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TRANSIENT STRESS PRODUCED IN INTERNAL SUSPENSION SPRINGS OF HERMETIC REFRIGERATION COMPRESSOR DURING START AND STOP OPERATIONS

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ABSTRACT

This paper presents the theoretical and experimental investigations on the transient stress of the suspension spring in a reciprocating hermetic refrigeration compressor. The torques acting to the compressor body of a multi-cylinder compressor and the transient vibration of the compressor body are analyzed over a wide range of start and stop operations. The dynamic stresses of the suspension springs are numerically calculated in connection with the transient motion of the compressor body. At the same time, the dynamic stress measurements are carried out on three suspension springs of a five-cylinder compressor under several operating conditions. As a result, it is confirmed that the theoretical results are generally in good agreement with the experimental results. The transient vibration of the compressor body during start and stop operations are revealed. The effects of the operating conditions and the frequency of power source on the dynamic stress of the suspension spring are also clarified.

INTRODUCTION

A reciprocating hermetic refrigeration compressor has a construction such that a compressor body is supported in a hermetic shell by several internal suspension springs with the aim to reduce the running speed torques and forces transmitted to the shell. The start and stop operations of the compressor, however, accompany a very abrupt torque change, thus leading to the transient vibration with the large magnitude in the compressor body. As the compressors are provided with higher performances, this transient vibration tends to increase. This situation motivates this study to clarify the generation mechanism and behavior of the transient stress of the internal suspension spring involved thereby.

Tanaka[1] has reported on the transient stress of the internal suspension springs during stop operation. This study was carried out from the viewpoint of the balance of energy, but was not so far reaching as to clarify the generation mechanism of the transient stress of the suspension springs.

In this study, an approximate analysis is made on the transient motion produced in the compressor body of a multi-cylinder hermetic refrigeration compressor during start and stop operations. The accompanying transient stress of the internal suspension springs is also analyzed. Stress measurements are carried out on a five-cylinder hermetic refrigeration compressor. Comparing the theoretical results with the experimental results, the generation mechanism of the transient stress produced in the internal suspension springs are discussed.

THEORY

Assumptions

Fig. 1 shows an idealized cylinder pressure-crank angle diagram used in this analysis. The following assumptions were made in order to analyze the transient stress generated in the internal suspension springs.

(i) During the compression stroke indicated by AB, the refrigerant inside the cylinder makes the adiabatic change which is expressed by:

$$pv^k = \text{const.} \tag{1}$$

(ii) During the discharge stroke indicated by BC, the cylinder pressure is equal to the discharge pressure and is maintained constant during this stroke.
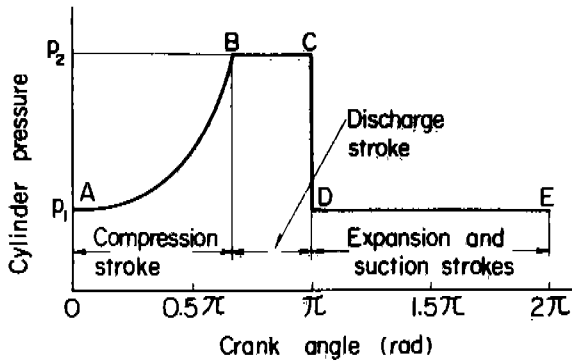
Fig.1 Relationship between cylinder pressure and crank angle.

Fig.2 Schematic diagram of piston-crank mechanism.

(iii) During the expansion and suction strokes indicated by DE, the torque generated by the refrigerant is ignored because it is relatively small[3]. The cylinder pressure, therefore, is regarded as equal to the suction pressure during these strokes.

(iv) The time to get the normal rotating speed of the crank shaft after motor start is very short. The suction and discharge pressures, therefore, are regarded as constant during start operation.

(v) The moment of inertia of the crank pin and crank arm are small in comparison with these of the piston and connecting rod. Accordingly, they are neglected in this analysis.

