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Improving the Reliability and Extending the Application Range of Reciprocating Refrigeration Compressors by Using a Modern Bearing Calculation Method

R. Vauth

H. Kruse

IMPROVING THE RELIABILITY AND EXTENDING THE APPLICATION RANGE OF RECIPROCATING REFRIGERATION COMPRESSORS BY USING A MODERN BEARING CALCULATION METHOD.

R. VAUTH, Dipl.-Ing.   Dr. H. KRUSE, Professor of Mechanical Engineering

Technical University of Hannover / Germany

INTRODUCTION

The application field of reciprocating refrigeration and heat pump compressors is principally limited by

1. the maximum condensing temperature or the greatest permissible condensing pressure p_{max} as given by the material stresses in the compressor body
2. the pressure difference across the compressor between the suction and discharge pressures which introduces great mechanical loads on the crankshaft assembly. Especially the maximum permissible loads on the crankshaft bearings limit this pressure difference to the maximum value Δp_{max}
3. the gas temperature in the discharge valve t_{hVmax}
4. the starting torque of the electrical motor M_{dAM}

The application limits are clearly illustrated in Fig. 1 /1/ and are obtained by plotting the cooling capacity against the condensing temperature t with the evaporating temperature t_0 as parameter.

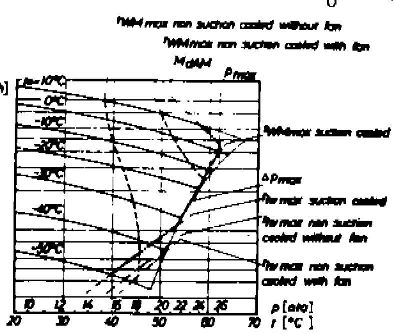

Fig. 1: Application field and limits of a reciprocating compressor /1/

Reciprocating refrigeration and heat pump compressors are often working under operating conditions very near to the described application limit Δ_{pmax}, as given by the maximum permissible load on the crankshaft bearings. Thus the loads of the hydrodynamically lubricated crankshaft bearings mainly determine the application limits of reciprocating compressors under those operating conditions.

In order to improve the reliability of those compressors a greater distance of the operating conditions from the application limit is favourable. This can be achieved by extending the application range to greater permissible bearing loads, which can only be done if a better prediction of bearing reliability can be made.

Because of the great number of design parameters and running conditions as for instance different evaporating and/ or condensing temperatures, different running speeds or cylinder unloading for capacity control purposes, for this prediction of reliability concerning bearing material fatigue, wear and cavitation, a calculation method is necessary to predict sufficiently the realistic behaviour of crankshaft bearings in refrigeration and heat pump compressors.

MODERN BEARING CALCULATION METHODS

Bearings in the crank assembly of reciprocating engines in general are dynamically loaded. They have to carry loads varying in magnitude and direction during a working cycle of the machine.

Those loads are given by the instationary gas and mass forces in the crank assembly.

A result of the unsteady bearing load is an instationary location of the journal in the bearing during a working cycle.

In contrary to steady loaded bearings where the journal centre has a certain location for a given load, in the case of dynamically loaded bearings the journal centre describes during each working cycle a so called journal centre cyclic path.

Calculation methods to determine this path have been developed and published first by HAHN /2/, HOLLAND /3/ and EBERHARD-LANG /4/ and later by other authors /5/, /6/.

Compared with measured shaft displacement pathes a good correlation between various calculation methods and the measurements can be stated, as Fig. 2 shows.

narrowest oil film gap in respect to a vertical line and, on the other, by the eccentricity e of the journal centre or, more favourably, as part of the nominal clearance between journal and bearing, by the eccentricity ratio ε.

