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A GENERALIZED COMPUTER PROGRAM FOR CALCULATING THE BEARING LOADS IN HELICAL TWIN SCREW COMPRESSORS

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ABSTRACT

This paper presents a suite of programs which together provide a generalised method for calculating bearing loads for a twin screw compressor. The method is general in the sense that it is based on a geometric characteristics program and a performance simulation program which can handle different rotor profiles, port designs, working media and running conditions. All the factors which influence bearing loads were taken into account, including rotor contact and the axial forces due to gas leakage across rotor end faces and the axial gas components on helical screw surfaces and their dispositions on the screw surfaces. Typical results of bearing loads are presented as a function of male rotor angle of rotation for a working model with a range of slide valve settings. The individual contributions of these loading factors are also given.

NOMENCLATURE

F	Force components of rotor contact and bearing reaction, N
h	Pitch length of rotors
r	Radial distance of contact point
sp, ep	Start and end points of contacting part of power transmission section of contact line
T	Torque applied to rotors due to compression gas
x,y,z	Coordinates for rotors
α	Male rotor angle of rotation
β	Helix angle
ε	Pressure angle
ϑ	Slide of tangent at contact point of rotors

Subscript (various combinations of the subscripts are used)

i	index for point number
m,f	male and female rotors, respectively
$n,n3$	normal direction in rotor transverse plane and in 3-dimension, respectively
s,d	suction and discharge end, respectively

INTRODUCTION

One of the big advantages of the helical screw compressor is its flexibility in operation. However, with a change in operating conditions, compression gas induced rotor bearing loads will also change. For instance, with different slide valve settings of a compressor in a refrigeration application, it may happen under certain circumstances that the bearing loads at full capacity are lower than those at partial capacity. Therefore, it is of importance to examine the bearing loads under various operating conditions at the compressor design stage.

An economic and fast way to carry out this task is to use a computer program to calculate bearing loads for various compressor designs and running conditions. In a refrigeration compressor for example the real compression process is complex, involving different refrigerants, oil injection, liquid and gaseous refrigerant injection, internal and external leakages, different port designs as well as different slide valve settings, etc. In order to take all these effects into account, a baring load computation must link the geometrical characteristics with an accurate simulation of the thermodynamic processes which take place.

In this paper, such an integration of the bearing load computation with geometrical calculation and performance simulation is presented, which together provide a generalised method for calculating bearing loads of helical twin screw compressors. Since part of these programs have been reported earlier [1-3], the purpose of this paper is to demonstrate the output of such an integrated program for a particular compressor running with a range of slide valve settings. Also presented is a newly implemented technique for taking the effects of rotor contact into account. The computation also reveals the contributions to the bearing loads of the factors such as rotor contact and axial gas forces on rotor end faces and on screw surfaces.

GEOMETRICAL CALCULATION PROGRAM

The geometrical computation program[1] is designed for the calculation of all the useful geometrical relationships and parameters of a helical screw compressor, the rotor end profile of which may be of any practical shape. The geometrical relationships include those which are required in a thermodynamic model for simulating the working processes of the compressor, such as the cross-sectional area bounded by the two rotors and the housing bores, the cavity volume, the axial and radial discharge port areas, the contact line length, the blow hole areas along the two cusps of the housing bores, the tip to bore sealing line lengths and the leakage path widths on the rotor end faces at suction and discharge, all of which are expressed in terms of the rotational angle of the male rotor. The geometrical parameters include those which are required both in the thermodynamic model and for the design of the compressor, such as the overlap constant, the full thread cross-sectional area, the theoretical and real cavity volumes, the suction-stop angle and discharge port opening angles, and the suction and discharge pocket volumes, etc.

PERFORMANCE SIMULATION PROGRAM

The performance simulation program[2] of a helical screw compressor was developed on the basis of real gas laws, taking all internal and external leakages into account. The influence of factors such as the degree of oil injection, liquid refrigerant injection, vapour charge from the economizer, different refrigerants and partial loading etc., are considered simultaneously and separately in the model. The program reads the data files generated by the geometrical calculation program and outputs the pressure-volume relation, real volumetric capacity, isentropic efficiency, etc. for the running conditions specified by the user.

BEARING LOAD COMPUTATION

Given the pressure-volume data, several methods can be found in the literature for the bearing load computation of a helical screw compressor [4-6]. However these methods either utilize assumptions which simplify the complex rotor profile geometry, which affects the magnitudes of the computed loads, or they use fairly crude methods to calculate axial forces.

