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ABSTRACT

Rolling piston type rotary compressor has been widely used with an inverter control which changes the commercial electric frequency driving a compressor motor into various frequencies in order to give an appropriate cooling capacity to the refrigeration and air-conditioning system. However, a conventional rotary compressor of single cylinder type has a considerable problem that the performance severely decreases as its rotational speed of the motor decreases. The reason of this problem could be made clear through the research for a dual cylinder rotary compressor and the analysis for the compressor performance with its rotational-speed fluctuations.

As a result of it, some improvement methods for the performance at low electric frequency operation could become clear theoretically and experimentally. This paper shows the results mentioned above with theoretical analysis and experimental data including comparison between a single and dual cylinder rotary compressor.

NOMENCLATURE

Ic : Inertia moment of crankshaft system
$\dot{}$: Time differential
$\ddot{\theta}$: Acceleration of crank angle
Tm : Motor output torque
Tv : Reaction torque
Tth : Friction torque of thrust bearing
Mc : Friction moment of piston inside
Wg : Bearing load for gas compression
e : Eccentricity of crank
α : Offset angle of rolling piston center
Pc : Pressure in compression chamber
Ps : Pressure in suction chamber
ω p : Angular velocity of piston
Fv : Friction force at vane tip
h : Oil-film thickness in bearing
ej : Eccentricity of journal from the bearing center
ε : =(ej/Cr)Eccentricity ratio
η : Viscosity of lubrication oil
ω j : Angular velocity of journal
s : Slip of rotor
L1 : Self-inductance of stator
L2 : Self-inductance of rotor
θ : Crank angle
$\dot{\theta}$: Velocity of crank angle
Tl : Required load torque
Tg : Gas compression torque
Tf : Friction torque
Ip : Inertia moment of rolling piston
Mb : Friction moment of piston edges
W : Bearing load
rp : Rolling piston radius
H : Rolling piston height
μ : Friction coefficient
r : Crankshaft radius
μ v : Friction coefficient of vane tip
Ff : Friction force at bearing
Cr : Radius clearance
rj : Journal radius
L : Bearing Length
ϕ : Attitude angle of bearing
ψ : Direction angle of load
ω b : Angular velocity of bearing
P : =(d/dt)Differential operator
R1 : Resistance of stator
R2 : Resistance of rotor
M : Mutual inductance between stator and rotor
$\dot{\theta}$ r: Instantaneous rotational-speed of electrical angle
Ids : d-axis current of stator
Idr : d-axis current of rotor
Vds : d-axis voltage of stator
Vdr : d-axis voltage of rotor
Iqs : q-axis current of stator
Iqr : q-axis current of rotor
Vqs : q-axis voltage of stator
Vqr : q-axis voltage of rotor

INTRODUCTION

It was often noted that the rotational speed in a single-cylinder rotary compressor driven by an induction motor must influence on its performance defined as C.O.P(Coefficient Of Performance) . Particularly it was a big problem that C.O.P fell down remarkably at the rotation below than 2000 R.P.M.

In this paper, some reasons bringing on this problem were discussed with dynamic analysis that the instantaneous values of motor torque, inertia torque and load torque for the compression process were solved as simultaneous equations. With this analysis, a certain way to improve C.O.P of rotary compressor under low speed rotation could be found, and it was shown in some figures that the results of simulation analysis were in approximate agreement with the results of experiment.

Consequently, it was able to say that the performance of rotary compressor driven by a frequency control system called an inverter, including a dual -cylinder rotary compressor, could be accurately estimated with this simulation analysis.

DYNAMIC ANALYSIS FOR ROTATIONAL FLUCTUATIONS

(1) Basic Equation

A basic equation as a function of the crankshaft angle, in regard to the rotational-direction system, can be expressed as follows.

$$Ic \cdot \ddot{\theta} + Tl(\theta) = Tm(\dot{\theta}) \qquad (1)$$

Hence, Ic is the value of inertia moment determined from the compressor specifications which consist of a distance of the eccentricity and the mass around the rotational center of the crankshaft system. Tm means motor output torque. Although Tm was treated as a constant value determined by motor characteristics with a constant slip speed so far, Tm($\dot{\theta}$)is considered as a function of $\dot{\theta}$ this time. Because an inductance term in the motor characteristics equations must be influenced by the electric current fluctuations due to the required load torque fluctuations for compression process, values caused by such an influence must take into the term of Tm($\dot{\theta}$) here.