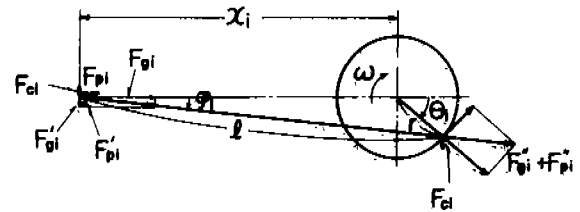Forces Acting to Piston

Force by Gas Pressure. Fig. 2 shows a

schematic diagram of the piston-crank mechanism used in the derivation of equations. During the compression stroke, the cylinder pressure at a crank angle θi is expressed by[2],[3]

$$P_i = P_1 \left(\frac{2 + C}{1 + C + \frac{\rho}{4} + \cos\theta i - \frac{\rho}{4}\cos 2\theta i} \right)^k \quad (2)$$

where $\rho = r/l$, $C = c/r$. Considering the pressure at the piston rear face is equal to the suction pressure, the force Fgi charged on the i-th piston gas pressure during the compression stroke is expressed by

$$Fgi = A_p P_1 \left\{ \left(\frac{2 + C}{1 + C + \frac{\rho}{4} + \cos\theta i - \frac{\rho}{4}\cos 2\theta i} \right)^k - 1 \right\} \quad (3)$$

During the discharge stroke, the discharge and suction pressures act to both sides of the piston. The force Fgi, therefore, is expressed by

$$Fgi = A_p (P_2 - P_1) \quad (4)$$

NOMENCLATURE

A_p = piston area, mm^2

c = top clearance, mm

d = bar diameter of spring, mm

I_R, I_S = moment of inertia of rotating portion and compressor body, N·mm^2

k = ratio of specific heats to gas

K = torsional stiffness, N·mm/rad

l = length of connecting rod, mm

L = distance between the crank shaft and spring center line, mm

n = number of active coils in the spring

P_1, P_2, P = suction pressure, discharge pressure, and cylinder pressure, Pa

r = length of crank arm, mm

R = coil radius, mm

T_R, T_S = torque for crank shaft and compressor body, N·mm

v = specific volume of gas in cylinder at any time, mm^3

ζ = viscous damping coefficient, N·mm·s

θ = revolution angle of crank shaft, rad

ϕ = angular displacement of motion of compressor body, rad

ω = angular speed of crank shaft, rad/s

τ = shear stress of suspension springs, N/mm^2

During the suction stroke, the assumption provides that the same pressure acts to both sides of the piston. The force Fgi, therefore, becomes

$$Fgi = 0 \qquad (5)$$

Inertia Force of Piston. The reciprocating movement of the i-th piston produces such an inertia force Fpi as shown in Fig. 2. This inertia force Fpi is expressed by[4]

$$Fpi = \frac{Wp}{g} r\omega^2 (\cos\theta i - \cos 2\theta i) \qquad (6)$$

where Wp and g are the piston weight and gravitational acceleration, respectively.

Force by Connecting Rod Motion. The motion of a connecting rod is considered as the composition of the line and rotating motions centering around the piston. Accordingly, the modified couple induced by this motion produces the force Fci shown in Fig. 2 at both ends of the i-th connecting rod. This force Fci is expressed by

$$Fci = \frac{Wc}{2g}\{h(1-h) - a^2\} \cdot r \cdot (\frac{\omega}{l})^2 \cdot \sin 2\theta i \qquad (7)$$

where Wc, a and h are the weight of connecting rod, the radius of gyration around the mass center of the connecting rod, and the distance between the mass center of the connecting rod and the crank pin, respectively.

Torque Acting to Compressor Body and Crank Shaft

Torque Acting to Compressor Body. As shown in Fig. 2, the forces acting to the i-th cylinder are the component force Fgi' of the gas pressure force Fgi, the component force Fpi' of the inertia force Fgi of the piston, and the connecting rod force Fci. These forces produce the torque Tsi acting to the compressor body around the compressor center line, which is expressed by

$$Tsi = (Fgi' + Fpi' + Fci) \cdot xi \qquad (8)$$

Expressing Fgi', Fpi' and xi with θi and then summarizing all the torques produced in the m-cylinder, the total torque Ts acting to the m-cylinder compressor body is expressed by

$$Ts = \sum_{i=1}^{m} [r\{(Fgi+Fpi)\rho\sin\theta i$$

$$x \quad (1 + \frac{\rho^2}{2} \sin^2\theta i) + Fci\}$$

$$x \quad (\frac{1}{\rho} - \frac{\rho}{4} - \cos\theta i + \frac{\rho}{4} \cos 2\theta i)] \qquad (9)$$

