New technologies

Start-up Torque Analysis of Vane Rotary Compressor

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Abstract

In recent years, reduction of the engine load due to each of the auxiliary components such as compressor, is required to improve the fuel efficiency at a vehicle level. In the case of fixed displacement vane rotary type compressor for air conditioning, although its advantage of low resistance load is recognized, sudden torque fluctuation at start-up timing could cause undesirable impact including additional engine control for transient stability. This paper describes the detailed study on this transient torque generation during the start up. To understand the mechanism, the authors have developed a numerical model for a vane rotary type compressor along with experimental verifications.

Key Words : Compressor, Clutch, Dynamic characteristic, Simulation / Vane rotary

1. Introduction

The concern about environmental issues such as CO₂ emission has been increasing. To offer a solution, we have been developing fuel consumption improvement technology that applies to our vane rotary type compressor products (CR series). In the previous Technical Review[1], we introduced one technological solution with the compressor energy consumption analysis that contributes to improvement of the compressor inherent characteristics under a steady operation. A further improvement is demanded to reduce the compressor's operation torque changes at its transient state for avoidance of an impact on the engine. To fulfill this demand, it is necessary to develop technologies that can estimate an accurate torque behavior and reduce the sudden torque changes. Therefore, focusing on the compressor start-up torque that significantly changes, we have developed a numerical model to find the torque characteristics that are fundamentally used in compressor development. This paper introduces the model outline and test validation results.

2. Structure of CR compressor

The CR compressor has a fixed-displacement concentric rotary vane structure. Shows the exploded view of CR compressor. Fig.2 shows the sectional view of a compressor body. Fig.3 shows the flow of refrigerant gas and oil. Power from the engine is transmitted to the compression body through a dry single disk electromagnetic clutch. The refrigerant is fed from a suction port installed at the front head to the compressor body, through a suction chamber sandwiched by the front head and front side block. A rotor with five vanes is provided in the compressor body enclosed by an oval cylinder and side blocks. Compression is performed for 10 times per revolution. The compressed refrigerant is discharged from a port after passing through a reed valve, a centrifugal type oil separator, and a discharge chamber surrounded by a case and rear side block. The refrigerant and oil are separated in the oil separator.
3. Numerical model

In this study, we found the torque characteristics by numerical modeling the rotor rotation behavior and electromagnetic clutch connection behavior at the time of A/C cycle start-up. The compression body torque and electromagnetic clutch torque generated by friction (hereinafter clutch friction torque) are applied to the inertial system of the rotor (assumed to be a rigid body), causing the rotational motion. The rotor behavior is expressed by the formula (1).

\[
I_\theta \frac{d^2 \theta}{dt^2} = T_{cl} - T_{comp} \quad \cdots \quad (1)
\]

- \( I_\theta \): Rotor inertia moment
- \( \frac{d^2 \theta}{dt^2} \): Angular acceleration of a rotor
- \( T_{cl} \): Clutch friction torque
- \( T_{comp} \): Compression body torque

The following explains the formula of each torque.

3.1. \( T_{comp} \) Compression body torque

After transient refrigerant pressure fluctuation was calculated, the compression body torque was found on the basis of the rotational force which is applied to the rotor from the fluctuating pressure.

3.1.1. Transient refrigerant pressure change

The transient pressure change was calculated with the numerical model in Fig. 4 that expresses one-dimensional refrigerant flow in the compressor. The flow rate at suction port, discharge port, and reed valve is calculated on the basis of respective refrigerant conduit resistances and pressure differences among those conduits. The suction port is provided with a check valve which feeds the refrigerant in the compressor when pressure in the suction chamber becomes lower than that in the suction pipe. At the compressor start-up, pressure in the suction, compression, and discharge chamber is equal and the check valve is closed. This model assumes the pressure in the discharge and suction pipe is constant since the scope is only at the compressor start-up short period. The compression chamber pressure in the adiabatic process is derived from the formula (2), including calculation of the geometric volume of the compression chamber, refrigerant pressure drop at the reed valve, and changes in refrigerant density due to recompression of internally leaked refrigerant. Since more than one chamber is parallel compressing the refrigerant, pressure in each chamber needs to be calculated.

\[
\frac{dP}{dt} = \kappa \frac{dP}{\rho \frac{dt}{dt}} \quad \cdots \quad (2)
\]

- \( P \): Compression chamber pressure
- \( \kappa \): Ratio of specific heat
- \( \rho \): Compression chamber density

