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PREDICTION OF NOISE FROM A SCROLL COMPRESSOR

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ABSTRACT

In an air-conditioner, the demand of low noise has been increasing recently. The reduction of the main noise source, i.e. a compressor is required in order to satisfy it. The authors have been developing the prediction system of the noise radiation from the compressor to take noise into account in the design stage. In this paper, the prediction procedure for a scroll compressor and the application to a middle cooling capacity scroll compressor are shown. The first application of the procedure has not given the satisfying results. Therefore, we continue the improvement of the accuracy of the procedure.

INTRODUCTION

Recently, low noise for a compressor is as important as its mechanical efficiency. But it is difficult to accomplish low noise with other demands such as low cost, high speed, light weight, small size and high power. Therefore, it is necessary to take noise into account from the beginning of the design stage.

We have been developing the prediction procedure for a scroll compressor. It consists of the following three stages:

1. Calculation of the excitation forces by the dynamic behavior analysis of the rotating elements
2. Determination of the frequency response of the structures
3. Prediction of the radiated noise

There are many papers which describe the kinematic model of a scroll compressor but they do not refer to the noise radiation.

This paper describes the features of the procedure from the calculation of excitation forces with the dynamic behavior model of a scroll compressor to the prediction of the radiated noise. The application of the procedure to a middle cooling capacity scroll compressor made by Mitsubishi Heavy Industries is also discussed.

CALCULATION OF THE EXCITATION FORCES

1. The outline of the dynamic behavior model

Fig. 1 shows the structure of a scroll compressor. The dynamic behavior model of a scroll compressor incorporating an oldham coupling in the scroll revolution mechanism is discussed. All elements except for a crank shaft are modeled as two-dimensional components in the cross-section perpendicular to the crank shaft axis and the crank shaft is modeled in three dimension.

In this chapter, the equations of the dynamic model described by the motion of an individual element and the forces generated by the operation were derived. (a) Constitutional elements of the model

Elements of the dynamic behavior model are categorized into stationary
elements and rotating elements. The rotating elements are an orbiting scroll, a swing link, a crank shaft and an oldham coupling. They interlocked with each other under the kinetic and geometric restriction, providing the interrelations illustrated in Fig. 2.

The dynamic behavior analysis of them is carried out by solving the set of simultaneous equations of the model and the excitation forces acting on the stationary elements are obtained. The stationary elements are a housing, a fixed scroll and a motor case.

(b) Elastic displacement of the model

It is supposed that the elastic displacement of the model is caused at the point of contact between the rotating elements, and between the rotating elements and the stationary elements and that there is no gap on the above points of contact and no-elastic displacement at the points of contact between the other elements. Therefore, the large displacement caused by the distortion and the bending of the crank shaft and so on are not dealt with. The points of elastic contact to be considered is as follows.

1. the points of contact between the fixed scroll and the orbiting scroll
2. the point of contact between the oldham coupling and the fixed scroll
3. the point of contact between the oldham coupling and the motor case
4. the point of contact between the drive bush and the main bearing
5. the point of contact between the drive pin and the swing link
6. the point of contact between the crank shaft and the main bearing

In order to obtain the relationship between the force and the displacement, the following equivalent stiffness at the point of contact is applied.

\[ \frac{1}{k_{eq}} = \frac{1}{k_A} + \frac{1}{k_B} \quad \ldots \ldots (1) \]

where

- \( k_{eq} \): equivalent stiffness at the point of contact
- \( k_A \): stiffness of the element A at the point of contact
- \( k_B \): stiffness of the element B at the point of contact

(c) Damping of the model

Only colamb damping at the points in (b) is taken into account as follows.

\[ F_t = \mu F \frac{v + \Delta v}{|v|} \quad \ldots \ldots (2) \]

where

- \( v \): relative velocity
- \( \Delta v \): relative velocity difference
- \( \mu \): coefficient of friction
- \( F \): force acting on a plane vertically
- \( F' \): friction force

(d) Forces generated by the operation

The forces generated by the operation are the force of the compressed gas pressure, the centrifugal force and the gravitational force of the rotating elements and the motor torque. To obtain the cylinder pressure, the pressure of two pockets formed at the same time are assumed to be equal. The determination of the cylinder pressure is shown below.

