Estimating Stress of Piping System Excited by Scroll Compressor*

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Abstract
An urgent need exists to predict the stress of a piping system connected to scroll compressors. Predicting the stress of a piping system is a useful way to help reduce the cost of developing products consisting of a piping system and scroll compressors. To meet this need, we investigated two main factors used to predict piping stress: the excitation forces of the scroll compressors and the vibration response of the piping system. We calculated the excitation forces and validated the results experimentally and calculated the vibration response by using the excitation forces and a finite-element model of the piping system. The calculated stress values agreed well with experimentally measured ones.

Key words: Scroll Compressor, Air Conditioners, Piping Stress, Excitation Force

1. Introduction
There is an urgent need to predict the stress of a piping system connected to scroll compressors. For example, predicting the stress of a piping system is a useful way to help reduce the cost of developing an air conditioner, which consists of a piping system and scroll compressors.

A number of researchers have studied the vibration of piping systems caused by compressors. Sawanobori et al. investigated theoretically the dynamic behavior of a piping system caused by unbalanced motion and the transient response of the scroll compressors(1)(2). Okutsu proposed a simple model of a piping system and a compressor for use in calculating such behavior(3).

These studies provide useful background for estimating the stress of piping systems excited by scroll compressors. However, although the compression process in a scroll compressor generates gas loads(4), the effect of the load on the dynamic behavior of the piping system is still unknown. In other words, there is little information on how the vibration response of a piping system excited by the scroll compressor can be predicted using the gas load.

We report here our study of the stress of a piping system excited by a scroll compressor. This work comprises two main components. 1) We obtained the excitation forces created by a scroll compressor on the basis of calculation and experimental validation. 2) We calculated the vibration response of the piping system by using the obtained excitation forces and a finite-element model of the piping system. The calculated stress values agreed well with experimentally measured ones.
2. Compression Principle of Scroll Compressor

The main elements of a scroll compressor are two identical involute spiral scrolls, a fixed scroll and an orbiting scroll, as shown in Fig. 1. These scrolls are assembled at a relative angle of 180°. This angle is maintained by means of an anti-rotation coupling located on the back of the orbiting scroll. The orbiting scroll is driven by a simple short-throw crank mechanism and an electric motor. The suction port of the compressor is at the periphery of the scrolls. As the orbiting scroll rotates clockwise, refrigerant gas is drawn in at the periphery, trapped in a pair of pockets (e.g., dotted area in Fig. 1), and compressed by volume reduction as it is moved toward the center of the scrolls. The compressed gas is exhausted through the discharge port at the center of the fixed scroll.

There is a variable gas load during the compression process, resulting in a reaction torque that acts on the crank shaft and motor. This torque may be an excitation force in the piping system.

3. Excitation Force of Scroll Compressor

3.1 Experimental device

Here we focus on a scroll compressor for an air conditioner. Its specifications are listed in Table 1. Figure 2 illustrates the experimental device we used for measuring the excitation force of such a compressor. It mainly consisted of a scroll compressor, a block with high rigidity, and four 3-component force sensors to measure the vertical force and circumferential force on each foot. As previously reported, the foundation of the testing system must be highly rigid for the excitation force to be measured correctly. We thus constructed a block with high rigidity, that is, a “high stiffness block.” It was an iron cube 450 x 450 x 60 mm in size. The feet were labeled anti-clockwise from Foot 1 at the front left side. All pipes of the compressor were connected to a refrigerant control system using flexible connection pipes. The connected refrigerant control system was kept in the ARI standard condition at each rotation speed of the compressor.
Table 1 Compressor specifications.

| Compressor type   | Scroll compressor |
|-------------------|-------------------|
| Refrigerant       | R410A             |
| Suction volume    | 36.0 cm³/rev.     |
| Exterior dimensions | φ141 x 428 mm    |
| Motor type        | Brushless DC      |
| Rated cooling capacity | 12.8 kW    |
| Speed Range       | 20–110 Hz         |
| Mass              | 24 kg             |

Figure 3 shows example measured frequency responses of the vertical force when the compressor was continuously operated at 100 Hz. The horizontal axis shows the frequency, and the vertical axis shows the excitation force.
The excitation force of the compressor was composed of several peak frequency components, and the amplitudes of the excitation forces on the four feet were almost equal. The peak frequency components resulted from the rotation speed of the compressor and its high harmonics. The highest peak level of the force occurred at 100 Hz, which is the fundamental component of the rotation frequency. Other measurement results showed that the peak level of the vertical and circumferential force occurred at each rotation frequency. Here, “excitation force of the compressor” refers specifically to the fundamental component of the rotation frequency.

