 14	0.0000152	0.0000254	94.3

Table III Model 1 Testing Result

V. MATHEMATICAL MODELLING FOR THE BEST SUITABLE TRIBOLOGICAL PARAMETERS.

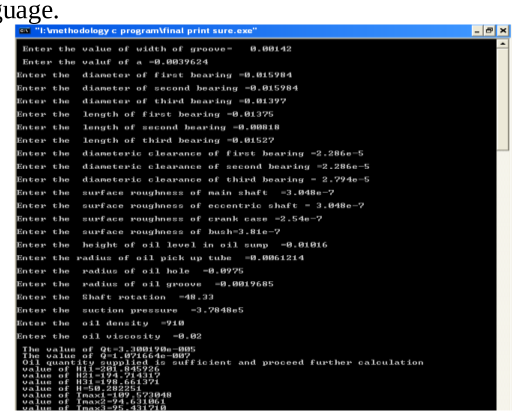
There is nonlinear relationship in between all parameters so the model is nonlinear. The equation H $= H'*\eta_e * N^2 * d^2 * b * \frac{d}{C_d}$ represents the frictional power loss. All the parameters are known except H' which is calculated from graph H' vs. ε forfor given b/d ratio. We prepared table for different b/d ratio with the help of polynomial equation. X is the variable parameter in polynomial equation. So if we vary b and d then for that b/d ratio value of H' is calculated by using LOOK UP function in excel. Since there are three bearings in our project so threedifferent tables are prepared .One column shows b/d value and other column shows H' value.First value of eccentricity ratio (X) is calculated. Then b/d ratio is calculated. By using LOOK UP function we select b/d value in table, corresponding to that b/d value of H' is calculated.

Similar procedure is followed for other bearings. So total frictional power loss calculated by using following formula.

$$H = + H_1'*\eta_e * N^2 * d1^2 * b1 * \frac{d1}{C_{d\,1}} + H_2'*\eta_e * N^2 * d2^2 * b2 * \frac{d2}{C_{d2}} + H_3'*\eta_e * N^2 * d_3^{\,2} * b_{31} * \frac{d_3}{C_{d3}}$$

VI. DEVELOPMENT OF METHODOLOGY FOR MATHEMATICAL MODELLING

Software is developed from mathematical modelling by using C language for comparing performance of two compressors. It will help us to select the optimum tribological parameters. Here performance of compressor is measured in terms of frictional power loss. This will help us to check the effect of different tribological parameters on the performance of compressor. This methodology is applicable to all compressor models. Below is the output of program developed by using C language.



VII. DISCUSSION ON THE RESULT

A method to find out frictional power loss due to journal bearings has been presented.Simultaneously three bearings performance can be analyse with the help of above methodology. The methodology which is developed by using 'C' language is much efficient compare to conventional method. We can compare two compressor models by using above methodology. It will helpful for selecting appropriate parameters to improve performance of compressor. For comparing performance, two different compressors are selected with different dimension. By using 'C' programming language two different sets of input variables are prepared. These inputs are taken from user as per their requirement. By giving two different sets of inputs we had compare their performance on the basis of frictional power loss and oil quantity required for three bearings.

Dia. of JB 1&2 (m)	Dia. of JB 3(m)	Fit for JB 1 & 2 (m)	Fit for JB 3 (m)	Theor. Power loss	Practical Power loss(W)
0.015984	0.013972	0.00001778(7 thou)	0.00002794 (11 thou)	54.7	81.6
0.015984	0.013 972	0.00002286(9 thou)	0.00003302(13 thou)	47.9	74

Table III Comparison bet^{n} theoretical and practical power loss for Model 1

Dia. of JB 1&2 (m)	Dia. of JB 3(m)	Fit for JB 1 & 2 (m)	Fit for JB 3 (m)	Theor. Power loss	Practical Power loss(W)
0.019077	0.01814	1.27E-05 (5 thou)	2.29E-05 (9 thou)	72.5	103
0.019077	0.01814	1.52E-05 (6 thou)	2.29E-05 (9 thou)	68.8	100
0.019077	0.01814	1.52E-05 (6 thou)	2.54E-05 (10 thou)	67.5	94.3

Table I Comparison bet^{n} theoretical and practical power loss for Model 2

Reason for large difference between theoretical and practical value is that whatever calculations are done theoretically it is only showing power loss due to crankshaft bearings. We don't consider the power loss due to piston-cylinder & motor in theoretical calculation. While practical value contributes effect of all parameter. That's why there is big difference in between practical value and theoretical value. It is observed that whatever calculations are done theoretically same pattern observed practically.

VIII. CONCLUSION

There is no separate methodology or software which can give the prediction of performance of compressor. Previously all the calculations are done manually that was time consuming. So attempt is made to develop a methodology which can help us for selection of best tribological parameters in less time &which will help to improve the performance of compressor. Theoretical trials get validated experimentally. The reason for difference between theoretical and practical result is induced due to piston cylinder and motor wattage. The relationship between frictional power loss due to bearings and tribological parameters of compressor has been established. It is observed that by minimizing clearance between journal and bearing there is increase in frictional power loss. Similarly with the increase in oil viscosity there is increase in frictional power loss. From all the data and formulas collected during project mathematical model is established. Software is developed by using mathematical model. Now it is easy to predict the performance of compressor within few minutes. If we want to change the geometry of compressor i.e. length, diameter or clearances to check effect on performance of compressor first off all we have to manufacture crankshaft according to that dimension then testing will be carried out. It will consume lot of time and also this process is costly. In order to minimize practical iteration so as to save time and cost this software will be helpful. This software will help us in selection of best tribological parameters.

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