1. the polytropic change from the suction to \( \beta \) point, at which a pocket joins with the neighboring pocket.
2. Over \( \beta \) point, using the measurement results

2. Dynamic behavior equations

The dynamic behavior equations for the rotating elements by applying the principle of d'Alambert to the terms of inertia are discussed. Fig. 3 shows the relationship of forces acting on each element. The simultaneous equations can be obtained as follows.

(a) Oldham coupling (Fig. 4)

Forces:

\[ -m_o \left( \begin{array}{c} \dot{x}_{oo} \\ \dot{\theta}_{oo} \end{array} \right) + F_{om} + F'_{om} + F_{oF} + F'_{oF} = -F_{tf} - F_{om} - F'_{om} \quad \ldots \ldots (3) \]

Moments with respect to the gravitational center of the oldham coupling:
where \( \mathbf{F}_{\text{or}} \) : reaction force from the stationary element
\( \mathbf{F}_{\text{em}} \) : reaction force from the orbiting scroll
\( \mathbf{F}_{\text{orf}} \) : friction force from the stationary element
\( \mathbf{F}_{\text{emf}} \) : friction force from the orbiting scroll

(b) Orbiting scroll (Fig.5)

\[
\begin{align*}
&- \mathbf{F}_{\text{cs}} = - (\mathbf{F}_{\text{em}} + \mathbf{F}'_{\text{em}}) + \mathbf{F}_p + \mathbf{F}_D = - P_c + \mathbf{F}_{\text{emf}} + \mathbf{F}'_{\text{emf}} + \mathbf{F}_{\text{orf}} + \mathbf{F}_D,
\end{align*}
\]

Moments with respect to the gravitational center of the orbiting scroll:
\[
-I \cdot \theta_{\text{cs}} - (\mathbf{r}_{\text{em}} \wedge \mathbf{r}_c) \wedge \mathbf{F}_{\text{em}} + (\mathbf{r}'_{\text{em}} \wedge \mathbf{r}_c) \wedge \mathbf{F}'_{\text{em}} + \mathbf{F}_p + \mathbf{F}_D = - (\frac{1}{2} \rho + \mathbf{r}_c) \wedge \mathbf{P}.\]

\( \mathbf{F}_p \): force from the fixed scroll
\( \mathbf{F}_D \): force from the drive bush
\( \mathbf{P}_c \): resultant force of compressed gas pressure
\( \mathbf{F}_{\text{orf}} \): friction force from the fixed scroll
\( \mathbf{F}_{\text{emf}} \): friction force from the drive bush

(c) Swing link (Fig.6)

\[
- \mathbf{F}_{\text{GL}} = - \mathbf{F}_D - \mathbf{F}_p = - \mathbf{F}_{\text{cw}} - \mathbf{F}_{\text{cb}}.
\]

Moments with respect to the gravitational center of the swing link:
\[
-I \cdot \theta_{\text{GL}} - (\mathbf{r}_{\text{cw}} - \mathbf{r}_c) \wedge \mathbf{F}_D + (\mathbf{r}_p - \mathbf{r}_{\text{cc}}) \wedge \mathbf{F}_p = 0.
\]

\( \mathbf{F}_p \): force from the drive pin
\( \mathbf{F}_{\text{cw}} \): centrifugal force acting on the swing link

(c) Crank shaft (Fig.7)

\[
- \mathbf{F}_{\text{cc}} = - \mathbf{F}_p + \mathbf{F}_{\text{bs1}} + \mathbf{F}_{\text{bs2}} = - (\mathbf{F}_1 + \mathbf{F}_3 + \mathbf{F}_4).
\]

Moments with respect to the gravitational center of the crank shaft:
\[
-I \cdot \theta_{\text{cc}} - (\mathbf{r}_{\text{bs1}} \wedge \mathbf{F}_p) = - (\mathbf{r}_{\text{bs1}} \wedge \mathbf{F}_{\text{bs1}} + \mathbf{r}_{\text{bs2}} \wedge \mathbf{F}_{\text{bs2}} - \mathbf{T}).
\]

\[
- (\mathbf{l}_{\text{mc}} + \mathbf{z}_{\text{c}}) - (\mathbf{l}_{\text{mc}} - \mathbf{z}_{\text{c}}) \wedge \mathbf{F}_p = (\mathbf{l}_{\text{mc}} + \mathbf{z}_{\text{c}}) \wedge \mathbf{F}_p - (\mathbf{l}_{\text{mc}} - \mathbf{z}_{\text{c}}) \wedge \mathbf{F}_p = - (\mathbf{F}_1 + \mathbf{F}_3 + \mathbf{F}_4).
\]