(2) Equations For The Required Load Torque

Tl(θ) means the required load torque for compression work in a defined small crankshaft angle, d θ, and Tl(θ) is given by the next equation.

$$Tl(\theta) = Tg(\theta) + Tv(\theta) + Tf(\theta) \qquad (2)$$

Gas compression torque Tg can be expressed as following equations and pressure in compression chamber Pc and pressure in suction chamber Ps are given by experimental data here.

$$Tg = \Sigma_i Wgi \cdot e \cdot \sin\{(\alpha i + \theta i)/2\} \qquad (3)$$

$$Wgi = (Pci - Psi) \cdot 2rp \cdot \sin\{(\alpha i + \theta i)/2\} \cdot H \qquad (4)$$

Hence, i = 1, 2 $\quad \theta 1 = \theta \quad \theta 2 = \theta + \pi$

Subscripts 1 and 2 in these equations mean the upper and lower cylinder side respectively in case of dealing with a dual-cylinder rotary compressor. For a single-cylinder one, only subscript 1 can be used in the equations.

Fig.1 and Fig.2 shows the schematic views indicating forces and moments acting on rolling piston and vane. Fv, Fs and Fd in Fig.1 and Fig.2 are constraint forces acting on vane tip and sides. These values can be obtained by solving the equilibrium equation and the motion equation of rolling piston described as an equation (5) considering the effect of angular velocity $\dot{\theta}$ and angular acceleration $\ddot{\theta}$ of crankshaft.

$$Ip \cdot \dot{\omega} p = Mc - rp \cdot \mu v \cdot Fv - Mb \qquad (5)$$

The reaction torque Tv can be obtained as follows.

$$Tv = -\Sigma_i Fvi \cdot e \cdot \sin(\alpha i + \theta i) \qquad (6)$$

Hence, i = 1, 2 for a single and dual cylinder rotary compressor respectively as explained in an equation (3) and (4).

The condition of bearings around the crankshaft can be reasonably assumed as fluid lubrication, and friction forces can be obtained by solving the basic equations (7) and (8) for journal bearing with a finite length under load fluctuations. In the equations the averaged velocity of crankshaft is used for simplicity. The journal bearing model discussed here are shown in Fig.3.

$$\frac{1}{rj^2}\cdot\frac{\delta}{\delta\,\theta\,b}\left(\frac{hj^3}{}\frac{\delta\,P}{\delta\,\theta\,b}\right)+\frac{hj^3\,\delta^2 P}{\delta\,Z^2}$$
$$=6\,\eta\cdot Cr\cdot[-\,\varepsilon\,\{\,\omega\,j+\omega\,b-2(\,\dot{\phi}+\dot{\psi}\,)\sin\theta\,b\}+2\,\dot{\varepsilon}\cdot\cos\theta\,b] \qquad (7)$$

$$F_f=\frac{\eta\cdot\omega\,j\cdot rj^2\cdot L}{Cr}\left[\left(\frac{\omega\,j-\omega\,b}{\omega\,j}\,\frac{2\,\pi}{\sqrt{(1-\varepsilon^2)}}+\frac{\varepsilon}{2}\left(\frac{Cr}{rj}\right)^2\frac{W\cdot\sin\phi}{\eta\cdot rj\cdot\omega\,j\cdot L}\right.\right.$$
$$\left.\left.+\frac{2\,\pi\cdot Cr\cdot\dot{\phi}}{rj\cdot\omega\,j}\left\{\frac{1}{\sqrt{(1-\varepsilon^2)}}\right\}\right)\right] \qquad (8)$$

Total friction torque Tf can be expressed as follows.

$$Tf=\Sigma_i\,Ffi\cdot ri+Tth \qquad (9)$$

Hence, $i = 1, 2, 3$ Subscripts 1,2 and 3 in an equation (9) mean the upper,eccentric and lower bearing respectively.