In the authors' previous paper[3] the effects of axial gas forces applied to rotor ends and to helical screw surfaces were taken into account. It is worth mentioning that although the resultant axial force applied to helical screw surfaces is zero in the non-contact region of the rotors[4], the moments of the axial forces are not zero, as shown in Figure 1. The presence of these moments of the axial forces have a significant influence on bearing radial loads as can be seen from the computed results in the next section (Table 1).

The authors' first attempt[3] at calculating forces and torques did not take into account inter-rotor contact forces, whereas the method presented here does. It is based on the following simplifications which the authors believe do not result in serious error.

- The rotor profile considered here and shown in Figure 2, is suitable for male rotor drive. Power is transmitted from the male to the female rotor through the transmission section of the profile (*A* to *K* in Figure 2) to overcome the compression gas-induced resistance torque. Since the compression is carried out in a cavity formed by a pair of rotor lobes, the resistance torque arising from cavity pressure is balanced by the torque due to lobe contact forces in the case of the female rotor which performs the function of an idler in a male rotor drive arrangement.

- The magnitude of the inter-rotor contact force will in reality vary along the contact line but in the model it is taken to be constant to make a solution achievable. The magnitude and orientation of this force do however vary with time as the rotor angle and cavity pressure vary.

From Figure 3 the following relations between the components of the contact forces can be obtained:

$$F_{n3f}^2 = F_{nfi}^2 + F_{afi}^2 \tag{1}$$

$$F_{nfi} = F_{tfi} \;/\; \cos(\varepsilon_{2i}) \tag{2}$$

$$F_{afi} = F_{tfi} \cdot \tan(\beta_{2i}) \tag{3}$$

Substituting Eqn. (2) and (3) into Eqn. (1) leads to

$$F_{tfi} = \frac{F_{n3f}}{\sqrt{1/\cos^2(\varepsilon_{2i}) + \tan^2(\beta_{2i})}} \tag{4}$$

According to the assumptions, the compression gas torque of the cavity equals

$$T_f(\alpha) = \int_{sp}^{ep} F_{tfi} \cdot r_{2i} \cong F_{n3f} \cdot \sum_{sp}^{ep} \frac{r_{2i}}{\sqrt{1/\cos^2(\varepsilon_{2i}) + \tan^2(\beta_{2i})}} \tag{5}$$

from which F_{n3f} is determined for a male rotor angle of rotation α. Then there exist

$$F_{nfi} = F_{n3f} \;/\; \sqrt{1 + \cos^2(\varepsilon_{2i}) \cdot \tan^2(\beta_{2i})} \tag{6}$$

and

$$F_{xfi} = -F_{nfi} \cdot \sin(\vartheta_i) \tag{7}$$

$$F_{yfi} = +F_{nfi} \cdot \cos(\vartheta_i) \tag{8}$$

$$F_{afi} = +F_{nfi} \cdot \tan(\beta_{2i}) \cdot \cos(\varepsilon_{2i}) \tag{9}$$

$$M_{xfi} = -F_{afi} \cdot y_{2i} \tag{10}$$

$$M_{yfi} = +F_{afi} \cdot x_{2i} \tag{11}$$

Since $F_{n3f} = F_{n3m}$, similar expressions can be derived for the male rotor. For a rotor angle of rotation α, the length of contacting part of the contact line for power transmission has to be decided first, i.e. the start and end points *sp* and *ep*. Then the effects of these force components on bearing loads can be determined by numerically integrating from *sp* to *ep*. The overall effects of all the cavities are finally obtained by superposing the effects of all individual cavities.

In addition to the factors mentioned above, also taken into account are the axial forces applied to shaft ends and the weights of the rotors. The computer program for bearing load computation is developed to be menu-driven. After reading rotor profile data, rotor structural data and pressure data, the rotor radial and axial forces and torques together with bearing loads can be computed. The results are saved to disk files and can be displayed on screen and printed or plotted to get hard copy.

