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VIBRATION ANALYSIS OF REFRIGERANT COMPRESSOR SHAFT

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ABSTRACT

An analysis of transverse vibrations of a compressor, using the modified Myklestad-Prohl method which is one of the transfer matrix methods, is explained. A 10 hp four-cylinder refrigerant compressor was used as an object. The shaft and the frame are modeled as beams which are coupled with the sets of a spring and a damper that are the models of bearing oil film, bearing pedestals or the magnetic pull force of the motor (the frame is supported with suspensions). The forces considered include unbalance forces and pressure forces. The experimental apparatus used to measure the vibrations of the shaft and the frame is described, and computed and measured traces illustrating the vibration amplitudes are presented.

INTRODUCTION

One of the most important problems in refrigerant compressor making is response problem of transverse shaft vibrations. Because, air conditioners with less vibrations and less noises are welcomed these days. Besides there are so many variable speed compressors driven by inverters. This paper presents an analysis of compressor shaft vibrations, taking an example of a 10 hp reciprocal refrigerant type, using multi level rotor dynamic calculation program, shown in Fig. 1.

METHOD

The modified Myklestad-Prohl method is applied to calculate transverse vibrations in this analysis. The method is widely used in turbomachinery that the explanation is omitted. Shown in Fig. 2, the shaft and the frame are modeled as beams. They are coupled in series with the sets of a spring and a damper which are connected at some points. These sets are the models of bearing oil film and those of bearing pedestals or the magnetic pull force of the motor. The frame is supported with suspensions. The vibrations of the shaft and the frame can be obtained by adding exciting forces. The forces consist of unbalances and gas pressures. A-A' surface in Fig. 2 is assumed to be a foundation. The assumption is valid when inner suspensions are very soft.

Fig. 1 Cross Section of Refrigerant Compressor
MODEL DEVELOPMENT

The modelings of the compressors components run as follows.

1) Rotor and Frame
As shown in Fig. 3, the shaft and the frame are modeled as beams which consist of many finite elements. Each element has a concentric mass, a polar mass moment of inertia and a transverse mass moment of inertia at the station, and also it has a length, a linear density mass, a bending stiffness and a shear stiffness at the section.

2) Motor, Piston, Connecting Rod and Balance Weights.
Those above are modeled as distributed masses which are jointed to the shaft or the frame. The motor and the balance weights have polar mass moments of inertia and transverse mass moments of inertia.

3) Bearing Oil Film
The main and the cage bearings, which are the full journal bearings, have the spring and damper effects due to oil films. As the main bearing is too long, it is divided into two parts. The linearized bearing coefficients are restricted to infinitely small perturbations about a steady state position. But, the coefficients are believed useful for the shaft amplitude up to perhaps 40 percent of the bearing clearance. So it is assumed that the behaviour can be adequately represented by a set of eight linearized spring and damping coefficients as shown in Fig. 4.
Consequently, the coefficients are given from charts of coefficients versus Sommerfeld number. One of them is presented in Fig. 5. The Sommerfeld number is obtained from the following equation.

\[ S = \left( \frac{R}{C} \right)^2 \frac{\mu F D L}{W} \]

Dimensional coefficients are obtained by multiplying the nondimensional coefficients of charts by \( \frac{C}{W} \).

(4) Bearing Pedestal
It is modeled as a cantilever beam, as shown in Fig. 6. The stiffness is obtained by calculating the displacement under the unit load.

(5) Suspension
It is assumed that there are springs with rotational and translational restricts at supporting points. Then, values of spring coefficients are decided geometrically. On values of damping coefficients, 5 percent of those companions, (spring coefficients) are given from experiences.

(6) Motor Force
It is modeled as a spring which has a negative linearized coefficient. The spring connects the motor rotor and the motor stator at the center. The value is obtained by experiment of the motor.

(7) Force
Each of unbalance forces by the crank pin and the balance weights are added to each station of the elements of the shaft.

About gas pressure of cylinders, a frequency constituent of the composed force is added to the elements stations of the shaft and the frame. In this case, the
resultant force of the opposed cylinders among 4 cylinders is added at the center of the crank pin and the frame. The moment forces are neglected.