\( \mathbf{F}_{\text{bs1}} \): force from the upper bearing
\( \mathbf{F}_{\text{bs2}} \): force from the lower bearing
\( \mathbf{F}_1 \): centrifugal force acting on the crank pin
\( \mathbf{F}_3 \): centrifugal force acting on the upper balance weight
\( \mathbf{F}_4 \): centrifugal force acting on the lower balance weight
\( \mathbf{F}_{\text{bs1f}} \): friction force from the upper bearing
\( \mathbf{F}_{\text{bs2f}} \): friction force from the lower bearing
\( \wedge \): vector product
\( \mathbf{F}_p (\theta) \): rotational matrix

By substituting the displacement in Eq.(1) for the forces at the point of contact and the displacement in Eq.(2) for the friction forces at the point of contact.
contact, the variables of the forces in Eqs.(3)-(11) are changed into the displacement. Then, the 1st order normalized differential equations were derived from the set of the equations and were solved by Runge-Kutta-Gill method.

3. Excitation forces

The forces transmitted to the stationary elements are as follows.

1. reaction moment from the motor torque
2. reaction force and moment from the fixed scroll
3. reaction force and moment from the oldham link
4. reaction force and moment from the bearing

Finally, the excitation forces are the resultant force and the moment of them at the gravitational center of a scroll compressor as shown below.

\[ \begin{bmatrix} F_x \\ F_y \end{bmatrix} = -F_{of} - F'_{of} - F_f - F_{b1} - F_{b2} - P_c \]

Moment with respect to the tip of the fixed scroll:

\[ \begin{bmatrix} M_x \\ M_y \end{bmatrix} = -z_1 \phi (\pi/2) (F_{of} - F'_{of}) z_f \phi (\pi/2) F_f \]
\[ -z_{b1} \phi (\pi/2) F_{b1} - 2 z_{b2} \phi (\pi/2) F_{b2} - z_f \phi (\pi/2) P_c \]

\[ M_z = - (\ell_f + r_{of} - \delta) \wedge F_{of} - (\ell' + r_{of} - \delta) \wedge F'_{of} \]
\[ - (r_{pc} - \delta) \wedge P_c - (r_f - \delta) \wedge F_f - T_f \quad \ldots \ldots \ldots (12) \]

They are transformed into the forces in frequency domain by Laplace transform.

4. Calculation results and discussion

The excitation forces were obtained from Eqs.(12)-(14). Fig.8 shows the excitation forces for the scroll compressor. It is difficult to obtain the forces experimentally. For the verification of the excitation forces, the timing of impact on the orbiting scroll was compared between the calculation and the experiment in Fig.(9). The maximum impact is generated at \( \theta^* = 40^\circ \) in both. Accordingly, it is confirmed that this dynamic behavior model represents the behavior of a scroll compressor basically.

DETERMINATION OF THE FREQUENCY RESPONSE OF THE STRUCTURES

For calculating the vibration of the housing which is the noise radiated surface, it is necessary to determine the frequency response (spatial average mobility of the housing). Therefore, the finite element model of the structures is set up and the modal parameters are obtained by real-eigenvalue analysis of it.

Finally, the frequency response can be determined from the modal parameters from the excitation points obtained in the previous chapter.

1. Finite element model of the structures

Finite element model of the structures consists of the stationary elements in the previous chapter. Fig.10 shows the finite element model. In this model, the housing is set up by shell elements, and the motor case and the fixed scroll is set up by solid elements. Additionally, the model is axisymetric model and the fixed scroll is modeled as a disk with the equivalent mass properties.

2. Modal parameters of the structures

The modal parameters are obtained by the real-eigenvalue analysis in NASTRAN from the finite element model. 30 modal parameters between 500 Hz and 4 KHz are
obtained for the scroll compressor.