WORKING MODEL

To demonstrate the capability of this integrated program an numerical example relating to an oil injected refrigeration compressor model is presented. Its main geometrical and running conditions are as follows:

Lobe combination:	4/6	Outer Diameter of rotors:	204 mm
Centre distance:	160 mm	Length vs diameter ratio:	1.65
Pressure ratio:	3	Male rotor rotating speed:	3000 rpm

At full volumetric capacity (slide valve closed) the variation of bearing radial and axial loads with male rotor angle of rotation is shown in Figure 4. As expected, the bearing loads vary with time periodically, with a frequency equivalent to the male rotor rotating speed multiplied by the number of male rotor lobes. It can be seen that the axial load on the male rotor is the largest one among all of the components and is about 3 times of that on female rotor. In contrast, the radial loads on the female rotor are higher than those on the male rotor, and the loads on the discharge end bearings are always larger than those on the suction end bearings.

With the integrated program, the bearing peak loads at different slide valve settings are easily obtained as shown in Figure 5, with volumetric capacity ranging from 18% to 100%. It can be seen that the second highest peak value appears at a volumetric capacity of about 60%. As mentioned before, this value may become larger than that at full capacity under certain circumstances.

For the compressor and running conditions studied, the computation also gives the contributions to the maximum bearing loads of the rotor contacting forces, the axial forces on the surfaces of the screws and the axial forces on the rotor ends, as shown below (a positive sign means for axial forces that the direction is from discharge to suction and for radial forces Figure 3 defines sign conventions). The table relates to maximum values for resultant forces which occur during a compression cycle.

Table 1 Contributions of various factors to maximum bearing loads

Factors	% of F_{rms}	% of F_{rmd}	% of F_{rfs}	% of F_{rfd}	% of F_{am}	% of F_{af}
Axial Forces on Screw Surface	-29.2	17.7	-21.6	14.0	43.1	15.4
Axial forces on Rotor Ends	2.12	-1.82	1.93	-1.31	51.8	100
Rotor Contact	1.37	-1.78	7.66	3.87	5.1	-15.4

It can been seen from this table that the influence of the axial forces applied to screw surfaces is of significance, and must not be ignored in the computation. The effects of the rotor contact forces and the axial forces on rotor ends are relatively small on bearing radial loads, but significant on bearing axial loads.

CONCLUSIONS

A bearing load program is integrated with a geometrical calculation program and a performance simulation program which is capable of computing bearing loads of a helical twin screw compressor with various geometrical parameters and running conditions. Sophisticated techniques were used in the bearing load program to calculate the contributions of all influencing factors such as rotor contact, axial forces on screw surfaces and on rotor ends. The capability of such an integrated program has been demonstrated by applying it to a compressor for a range of slide valve settings.

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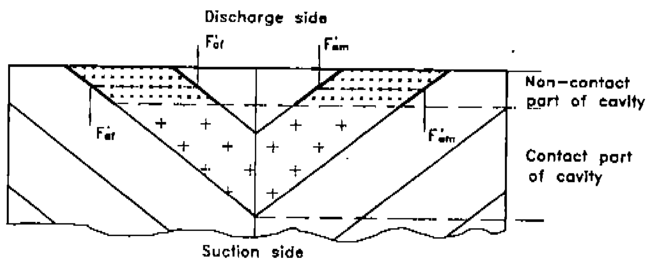