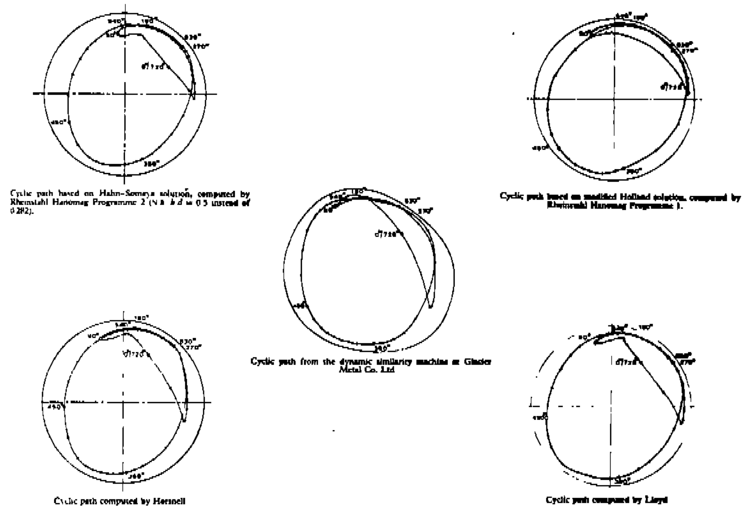

Fig. 2: Comparison between one measured and four computed journal centre cyclic pathes for same conditions /7/

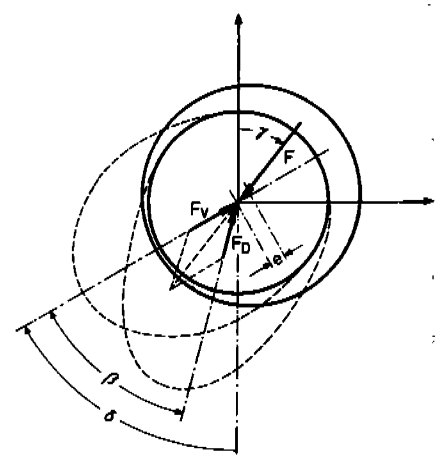

Fig. 3: Momentary position of an instationary loaded journal with oil film pressures and forces

Therefore the calculation methods for computing the journal centre cyclic path are an excellent working tool for a realistic prediction of the bearing behaviour in reciprocating engines, and, in this respect, they are superior to other methods as far as accuracy is concerned.

Because of this advantage these methods are widely used in the automotive industry for the bearing calculation to the internal combustion engines.

For refrigeration compressors, however, up to now they have not been applied generally although recommendations in this direction have been given already some years ago, /8/, /9/, /10/.
The reason for seldom using this calculation methods in compressor industry seems to be the great difficulty to realize the necessary complex computing programs in their smaller computer facilities, compared with those of the automotive industry.

Nevertheless the obvious advantages of the shaft displacement calculation method are also for refrigeration and heat pump compressors so substantial that its application is recommended again in view of the better prediction of the reliability of those machines.

COMPUTING METHOD FOR THE CRANKSHAFT DISPLACMENT PATH

The computing method of the shaft displacement in a dynamically loaded bearing determines the various positions of the journal in the bearing during a working cycle. At any given moment, the position of the journal, as shown in Fig. 3, can be described on the one hand by the angular position δ of the

Both values, ε and δ,thus describe the momentary position of the journal in the bearing under the influence of the external load F, given by the gas and mass forces in the crank assembly, and varying in magnitude and direction during each working cycle.

This load in the case of hydrodynamic lubrication, has to be carried by the hydrodynamic pressures generated in the oil film.

According to Reynolds equation of lubrication in dynamically loaded bearings the hydrodynamic pressures are generated by two effects, the wedge effect and the squeeze effect.

The wedge effect generates oil pressures by transporting oil by friction at the rotating surfaces towards the narrowest film gap. The for this transport effective angular velocity $\bar{\omega}$ consists according to

$$\bar{\omega} = \omega + \omega_S - 2\,\frac{d\delta}{dt} \qquad (1)$$

of the angular velocities of the journal ω, the bearing ω_S and the smallest oil film gap $d\delta/dt$.