3.2 Vertical excitation force

Figure 4 shows the measured and calculated vertical force of the compressor for each rotation speed. The horizontal axis shows the rotation speed of the compressor, and the vertical axis shows the excitation force. As you can see, an increase in the rotation speed led to a parabolic increase in the vertical force. This means that the vertical force is highly affected by a whirling motion.

It has been considered, however, that gas loads in the compressor simultaneously generate a reaction force in the radial (centrifugal) direction. The effects of this radially directed reaction force should be included in the results of Fig. 4. However, the operating conditions of the compressor described in this paper call for a constant pressure at each rotation speed, which would mean that this radially directed reaction force caused by gas loads is theoretically a constant value. It can be seen from Fig. 4, though, that the characteristics of a radially directed reaction force caused by gas loads do not agree with the experimental results, which suggests that the effects of centrifugal force are great.

According to the rigid model of the compressor as shown Fig. 5, vertical force $F_T$ can be expressed as

$$F_T = \frac{m r \omega^2 H}{2R} = \frac{2\pi^2 m r H}{R} f^2,$$

where $m r$ is the out of balance (equivalent unbalanced mass times radius of mass), $\omega$ is the angular velocity, $H$ is the distance from the center of the mass to the bottom of the compressor, $R$ is the supporting radius of the feet, and $f$ is the rotation frequency of the compressor. The calculated values agree well with the experimentally measured ones, as shown in Fig. 4.
3.3 Torque pulsation

The calculated torque pulsation was determined by using the geometric theory of scroll compressors\(^4\). The reaction torque \(T\) can be expressed as

\[
T = \frac{1}{2} P_s p (p - 2t) h \sum_{i}^{N} \left( 2i - \frac{\theta}{\pi} \right) \left( \rho_i - \rho_{i+1} \right),
\]

where

- \(P_s\): suction pressure
- \(p\): scroll pitch
- \(t\): scroll tooth thickness
- \(h\): scroll tooth height
- \(\theta\): rotation angle of orbiting scroll
- \(\rho_i\): compression ratio \(P_i / P_s\) of \(i\)-th compression pocket

Figure 6 shows the calculated reaction torque at each rotation angle of an orbiting scroll. The horizontal axis shows the rotation angle of the orbiting scroll, and the vertical axis shows the calculated reaction torque. We normalized the torque by the maximum calculated value. As you can see, the reaction torque alternates during rotation.

Figure 7 shows the results of a frequency analysis performed on the calculated reaction torque. They were obtained by creating temporal waveforms using the reaction torque plotted in Fig. 6 and conducting a frequency analysis. These original temporal waveforms were prepared given a compressor rotation speed of 100 Hz. As in the results plotted in Fig. 3, torque pulsation was maximum at this fundamental component of rotation speed.
Figure 8 shows the measured and calculated torque pulsation of the compressor for each rotation speed. The horizontal axis shows the rotation speed of the compressor, and the vertical axis shows the torque pulsation. In the experiment, the measured circumferential forces were transformed into torque pulsation. The experimental torque pulsations, $T_{13}$ and $T_{24}$, can be expressed using the opposing circumferential forces, $F_{C1}$, $F_{C3}$ and $F_{C2}$, $F_{C4}$:

$$T_{13} = \frac{4R(F_{C1} + F_{C3})}{2} = 2R(F_{C1} + F_{C3})$$

$$T_{24} = \frac{4R(F_{C2} + F_{C4})}{2} = 2R(F_{C2} + F_{C4})$$

where $R$ is the supporting radius of the feet, and $F_{Ci}$ is the measured circumferential force on the $i$-th foot of the compressor. The averaged value of $T_{13}$ and $T_{24}$ is shown in Fig. 8 as the experimental value.

The calculated (theoretical) torque pulsation, moreover, was constant, as shown in the figure. This is because the assumed operating conditions of the compressor include a constant pressure at each rotation speed.

The experimental and calculated results both show that if the operational pressure condition is constant, increasing the rotation speed has little effect on the torque pulsation.
4. Vibration Response Analysis of Piping System

4.1 Modeling

We used the excitation forces of the compressor described above to analyze the vibration response of a piping system with a scroll compressor. A finite-element model of the piping system for the air conditioner was created, as shown in Fig. 9, and the piping stresses were determined by using the ANSYS® finite-element method (FEM) program. The copper piping was modeled as a beam element, and the compressor was modeled as a concentrated mass and a rigid body element.