Consequently,the required load torque for compression work $Tl(\theta)$ should be expanded in Fourier series like an equation (10) in order to express Tl as a function of θ. It is necessary to obtain $Tm(\dot{\theta})$ and $\ddot{\theta}$ with $Tl(\theta)$ simultaneously.

$$Tl(\theta)=Tl_o\cdot\{A_o+\sum_{n=1}^{k}An\cdot\cos(n\,\omega\,t)+\sum_{n=1}^{k}Bn\cdot\sin(n\,\omega\,t)\} \qquad (10)$$

Tl_o : the maximum value of load torque

k : the required number of term for appropriate solution

(3) Equations For Motor Output Torque

An Equivalent circuit indicated in Fig.4 is generally authorized and used for analysis of a conventional induction motor. Electric current and voltage equations for dynamic analysis of induction motor can be defined as a matrix equation (11). Equation (11) is obtained by the rotational coordinates transformation called d-q-0 axis transformation usually that the rotary coordinate-system of stator with revolving magnetic field at synchronous speed of the electric frequency can be changed into the fixed coordinate-system. As a result of this transformation,the instantaneous values regarding the motor torque expressed as an equation (12) can be calculated with the instantaneous values of electric currents given by solving equation (11).

$$\begin{pmatrix}Vds\\Vqs\\Vdr\\Vqr\end{pmatrix}=\begin{pmatrix}R1+PL1 & 0 & MP & 0\\ 0 & R1+PL1 & 0 & MP\\ MP & M\,\dot{\theta}\,r & R2+PL2 & L2\,\dot{\theta}\,r\\ -M\,\dot{\theta}\,r & MP & -L2\,\dot{\theta}\,r & R2+PL2\end{pmatrix}\begin{pmatrix}Ids\\Iqs\\Idr\\Iqr\end{pmatrix} \qquad (11)$$

$$Tm(\dot{\theta})=M(Ids\cdot Idr-Ids\cdot Iqr) \qquad (12)$$

It is required that all equations noted above are solved simultaneously and calculations are continued until solutions are reasonably converged. In these numerical calculation, the friction coefficients are determined by the method that the calculated shaft-power agreed with that of the experimental model measured actually.

MODELS FOR ANALYSIS AND EXPERIMENT

(1) Main Specifications

The fundamental construction of a single and dual cylinder rotary compressor which are subjects for this study are shown in Fig.5 and Fig.6, respectively.

A dual-cylinder rotary compressor discussed here has a partition plate located between the upper and lower cylinder, and two eccentric cams located with a phase difference of 180 degrees between the two. However, mechanism for compression in each cylinder is completely as same as the case of a conventional single-cylinder rotary compressor.

A single-cylinder rotary compressor can install some flywheel which can be adjusted and detached on the lower side of the rotor as shown in Fig.5 It is a quite useful way to investigate a relation between the compressor performance and the amount of inertia moment effecting on the rotational fluctuations. The main specifications of the models for this study are described in table 1.

Table 1 : Main Specifications

Compressor Type	Dual-Cyl.	Single-Cyl.		
Swept Volume(cm^3/rev.)	13.0	12.7		
Inertia Moment(N·m)	$0.434*10^{-2}$	$0.408*10^{-2}$	$0.981*10^{-2}$	$1.749*10^{-2}$
Flywheel	Without	Without	With	With

(2) Experimental Apparatus And Method

The experimental models of a single and dual-cylinder rotary compressor were installed in a secondary refrigerant compressor calorimeter to measure gas flow rate and required input power. Pressure in a compression and suction chamber of the cylinder were measured by small piezo type pressure transducers and strain gage type pressure transducers. Temperatures of refrigerant R-22 in the refrigeration cycle were measured with thermocouples.

Magnetic encoder type rotational-speed sensor was put in the edge of crankshaft and lower bearing, as shown in Fig.5 and Fig.6, in order to check the fluctuational values during one turn of crankshaft.

All theoretical and experimental studies in this paper were performed under a certain operating condition indicated in Table 2.

Table 2 : Operating Condition

Condensing Temperature	52 (° C)
Evaporating Temperature	5 (° C)
Return Gas Temperature	15 (° C)
Liquid Temperature Entering Exp. Valve	47 (° C)
Ambient Temperature	35 (° C)
Power Source	3 ∅, Variable
Freq.and Volt.Converter	Freq. and Volt.

RESULTS OF ANALYSIS AND EXPERIMENT

The compressor performance influenced by the operating electric frequency for induction motor can be shown in Fig.7. There is a tendency that the performance deceases as the electric frequency driving a compressor motor becomes lower. Particularly in case of a single-cylinder type, the tendency is more remarkable than that of a dual-cylinder type. The rotational-speed fluctuations measured in the experiment for a single-cylinder one are in Fig.8. There is also a tendency the fluctuational values become larger as the operating electric frequency becomes lower. A relation between two tendencies mentioned above supposes that the rotational fluctuations could effect on the compressor performance.