The pressure profile as generated by the wedge effect is indicated in the figure by the right hand dotted line. Its integration results in the force F_D generated by the wedge effect which can be expressed according to

$$F_D = \frac{So_D \cdot b \cdot d \cdot \eta \cdot \bar{\omega}}{\Psi^2} \qquad (2)$$

by the dimensionless Sommerfeld-Number So_D for the momentary position of the journal, by the length b, diameter d and dimensionless nominal clearance Ψ of the bearing, further by the oil viscosity η and the effective angular velocity $\bar{\omega}$, as given by equation (1).

So, by combining both equations a differential equation for the angular position δ of the narrowest oil film exists.

The squeeze effect generates oil pressures by the radial displacement of the journal towards the bearing, by which oil has to be squeezed out of the gap.

The pressure profile as generated by this squeeze effect is indicated in the figure by the left hand dotted line and is situated symmetrically to the narrowest oil film gap.

The integration of this pressures profile results in the force F_V generated by the squeeze effect and depends according to

$$F_V = \frac{So_V \cdot b \cdot d \cdot \eta}{\psi^2} \cdot \frac{d\varepsilon}{dt} \qquad (3)$$

in a similiar way on the corresponding Sommerfeld-Number of radial displacement So_V, on the same geometrical bearing parameters b, d, ψ, η, and on the radial velocity of the shaft $d\varepsilon/dt$.

By this differential equation the momentary radial position ε of the journal centre is given.

Thus, by integrating both differential equations (2) and (3), by numerical methods on a computer the position of the journal in the bearing at each moment during a working cycle can be determined gaining so the journal centre cyclic path, can be evaluated in respect to reliability concerning wear, material fatigue, cavitation etc.

EVALUATION OF A JOURNAL CENTRE CYCLIC PATH FOR A RECIPROCATING COMPRESSOR

In order to demonstrate the described method and the evaluation of the journal centre cyclic path concerning bearing reliability, a calculation was made for a six cylinder compressor as designed for refrigeration and heat pump application.

For operating conditions at the application limit of this compressor, namely with an evaporating temperature of - 40 °C and a condensing temperature of + 55 °C for the refrigerant R 502, the result of the calculation was a journal centre cyclic path for the front end bearing No. 1, as shown in Fig. 4

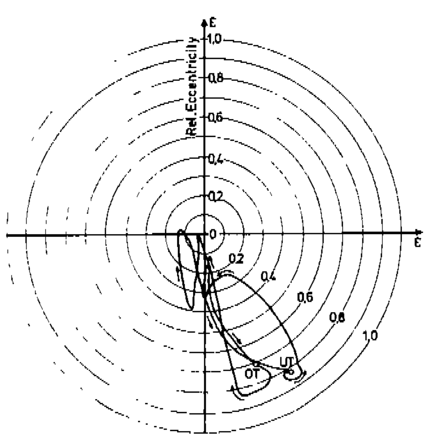

Fig.4: Journal centre cyclic path for bearing 1

Conditions: directly driven, counterclockwise, fully loaded

for directly, counterclockwise driven, fully loaded six cylinder compressor.

An evaluation of this plot concerning the bearing reliability is possible regarding the following points:

Wear

The greatest eccentricity of the journal and its duration during the working cycle is the most important evaluation of the displacement path concerning the bearing reliability especially bearing wear. Bearing wear begins when a contact between journal and bearing occurs, i. e. under ideal surface conditions at an eccentricity ratio ε of 1.0.

In a real bearing with certain surface roughnesses on both sliding surfaces a maximum eccentricity of 1.0 is not obtainable without wear, because already at lower eccentricities the peaks of the surface roughnesses come into contact with each other.

So a minimum oil film thickness is necessary to prevent wear. It should exceed at least the sum of the surface roughnesses although in certain cases, because of the surface deformation under high oil pressures, a value down to 70 to 85 percent of the roughness sum /11/ can be tolerated if the duration of the high eccentricity is only very short, namely below 20 to 30 percentage in time.