Torque by Crank Shaft Motion. The forces acting to the crank pin are the component force Fgi" of the gas pressure force Fgi, the component force Fpi" of the inertia force Fpi of the piston, and the connecting rod force Fci. The total torque for the crank shaft of the m-cylinder compressor, therefore, is expressed by

$$T_R = \sum_{i=1}^{m} r\{(Fgi + Fpi)(\sin\theta i - \frac{\rho}{2} \sin 2\theta i)$$

$$+ Fci \cos\theta i\} \qquad (10)$$

Starting Torque of Motor. In the hermetic refrigeration compressor, two-pole motor drives the compressor directly. When the power source is supplied to the motor, the starting torque of motor acts to the compressor body as the reactional torque. The starting torque of the motor is generally expressed by[5]

$$Tm = \frac{3PV^2 (\frac{r_1}{s})}{4\pi fg\{ (r_0 + \frac{r_1}{s})^2 + (X_1 + X_2)^2\}} \qquad (11)$$

where P = number of pole; V = voltage of power source; f = frequency of power source; r_0 and r_1 = primary resistance of coil and secondary resistance in terms of primary resistance, respectively, X_1 and X_2 = primary leakage reactance of coil and secondary reactance in terms of primary leakage reactance, respectively, and s is slip of motor which is given by

$$s = 1 - \frac{PN}{120f} \qquad (12)$$

where N is rotating speed of rotor.

Rotating Speed of Crank Shaft

As can be seen in Eqs. (9) ∿ (11), the torques acting to the compressor body and crank shaft during start and stop operations are expressed as a function of the crank angle. In order to obtain the crank angle at the arbitrary time during start and stop operations, the rotating speed of the crank shaft should be analyzed.

The assumptions are made in this analysis that the angular speed of the crank shaft is ω_0 before the power source of the motor is turned on or off, and that is ω at the

arbitrary time t after motor start or shut off. The other assumptions are also made that during this period, the work made by the starting torque Tm of the motor is almost consumed for the increase of the kinetic energy at the moving section of the compressor, and for the compression action of the refrigerant. That is, there is no viscous dissipation during this period because the time to get the normal rotating speed or standstill of the crank shaft after motor start or shut off is very short. These assumptions lead to the following equation established in regard to the angular speed of the crank shaft at the arbitrary time t.

$$\frac{I_R}{2g}(\omega_o^2 - \omega^2) = \int_0^t (T_R - Tm)dt \qquad (13)$$

Transient Motion of Compressor Body

Fig. 3 shows a cross sectional view of the hermetic refrigeration compressor to be tested. The compressor body is supported to the axial direction by the upper and lower springs at the center portion, and to the circumferential direction by three springs at the outer peripheral portion. During the start and stop operations of the compressor, both the torques Ts and Tm act to the circumferential direction of the compressor body. These torques induce to the compressor body the torsional vibration centering around the crank shaft which passes through its mass center.

The system in which the transient motion will occur consists of the compressor body, the motor, and the circumferential springs. The mathematical model with one degree of freedom system, therefore, can be helpful to analyze the transient motion produced in the compressor body.

When the power source is turned on, the starting torque Tm of the motor acts to the compressor body in a step manner. At the
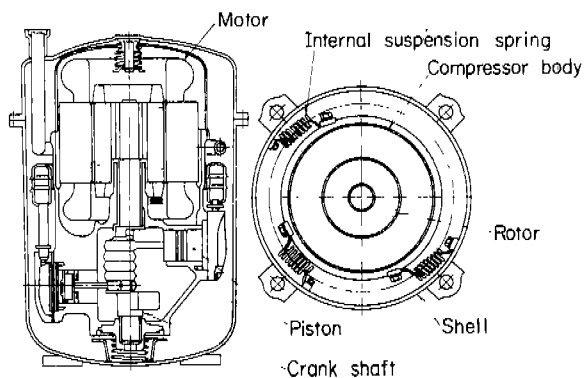
same time, the torque Ts caused by the compression action of the refrigerant acts to the compressor body to the transverse direction to the rotating direction of the crank shaft. These torques will be generated on the compressor body as correspond to the difference between two torques. When the crank shaft reaches the normal revolution, the above two torques come to the equilibrium condition, thus leading to no torque apparently acting to the compressor body.

When the power source is turned off under the normal operating conditions of the crank shaft, the equilibrium condition between the above torques is collapsed abruptly. The torque Ts mainly resulting from the compression action of the refrigerant acts to the compressor body in a step manner. This phenomenon is maintained until the rotating speed of the crank shaft becomes zero. The following equations, therefore, are obtained on the torsional vibration of the compressor body.