3. Spatial average mobility of the housing

Spatial average mobility of the housing \( <Y_1> \) for the excitation point \( <j> \) is obtained from the modal parameters as follows.

\[
<Y_1>^2 = \frac{1}{S} \sum_{s=1}^{N_m} \left| \frac{V_e}{F_s} \right|^2 \Delta S_e
\]

\[
= \frac{1}{S} \sum_{s=1}^{N_m} \left| \sum_{s=1}^{N_e} m_s \left( \omega_n^2 - \omega^2 \pm 2j \omega n \right) \phi_{s e} \phi_{s e}^* \right|^2 \Delta S_e
\]

where
- \( \zeta_s \): n-th modal damping
- \( N_m \): numbers of total modes
- \( \phi_{s e} \): n-th normal mode shape of the e-th element
- \( N_e \): numbers of elements of the housing
- \( \Delta S_e \): area of the e-th element
- \( S \): total area of the housing

4. Calculation results and discussion

The spatial average mobility for the scroll compressor when the structure is excited vertically on the fixed scroll is shown in Fig. 11(a) and that when the structure is excited radially on the motor case is shown in Fig. 11(b). They show the comparison between the calculation and the experiment. The frequency response curve of the spatial average mobility can be predicted within an accuracy of 10dB error. Even from the rough finite element model mentioned above, good agreement between the calculation and the experiment has been obtained.

PREDICTION OF THE RADIATED NOISE

The following two methods can be applied to predict the noise radiation from the housing. The spatial average velocity of the housing is calculated from the excitation forces and the spatial average mobility of the housing calculated from the mobilities of the housing. The acoustic power radiated from the housing can be obtained by multiplying the spatial average velocity and the radiation coefficient of the housing.

In the alternative method, the velocities of the housing elements in the finite element model are calculated from the excitation forces and the modal parameters. The acoustic pressure radiated from the housing can be obtained by employing the Boundary Element Method.

In this chapter, the first method is described.

1. Spatial average velocity

The spatial average velocity of the housing \( <V(\omega)> \) can be calculated from the excitation forces \( F(\omega) \) and the spatial average mobility \( <Y(\omega)> \) as follows.

\[
<V^2(\omega)> = <Y^2(\omega)> |F(\omega)|^2
\]

and the radiated noise (acoustic power) can be calculated as follows.

\[
L_w(\omega) = 10 \log \left( \frac{\rho c S \sigma(\omega) <Y^2(\omega)>}{10^{-12}} \right)
\]

where
- \( \rho \): density of the air
- \( c \): velocity of sound
- \( S \): area of the housing
- \( \sigma \): radiation coefficient
2. Calculation results and discussion

Fig. 12 shows the frequency response curve of the acoustic power level radiated from the scroll compressor. The calculation and the experiment are compared. Large error are observed in all frequency range. It is supposed that it is caused by the error of the excitation forces estimation. For good agreement of the radiated noise from the housing, we are improving the dynamic behavior model of a scroll compressor as described below.

1) correction of the stiffness at the points of contact
2) consideration of viscosity of the lubricating oil
3) consideration of the modal response of the rotating elements

CONCLUSIONS

We are developing the prediction system of the radiated noise from a scroll compressor, which consists of following three stages:

1) Calculation of the forces,
2) Determination of the frequency response of the structures,
3) Prediction of the radiated noise.

This paper has presented the procedure and the application to a middle cooling capacity scroll compressor. We have obtained the following conclusions.

1) We have made the model to express the dynamic behavior of the rotating elements of a scroll compressor. From the calculation results of the excitation forces obtained by solving the model, the model was shown to represent the behavior of a scroll compressor basically.
2) The accuracy of the frequency response of the structures which are the stationary elements was fairly good.
3) The radiated noise from the housing was predicted within the accuracy of 20dB error.

The first application have not given the satisfying results. Therefore, the improvement of the procedure are being carried on.

REFERENCES

Fig. 1: Structure of a scroll compressor

Fig. 2: Elements of a scroll compressor

Fig. 3: Forces acting on each element
Fig. 4 Oldham coupling model

Fig. 5 Orbit scroll model

Fig. 6 Swing link model

Fig. 7 Crank shaft model
Fig. 8 Frequency response curve of excitation forces in the scroll compressor

measuring points on the fixed scroll

Fig. 9 Timing of impact on the orbiting scroll
Fig. 10 Finite element model for the scroll compressor

Fig. 11 Frequency response curve of spatial average mobility compared between the calculation and the experiment in the scroll compressor

(a) Vertical excitation on the fixed scroll

(b) Radial excitation on the motor case

Fig. 12 Frequency response curve of acoustic power level compared between the calculation and the experiment in the scroll compressor