With the journal and bearing surface finish of the given compressor a maximum eccentricity ratio of 0.95 according to the sum of surface roughnesses is the safety limit.

Because during the whole journal centre cyclic path, the maximum eccentricity ratio remains below 0.90, not only enough a safety margin against wear can be evaluated from the plot under the assumed running conditions, but also a higher bearing load would still be possible to a certain degree, if an extension of the application range of the compressor to higher condensing temperatures would be necessary.

Material Fatigue And Cavitation

The form of the journal centre cyclic path, besides of its maximum eccentricity, is also important concerning reliability.

Bearing material fatigue depends on high oil pressures, especially if varying very rapidly. Thus, if any parts of the bearing surface get more than once during a working cycle under the influence of great radial velocities of the journal towards the bearing, a fatigue of the bearing material there can occur.

If great radial velocities of the journal are directed away from the surface, the decreasing oil film pressure can reach the vapor pressure causing eventually a destroying of the surface material by cavitation.

Oil Feeding

The journal centre cyclic path shows further the regions with only small oil film pressures during the working cycle, namely there where the smallest pressure built ups by the wedge and the squeeze effect can be stated.

Since the here calculated journal centre cyclic path shows that the shaft is located during the whole working cycle in the lower part of the bearing, no pressure built up by the wedge effect can occur in the upper part. Also no substantial oil film pressures by the radial shaft movements towards the surface can be expected in the upper bearing part, especially not in the right hand upper part because of the absence of radial shaft velocities into this direction.

So a suitable oil feeding could be located there, further probably in the horizontal centre axis.

For other running conditions and design parameters this statement has to be proved.

Design Parameters And Operating Conditions

Concerning compressor design and operating conditions the influences of different parameters such as speed, mass balance, bearing geometry, oil viscosity and different working conditions on the bearing reliability can be studied by the evaluation of the varying journal centre cyclic pathes.

In order to demonstrate the influence of different operating conditions, some calculations of the shaft displacement path were made for the six cylinder compressor with different unloaded conditions given by the capacity control device, different drive directions and for either a directly or belt driven crankshaft.

Compressor Unloaded

If all six cylinders of the compressor are unloaded by the capacity control device a journal centre cyclic path can be calculated as shown in Fig. 5

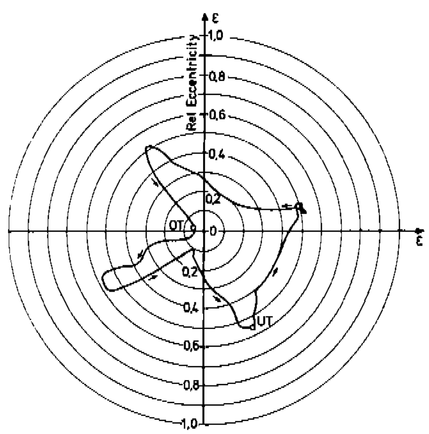

Fig.5: Journal centre cyclic path for bearing 1
Conditions: directly driven, counterclockwise, fully unloaded

Because of the now lower gas forces the maximum eccentricity decreases substantially from below 0.9 under fully loaded conditions to below 0.6 when unloading the compressor.

On the other hand the now dominating mass forces of the crank assembly lift the shaft during a certain time of the working cycle into the upper part of the bearing, so that also a wedge oil film pressure is built up here.

Therefore the location of oil feeding grooves in the upper part of the bearing has to be chosen carefully especially under the aspect of these unloaded running conditions. Besides of a region in the upper right hand part where no radial movements of the shaft towards the surface occur and only a small wedge pressure is built up, a location of the oil feeding near the left horizontal centre axis would also be favourable under these running conditions of the compressor.