Fig.9 shows the comparison of the experimental results with calculated results regarding the rotational fluctuations of a single and dual cylinder compressor driving at 34 Hz electric frequency. Through this comparison, theoretically analytical values agree well with the experimental values in both of a single and dual cylinder rotary compressor, and it can be said that this analysis can be used to simulate the other cases.

Fig.10 shows the instantaneous values of Tm, motor efficiency and the rotational-speed fluctuations obtained by solving a basic equation (1) on the basis of the required-load torque Tl for a conventional single-cylinder compressor driven at 34 Hz electric frequency. The rotational-speed fluctuations are expressed as a unit of electric frequency in Fig.10.

Considering a basic equation (1), it seems that the value of inertia moment Ic must influence on the rotational system, so a certain flywheel put on the rotor shown in Fig.5 in order to raise the amount of inertia moment around the crankshaft system of a single-cylinder rotary compressor. As a result of this trial with a flywheel, the rotational-speed fluctuations could be reduced to a sufficiently low level indicated in Fig.11, and the required input power could be improved in a range of frequency less than 40 Hz as shown in Fig.12. Nevertheless, it could make clear that the input power increased in a range of frequency more than 40 Hz due to windage loss caused by the flywheel rotation in discharged gas. Fig.13 shows a relation between the motor efficiency and the driving electric frequency with the effect of flywheel as an inertia moment value.

In other words, these results can be explained from the aspect of motor characteristics in Fig.14 which indicates the instantaneous torque of motor output with the instantaneous rotational-speed. And it is obviously evident that the value of inertia moment effecting on the rotational stability against the load fluctuations must be important for a single-cylinder rotary compressor under a slow-speed rotation.

On the other hand, Fig.15 and Fig.16 explain that the motor caracteristics for a dual-cylinder one driving at a low electric frequency has a small efficiency declining in comparison with a single-cylinder one, because the rotational fluctuations of a dual-cylinder one, having small load fluctuations structurally, is as small as that of a single-cylinder one having a large value of inertia moment. In addition, the performance of a dual-cylinder one can maintain high efficiency in a range of high speed rotation.

CONCLUSION

(1) It was made clear that a main reason why the rotary compressor performance was remarkably decreased as its rotational-speed decreased was the decline of motor efficiency caused by the rotational fluctuations during a turn of crankshaft system for compression work. The reason could be supported by the results of simulation analysis and experiment in this study.

(2) It could be found that the amount of the inertia moment about the crankshaft system was deeply related to the fluctuational values of rotation and the relationship was quantitatively studied by simulation analysis and some unique experimental methods.

(3) Dual-cylinder rotary compressor has a superior characteristics in its structure with regard to the load-torque fluctuations specially under a slow rotational-speed. In consequence of this investigation to compare a single-cylinder rotary compressor adjusting the amount of inertia moment with a dual-cylinder one, it can be made clear that a dual-cylinder type must be a suitable one to be driven by a wide range of electric frequency control in refrigeration and air-conditioning.

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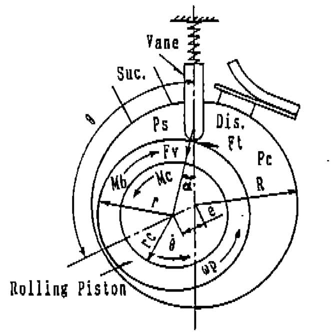