Fig. 4 Numerical model of refrigerant flow

3.1.2. Torque applied to the rotor

As Fig. 5 shows, the rotational torque applied to the rotor is calculated from the sum of the vane pressing force, sliding friction force, lubricant film force, and pressures acting on the rotor slit. To calculate the vane pressing force and sliding friction force, the simultaneous equation balanced between the force and the rotation moment is used. The pressure in the compression chamber, pressure applied to the rotor slit, centrifugal force, inertia force, Coriolis force, and lubricant film force calculated in the previous section are assigned to the equation.
3.2. Clutch friction torque $T_{cl}$

Formula (3) calculates clutch friction torque. When the electromagnetic clutch is activated by voltage application, it raises. It makes a contact between friction surfaces and causes a dynamic friction torque. When the clutch connection completes at a constant clutch pulley rotation speed, the friction surfaces turn to static friction state so that rotation speeds become equal between the pulley and the rotor and that the clutch friction torque equals to the compression body one.

$$T_{cl} = \begin{cases} \mu_{cl} F_{cl} R_{cl} & \text{During connection} \\ T_{comp} & \text{After connection (at constant rotation speed)} \end{cases} \cdot (3)$$

- $\mu_{cl}$: Coefficient of dynamic friction on clutch friction surface
- $F_{cl}$: Clutch suction force
- $R_{cl}$: Effective radius on clutch friction surface

4. Experimental apparatus and measurement method

Fig. 6 shows the experimental apparatus. The compressor electromagnetic clutch is driven by a variable speed motor via a belt. The compressor is connected to a car air conditioner cycle system that consists of a condenser, an expansion valve, and an evaporator. Refrigerant and lubricant used for the cycle are HFC134a and PAG oil.

After a steady running, compressor restarts to run at an arbitrary timing. Pressure transducers and thermocouples are respectively provided at a suction chamber and a discharge chamber to measure pressure and temperature of the refrigerant. Angular rate and angular acceleration of a rotor are found by photographing a quadrant marker on the clutch surface with high-speed camera and then by processing the photographs with an image analysis. As the formula (4) expresses, the connected clutch friction torque $T_{cl}$ can be calculated from the sum of the inertia torque and compression body torque.

$$\ddot{\theta} + \frac{l_{p}}{I_{p}} \frac{d\theta}{dt} + \ddot{\theta}_{cl} = \frac{l_{p}}{I_{p}} \frac{d\theta}{dt} + \ddot{\theta}_{comp} \cdots \cdots (4)$$

5. Results and consideration

We calculated the compressor start-up torque and validated in testing under the steady operation conditions of 5000rpm rotation speed, 0.2MpaG suction pipe pressure, and 1.0MpaG discharge pipe pressure. Fig.7 and 8 show the calculation results and test results respectively. The graph (a) shows how the changes appear in the compression body torque, clutch friction torque, and rotor rotation speed. Meanwhile, the graph (b) shows how each chamber pressure changes. In a horizontal axis in each graph, 0 second indicates the time when the clutch starts, the friction surfaces start to contact, and the rotation starts. The calculation results in Fig. 7 (a) show that rotor speed accelerates as the clutch friction torque increases. The rotation of the rotor fluctuates the compression chamber pressure along the suction, compression, and discharge processes in the compression body as shown in Fig. 7 (b). The arrow $P_{o}$ in the graph indicates that the maximum pressure becomes higher than the discharge chamber pressure due to the refrigerant conduit resistance. The suction chamber pressure and refrigerant density gradually decrease and eventually saturate at $P_{e}$ because the refrigerant is suctioned in the compression body. On the contrary, the discharge chamber pressure gradually rises due to the discharge from the compression body.

When the clutch connection completes at $t_{c}$, the maximum rotation speed becomes stable whereas the suction chamber pressure continues to decrease. As a result, mass flow rate becomes maximum at $t_{c}$. Therefore, the compression chamber pressure becomes maximum in the vicinity of $t_{c}$ and the resulting compression body torque also becomes maximum as a reaction force.
As a test result in Fig. 8 (a) shows, the clutch friction torque increases until the connection completion $t_e$. This is because the coil suction force rises. In correlation with the calculation results, the compression body torque peaks in the vicinity of $t_e$ and the suction chamber pressure $p_s$ is higher than the saturation level $p_s$ in Fig. 8 (b). The suction chamber pressure in the model decreases faster than that in the test because the volumetric efficiency is not considered in the numerical model. Therefore, the model suctions the refrigerant faster than the test does, causing the pressure to decrease.

6. Conclusion
For accurate calculation of the start-up torque in the vane rotary compressor, we have developed the numerical model and validated in testing. It reveals reproducibility of the torque characteristics with the torque analysis. As a result, we find that, at the time of the electromagnetic clutch connection completion, high suction chamber pressure causes over-compression and peak in the compression body torque. With accurate estimate of the torque behavior, we will develop a compressor that has less torque changes on the basis of this principle for further contribution to vehicle fuel consumption.

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Reference