This statement has to be proved further for other operating conditions

Direction Of Crankshaft Rotation Changed

If under again fully loaded conditions the compressor is directly driven clockwise instead of counterclockwise a substantial change of the journal centre cyclic path occurs as shown in Fig. 6

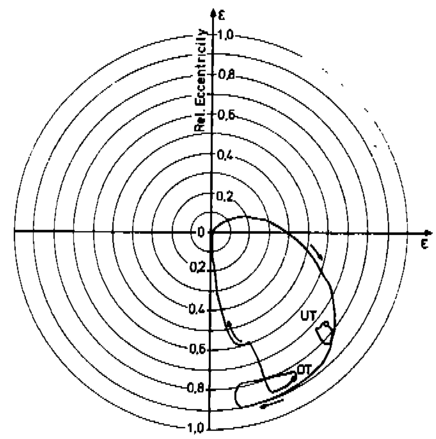

Fig. 6: Journal centre cyclic path for bearing 1
Conditions: directly driven, clockwise, fully loaded

compared with Fig. 4.

The evaluation concerning the maximum eccentricity states that this running direction is more favourable than the opposite direction of rotation.

The maximum eccentricity on the one hand is now higher and reaches a ratio of 0,9, but still in the safety region above the sum of surface roughnesses according to a value of 0.95. On the other hand this higher eccentricity is combined with a longer duration which is more unfavourable because in this region by friction much more heat is generated. This heat may cause a local decreasing of oil viscosity and less load capacity, resulting then in smaller filmthicknesses than calculated.

Furthermore in this case, if once boundary friction has been occurred, it is difficult to reach again hydrodynamic lubrication.

The predominant tangential velocity of the journal in the direction of shaft rotation is unfavourable for producing the wedge effect in the oil film.

Concerning the problem of material fatigue and ca-

vitation this direction of crankshaft rotation shows only two substantial radial displacements compared with four in the opposite direction in Fig. 4.
So from this point of view the clockwise running condition is more favourable whereas concerning wear it is less favourable than the first investigated example.

In the case of operating conditions very near to the application limit a better safety margin for the reliability of the bearing can be achieved by prescribing the direction of rotation, counterclockwise or clockwise, whether wear or material fatigue is the dominant reliability criteria.

A suitable oil feeding again would be possible, also under opposite running direction, in the left horizontal centre axis and the whole upper part of the bearing.

Compressor Belt Driven

Another variation of possible operating conditions is the belt driven compressor. The belt drive introduces high radial loads into the bearings, especially into the bearing no. 2 near the drive.

Regarding at first the influence on the other end bearing no. 1, as already treated in the before mentioned calculations, Fig. 7

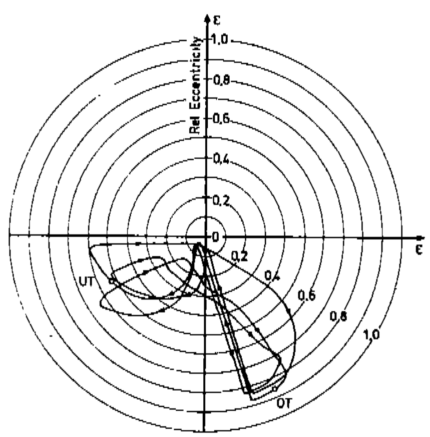

Fig.7: Journal centre cyclic path for bearing 1
Conditions: belt driven, clockwise, fully loaded

shows by comparison with Fig. 6 the change in the journal centre cyclic path.

The maximum eccentricity has been slightly reduced, also its duration, so that better conditions concerning wear can be stated.

Regarding material fatigue and cavitation, however, the situation is much worse. Instead of two substantial radial displacements in the comparable case and four in the first treated example with other direction of rotation, here now six radial movements of the journal in respect to the bearing surface occur in the most loaded region, and the danger of material fatigue and cavitation is now much greater.

Favourable locations of oil feeding grooves are here again in the left horizontal centre axis and the upper right hand part of the bearing since there no radial velocities of the journal centre movement against the bearing surface can be stated.