(i) $0 \leq t \leq ts$

$$\frac{Is}{g}\frac{d^2\phi}{dt^2} + \zeta\frac{d\phi}{dt} + K\phi = Tm - Ts \qquad (14)$$

(ii) $t \geq ts$

$$\frac{Is}{g}\frac{d^2\phi}{dt^2} + \zeta\frac{d\phi}{dt} + K\phi = 0 \qquad (15)$$

where ts is the time to get the normal rotating speed or standstill of the crank shaft after motor start or shut off, respectively.

The transient vibration caused on the compressor body during start and stop operations can be obtained by solving Eqs. (14) and (15) which contain the non-linear external forces Ts and Tm.

Transient Stress of Internal Suspension Spring

Using the angular displacement ϕ of the compressor body caused by the transient vibration, the shear stress on the outside surface of the coil of the internal suspension spring is given by[6]

$$\tau = (\frac{8R + d}{8R - 4d} - \frac{0.615d}{2R})\frac{dGL}{4\pi nR^2}\phi \qquad (16)$$

where G and L are the modulus of elasticity in torsion and the distance between the crank shaft and the spring center line, respectively.

Fig.3   Cross sectional view of tested refrigeration compressor.

NUMERICAL CALCULATION

As Eqs. (13) and (14) involve the non-linear torques Tm and Ts which depend on the crank angle θi, no analytical solution can be obtained. The following numerical procedure was used to get the solution of Eqs. (13) $\sim$ (15) in this study.

Numerical Procedure for Starting Period

(i) Under the initial conditions at t = 0, the angular speed of the crank shaft is $\omega_0 = 0$, and the angular displacement and angular velocity of the compressor body are $\phi = \phi' = 0$, respectively. The angle of the first crank is regarded as $\theta_1 = 0$.

(ii) Simpson's formula is applied to Eq. (13) and the 4th Lunge-Kutta method is applied to Eq. (14). Both equations are simultaneously arranged to get ω and ϕ.

(iii) The time step in this calculation is selected with t = 0.001 s, and it is further divided into 10 sub-sections for the numerical integration.

(iv) The time ts when the change of the angular speed of the crank shaft is less than 0.1 rad/s is defined as the start-up time of the compressor.

(v) Expressing the initial conditions with the angular displacement and angular speed of the compressor body at the time ts, Eq. (15) is solved by the 4th Lunge-Kutta method.

(vi) Substituting the angular displacement ϕ of the compressor body into Eq. (16) at each time, the dynamic stress produced in the internal suspension spring is obtained.

Numerical Procedure for Stopping Period

(i) Under the initial conditions at t = 0, the assumptions are made that ω_0 = 120 rad/s, $\phi = \phi' = 0$, and $\theta_1 = 0$.

(ii) Simpson's formula is applied to Eq. (13) and the 4th Lunge-Kutta method is applied to Eq. (14) under Tm = 0. Both equations are simultaneously arranged to get ω and ϕ.

(iii) The time step for calculation is t = 0.001 s.

(iv) The time ts when the angular speed of the crank shaft becomes zero is defined as the time to get no rotating speed of the crank shaft.

(v) Expressing the initial condtions with the angular displacement and

speed of the compressor body at the time ts, Eq. (15) is solved by the 4th Lunge-Kutta method.

(vi) Substituting the angular displacement of the compressor body into Eq. (16), the transient stress of the internal suspension spring is obtained.

EXPERIMENTAL PROCEDURE AND RESULTS

Hermetic Refrigeration Compressor used for Experiment

Figs. 4 and 5 show a block diagram of the experimental arrangement and a view of the experimental equipment, respectively. Table 1 gives the specifications for the hermetic refrigeration compressor which was used for this experiment. The compressor used for the experiment is provided with five-cylinders. The compressor body is supported to the vertical direction by the upper and lower springs at the center portion and to the circumferential direction by three springs at the peripheral portion. The hermetic shell of the compressor is divided into two sections and these are
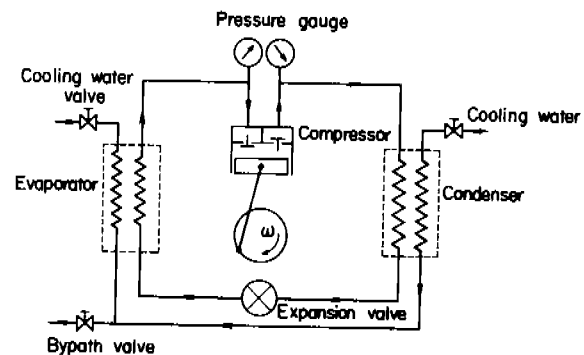