Fig.1 Schematic view, Forces and Moments acting on Rolling Piston and Vane

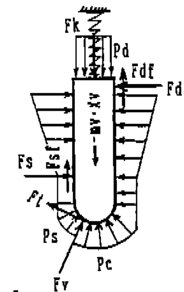

Fig.2 Forces acting on Vane

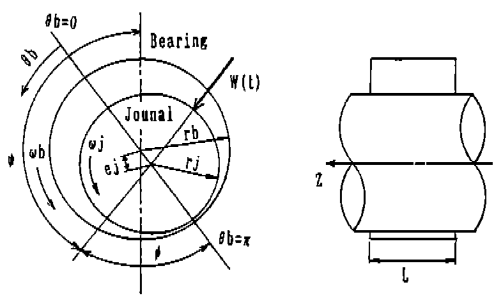

Fig.3 Model of jounal bearing

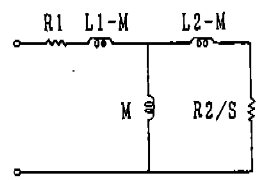

Fig.4 Equivalent circuit for induction motor

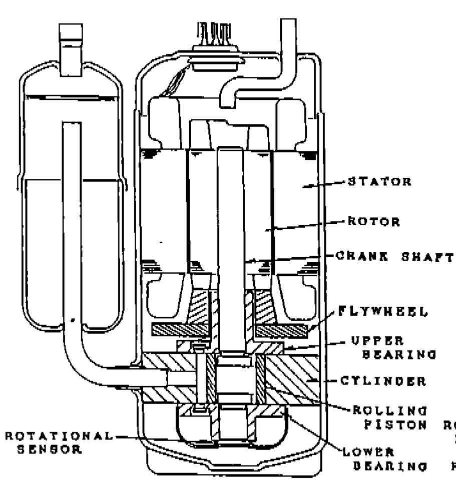

Fig.5 Section view of experimental model for single-cylinder type

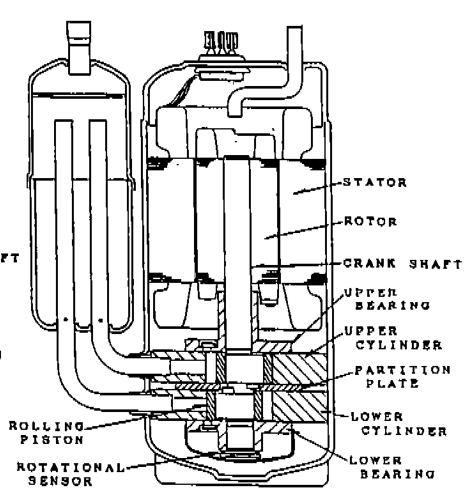

Fig.6 Section view of experimental model for dual-cylinder type

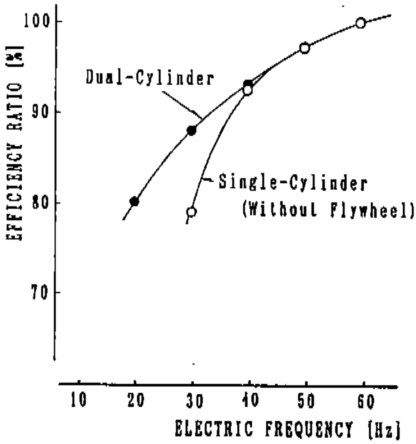

Fig.7 Efficiency comparison

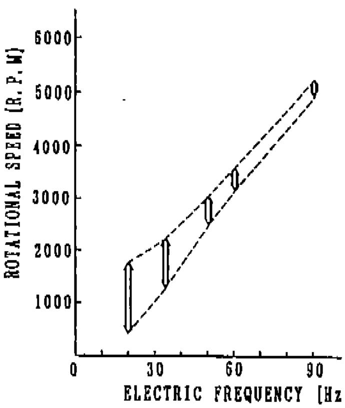

FIG.8 Rotational-speed fluctuations of a conventional single-cylinder compressor

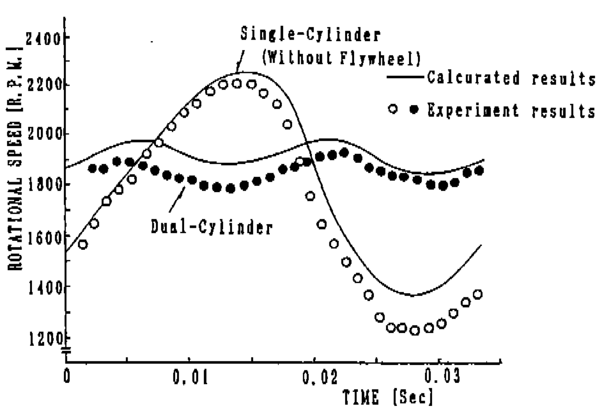