Especially in the left centre horizontal axis an oil feeding could be favourable there because of the high journal velocity towards the bearing centre, thereby deminishing the danger of cavitation.

By the belt drive the whole central part of the displacement path is extended substantially into the left part of the bearing, indicating a tilting of the crankshaft in both bearings, since the belt load is directed horizontally towards the right. The result is in general a displacement of the shaft in bearing no. 1 to the left and bearing no.2 to the right.

In order to study also the influence of the belt drive on that bearing no. 2 near the compressor drive its journal centre cyclic pathes were computed for the directly and the belt driven fully loaded compressor version.

The eccentricity path for the journal centre of the drive end bearing no. 2, as calculated under the assumption of direct compressor drive is shown in Fig. 8

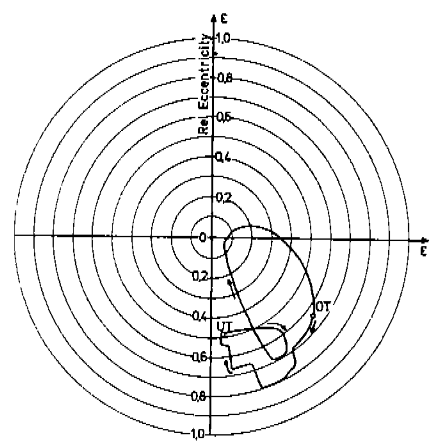

Fig. 8: Journal centre cyclic path for bearing 2
Conditions: directly driven, clockwise, fully loaded

compared with the displacement path of the front end bearing no. 1 under the same conditions, in Fig. 6 a similar form of the curve can be stated with a maximum eccentricity ratio of 0.8 instead of 0.9, indicating a greater safety margin in the drive end bearing, which is a result of a lower bearing load because of double equal bearings on this side.

In the case of the belt driven compressor, Fig. 9, shows that this bearing length is, however, necessary to carry the greater load now resulting from the great horizontal belt force, which was assumed to be extremely high. This force tilts indeed the crankshaft, as already mentioned, so that the journal in the drive end bearing is displaced in general to the right, whereas before for bearing no. 1 a displacement to the left was stated.

The high constant belt force leads to only a very

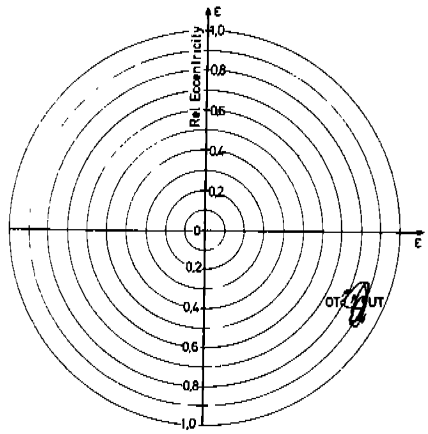

Fig. 9: Journal centre cyclic path for bearing 2
Conditions: belt driven, clockwise, fully loaded

small extension of the journal centre cyclic path with high eccentricity ratios of up to 0.93. The bearing behaviour is similar to that of a steady loaded bearing with a high load.

Thus, under the given running conditions this bearing in the belt driven compressor version must operate with an eccentricity very close to the maximum permissible value as given by the sum of the surface roughnesses.

So it is working approximately at the application limit with only a small safety margin. Still greater belt forces would affect the reliability of the bearing and of the compressor consequently.

CONCLUSION

The described bearing calculation method by which the journal centre cyclic path can be computed and its demonstrated evaluation possibilities offer a good chance to predict in a realistic way the reliability of journal bearings in refrigeration and heat pump compressors.

Therefore by this more realistic modern calculation possibility, as compared with convential methods, also an extension of the application range of those compressors may be achievable.

Because of these advantages this modern bearing calculation method, as already used widely in the automotive industry, is recommended also for the application in reciprocating compressor industry.

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