The terminating portions of the pipes were connected to either the compressor or a refrigerant tank. The tank was made of steel and fixed directly to the base to ensure high rigidity. The termination portions of the pipes were therefore completely restrained at the points where the pipes connected with the refrigerant tank.

Table 2 lists the measured and calculated natural frequency values for four local modes. The piping system had many local natural frequency modes due to the elasticity of the copper piping. The measured and calculated values agree well.

Figure 10 shows a measured and calculated vibration mode (local mode No. 4 in Table 2). In the experimental result, the modal displacement is represented as a vector, and, in the FEM result, it is represented as a color map. As you can see, an anti-node appears at the same location for both results. This means that the FEM modeled the piping system sufficiently well.

Table 2   Example of local natural frequency (Unit: Hz).

| Local mode No. | Measured (experiment) | Calculated (FEM) |
|----------------|-----------------------|------------------|
| 1              | 14.9                  | 13.9             |
| 2              | 43.0                  | 41.9             |
| 3              | 59.6                  | 53.8             |
| 4              | 108                   | 109              |
4.2 Vibration response analysis of piping system

The piping stress was calculated using the FEM by inputting the excitation force results from whirling motion and the torque pulsation on the center of mass of the compressor. The calculated piping stress with and without torque pulsation is plotted in Fig. 11. We normalized the stress by the maximum experimentally obtained value. The location used for comparing the calculated stresses was the point where the refrigerant tank connected with the compressor suction pipe, as indicated in Fig. 11. These results show that a sensitive resonance effect appeared only when there was torque pulsation and that this effect peaked near 60 Hz. On the other hand, calculated stress without torque pulsation increases gradual from 55 Hz to 70 Hz. It is assumed that the piping system has several local natural frequencies at these frequency range. The calculated stress with torque pulsation was larger than that without torque pulsation.

The calculated and experimentally measured piping stresses at the same point as above are compared in Fig. 12. To measure the maximum piping stress, the piping system was excited continuously by the compressor at 60 Hz, which was chosen to match the resonance frequency of the piping system. The calculated values with torque pulsation agreed well with the experimentally measured ones. Therefore, we can conclude that calculating the
excitation forces using the whirling motion and torque pulsation enables us to accurately determine the stress of a piping system.

5. Conclusion

We developed a means of estimating the stress of a piping system excited by a scroll compressor. The piping stress was determined by using both the calculated excitation force of the compressor and a finite-element model of a piping system with a scroll compressor. Accurately calculating the stress of the piping system requires not only the excitation force results from a whirling motion of the scroll compressor but also a torque pulsation caused by a variable gas load of the compressor.

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References

(1) Sawanobori, T., Nakamura, M., Nagamatsu, A., Kitazawa, H., Okamoto, J., “A Vibration Analysis of Compressor-Piping Coupled System in Air Conditioning Units,” Proc. of 1995 ASME/JSME PVP Conference, (1995), pp. 23–27.
(2) Sawanobori, T., Nakamura, M., Nagamatsu, A., Okamoto, J., “Transient Response Analysis of Compressor-Piping Coupled System in Air Conditioning Units,” Proc. of 1995 Design Engineering Technical Conference, Vol. 3, (1995), pp. 479–485.
(3) Okutsu, N., “Study on the Vibration and Stress of Copper Tubes in Refrigerators and Air Conditioners,” Trans. Jpn. Soc. Mech. Eng., (in Japanese), Vol. 63, No. 611, (1997), pp. 2201–2205.
(4) Morishita, E., Sugihara, M., “Geometric Theory of Scroll Compressors,” Journal of Turbo Machinery, (in Japanese), Vol. 13, No. 4, (1985), pp. 209–219.
(5) Tojo, K., Ikegawa, M., Shibaiaishi, M., Arai, N., Arai, A., Uchikawa, N., “A Scroll Compressor for Air Conditioners,” Proc. of Int. Compressor Eng. Conf. Purdue, (1984).
(6) Sato, T., Taguchi, M., Iwata, Y., Okamoto, J., Nagahashi, K., Tanaka, K., “Methods for measuring excitation force of an electric motor in operation,” DYMAC 99, (1999), pp. 331–336.
(7) ARI standard 520-90, “Positive Displacement Refrigerant Compressors, Compressor Units and Condensing Units,” Air-Conditioning and Refrigeration Institute, (1990).