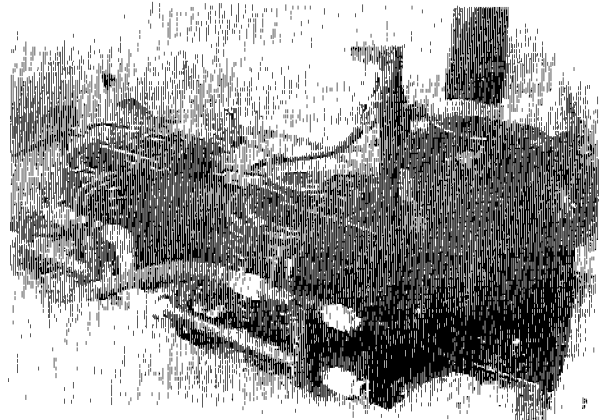Fig.4 Schematic diagram of experimental arrangement.

Fig.5 View of experimental equipment.

Table 1   Specifications for tested
          compressor

Compressor	Cylinder diameter (mm)	42.9
	Stroke (mm)	25.9
	Number of cylinder	5
	Moment of inertia of compressor body (N·cm²)	27800
	Moment of inertia of rotating portion (N·cm²)	931
	Volume of cylinder (cm³/rev)	187.0
Motor	Rating (W)	7500
	Rated voltage (V)	200
	Rated frequency (Hz)	50/60
	Pole number	2
Internal suspension spring	Bar diameter (mm)	3.2
	Coil radius (mm)	7.95
	Number of active coils in spring	5.0
	Distance between crank shaft and spring center line (mm)	120.6

connected by the flange. This construction
makes easy to take out the lead wires from
the strain gauges.

Fig. 6 shows a view of the strain gauges
mounted on the internal suspension springs.
The one-element strain gauges were so
oriented on the outside surface of the
spring coil to detect a normal strain pro-
portional to the shear strain. They were
bonded with an angle of 45 degrees to the
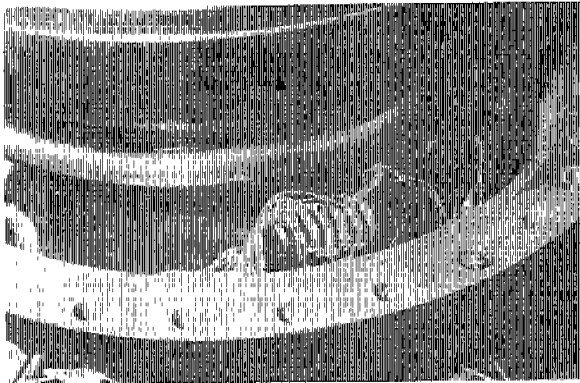spring coil axis.

Fig.6   Arrangement for strain measurement
        of internal suspension spring.

Experimental Procedure

The test conditions are given in Table 2.
By changing the valve settings for the
cooling water and the refrigerant charge in
the system, the test conditions of the com-
pressor were adjusted to any required
operating condition. R 22 was used as the
refrigerant in this test. The power source
voltage and frequency were 200 V and 60 Hz,
respectively. In this experiment, the
variations in the displacement amplitude of
the compressor body were also measured
under the damped free vibration. And the
damping coefficient was estimated from the
logarithmic decrement.

Table 2   Test conditions

	Suction pressure p_1 ($\times 10^5$ Pa)	Discharge pressure p_2 ($\times 10^5$ Pa)
Starting	7.0	19.0
	9.2	9.2
Stopping	5.0	20.0
	6.4	28.6
	2.0	28.6

Experimental Results

Fig. 7 shows a typical example of the
results obtained from the measurement of
the shear stress generated in the internal
suspension spring during start operation.
In this figure, the shear stresses were
derived by substituting the measured normal
strains into the following equation.

$$\tau = 2 \, G \, \varepsilon \tag{17}$$

where G and ε are the modulus of elasticity
in torsion and the measured normal strain,
respectively.