Fig. 1 Moments of axial forces applied to screw surfaces

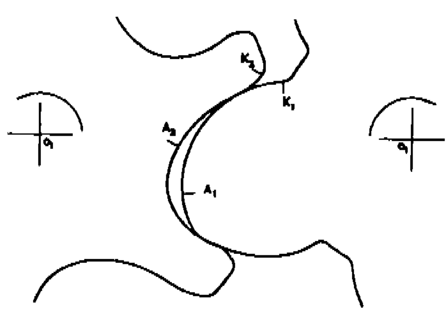

Fig. 2 Power transmission section of the rotor profile concerned

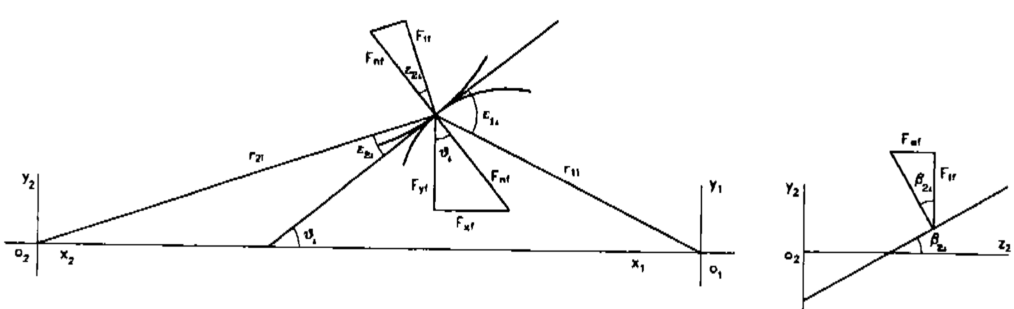

Fig. 3 Relations between components of rotor contact forces

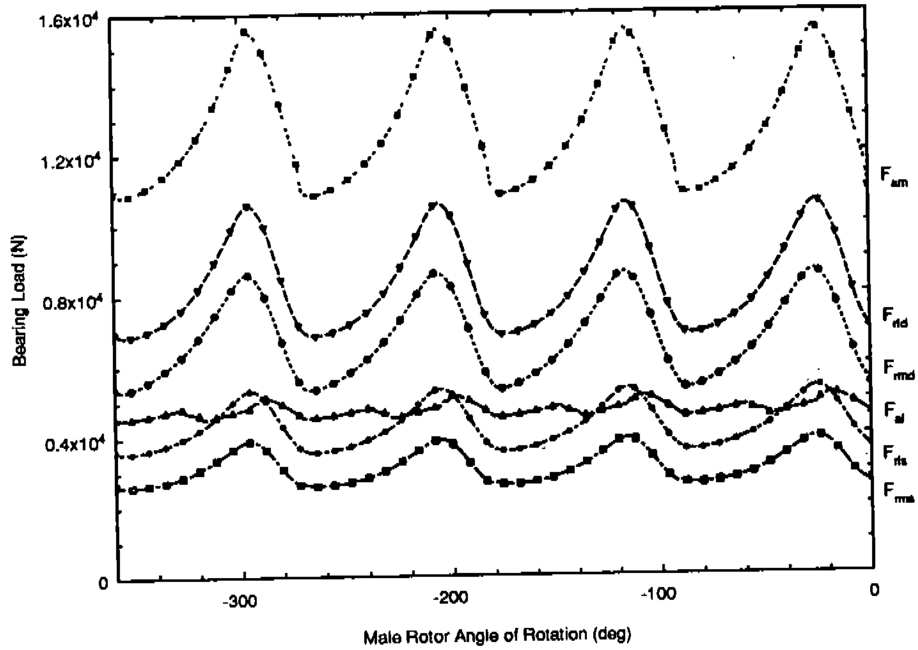

Fig. 4 Variation of bearing loads with male rotor angle of rotation

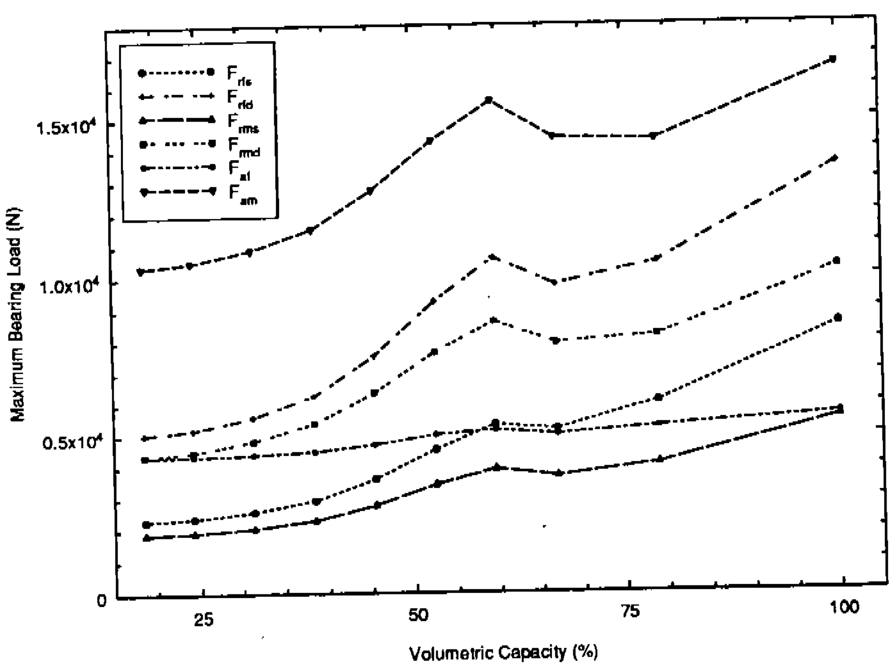

Fig. 5 Variation of maximum baring loads vs volumetric capacity