(8) Others
The spring effects of the compressed gasses and the effect of the lubricating friction forces between the pistons and cylinders are neglected.

Fig. 8 Schematic Diagram of Experimental Apparatus

EXPERIMENT

The amplitudes at the tops of the shaft and the frame were measured, as it is difficult to measure the vibration modes. A schematic diagram of the experimental apparatus is shown in Fig. 8. A noncontact eddy-current probe was used to measure the displacement of the shaft, and an accelerometer was used to measure the vibration of the frame. For the comparison with theory, a 2-channel FFT analyzer was used. The compressor was operated with the air gas under the variable line frequency from 45 Hz to 65 Hz. IBM 360 computer was used to calculate the vibrations. The dimensions used for the calculations are shown in Table 1.

RESULT

(1) Shown in Fig. 9, the measured curve of amplitude and the computed one are approximately same. The vibration mode at the resonance frequency (about 270 Hz) in Fig. 9 is shown in Fig. 10. According to Fig. 11, the vibration mode of $K = 10^4 \sim 10^5$ at 270 Hz can be said the first mode of the crank shaft.

(2) In Fig. 12, the calculated values are smaller than the experimental ones. The reason for the difference seems to be caused from probable errors in manufacturing.
CONCLUSION

(1) The modified Myklestad - Prohl method is very useful for the analysis of transverse vibrations of the crank shaft, especially for telling critical speed of compressors.

(2) The most important problem is the estimations of bearing coefficients at fluctuating loads, so it will be needed to research more thoroughly.

(3) By improving the items, mentioned above, to develop this analysis, it will be possible to predict the transverse vibrations accurately. And that will enable to produce better compressors.

Fig. 11 Critical Speed - Bearing Stiffness Map

Fig. 12 IN-Frequency Responce Characteristics of Shaft and Frame

NOTATION

S  Sommerfeld number
D  shaft diameter
R  D/2
L  bearing axial length
C  bearing radial clearance
μ  viscosity
W  static force on bearing
ω  shaft rotational frequency
2ω
K_{ij}  direct and cross-coupled spring coefficient for the oil film
C_{ij}  direct and cross-coupled damping coefficient for the oil film
K_{pii}  direct spring coefficient for the pedestal
C_{pii}  direct damping coefficient for the pedestal
P_d  discharge pressure
P_s  suction pressure
P_c  pressure in a cylinder (i, j=x or y)
BIBLIOGRAPHY


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<th>Table 1 Input Data</th>
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<td>(1) $R_{1r}$, $R_{2r}$, $P_{d-P_s}=2$ Kg/cm²</td>
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**CAGE BEARING**
- $L$: 3.9 cm, 3.0 Kg
- $W$: 5.6 Kg
- $S$: 0.23 x $10^4$ Kg/cm
- $K_{xy}$: $-0.2$
- $K_{yx}$: 5.6
- $K_{yy}$: 0.65
- $S_{xy}$: 0.1
- $S_{yx}$: 1.3
- $K_{xy}$: 8.4
- $K_{yx}$: 17.3
- $S_{xy}$: 0.7
- $S_{yx}$: 17.3
- $K_{sxy}$, $K_{syy}$: $3.5 \times 10^5$ Kg/cm
- $C_{sxy}$, $C_{syy}$: 0.05 Kg/cm

**MAIN BEARING**
- $L$: 8.3 cm
- $W$: 5.5 Kg
- $S$: 0.25 $x 10^4$ Kg/cm
- $K_{xy}$: $-15.5$
- $K_{yx}$: 14.0
- $K_{yy}$: 0.15
- $S_{xy}$: 38.8
- $S_{yx}$: 4.2
- $S_{xy}$: 2.8
- $S_{yx}$: 17.3
- $K_{sxy}$, $K_{syy}$: $4.8 \times 10^5$ Kg/cm
- $C_{sxy}$, $C_{syy}$: 0.05 Kg/cm

**COMMON**
- $D$: 300 cm
- $C$: 22 $\mu$m
- $\mu$: $6.1 \times 10^{-6}$ Nm²/cm

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