Fig.9 The rotational-speed fluctuations at 34Hz frecuency operation

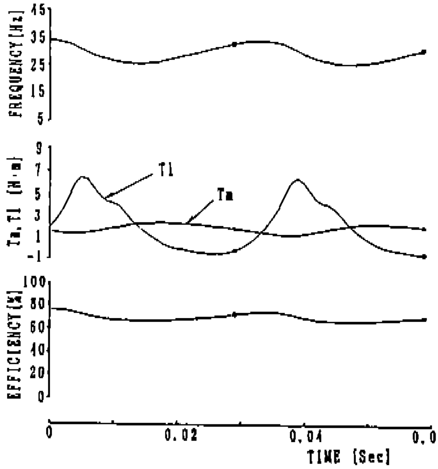

Fig. 10 Analytical results

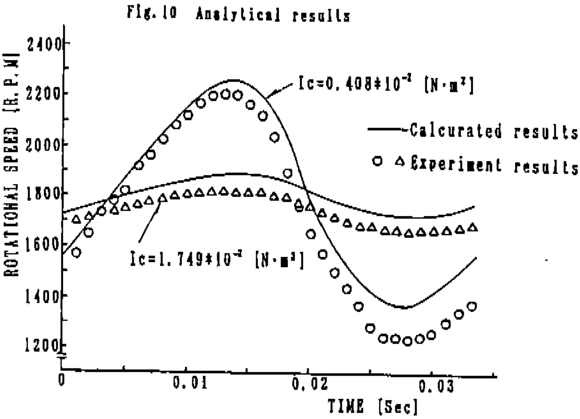

Fig. 11 The effect of flywheel at 34Hz frequency operation

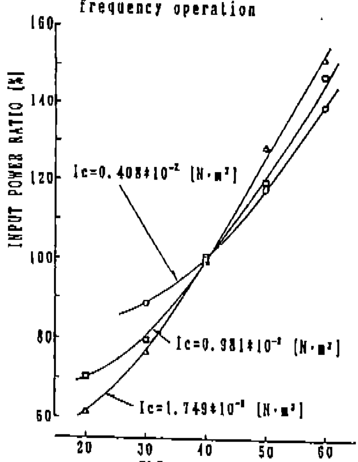

Fig. 12 The effect of flywheel on input power

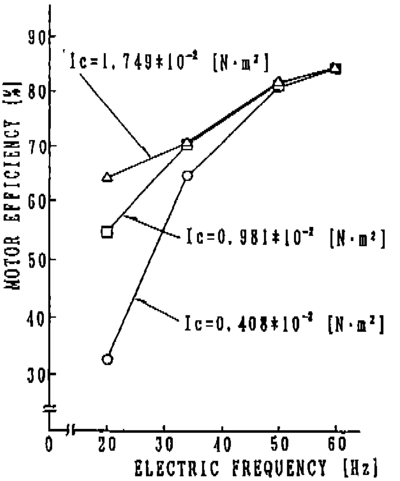

Fig. 13 The effect of inertia moment on motor efficiency

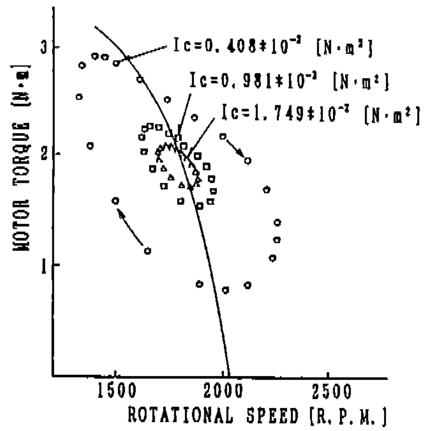

Fig. 14 Instantaneous rotation and torque at 34Hz frequency operation

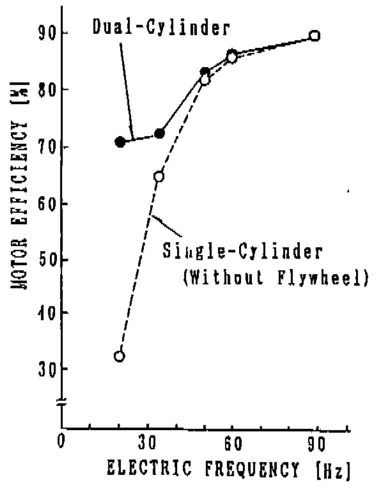

Fig. 15 The effect of load flucutuations on motor efficiency

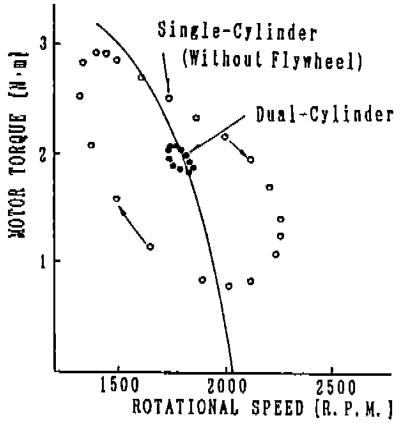

Fig. 16 Instantaneous rotation and torque at 34Hz frequency operation