The signs from I to III shown in Fig. 7
represent the shear stress at the center
portion of the spring. The signs from IV
to VI indicate the shear stress at every
other pitch from the end portion. Fig. 8
shows a typical example of the measured
shear stress at two different internal sus-
pension springs during stop operation. The
signs from I' to III' shown in Fig. 8
represent the shear stress at the center
portion of the spring. The signs from IV'
to VI' represent the shear stress at every
other pitch from the end portion. The
lines with the signs from IV' to VI' shown
in Fig. 8 are the data for another spring
and are recorded with the reversed sign in
order to distinguish the data from that of
the former spring.

When the power source is supplied to the
compressor, the stress variations shown in
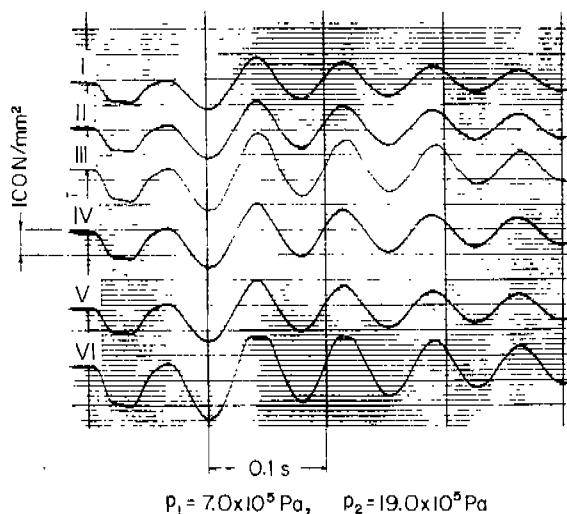Fig. 7 are caused in the internal suspen-

Fig.7 Stress fluctuations of internal
suspension spring during start
operation.

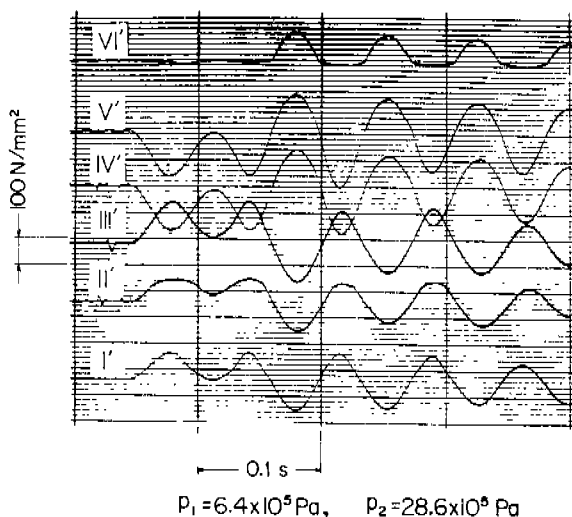$P_1 = 7.0 \times 10^5 \, Pa, \quad P_2 = 19.0 \times 10^5 \, Pa$

Fig.8 Stress fluctuations of internal
suspension spring during stop
operation.

sion springs. As the rotating speed of the
crank shaft reaches the normal revolution,
the stress variations are damped gradually,
following to disappear. If the power
source is turned off under this operating
condition, the stress variations appear
again in the internal suspension spring.
When the rotation of crank shaft comes to
standstill, this stress variation is damped
slowly until it disappears.

As is clearly shown in Figs. 7 and 8, no
phase difference is found among the stress
variations produced in the internal suspen-
sion spring during start and stop opera-
tions, even if the stress measuring loca-
tions and the springs are not the same.

These phenomena indicate that the internal
suspension springs are almost evenly
deformed to these axial directions. The
stress variations at the end portion of the
spring shown with the signs VI and VI',
however, are somewhat different from those
at the central portion. This may be caused
by the contact of the spring coil with the
spring supports at the end portions.

DISCUSSION

Torque Acting to Compressor Body

Starting Torque Variations of Motor. Fig.
9 shows the theoretical result of the
starting torque-time curves when the test
compressor is started under the power
source with a voltage of 200 V and a fre-
quency of 60 Hz. When the power source is
supplied to the compressor under a suction
pressure of 7.0×10^5 Pa, and a discharge
pressure of 19.0×10^5 Pa, the starting
torque is maximum at about 0.10 s. If
there is no pressure difference between the
suction and discharge pressures, the start-
ing torque becomes maximum at about 0.06 s
because of the reduction in the compressor
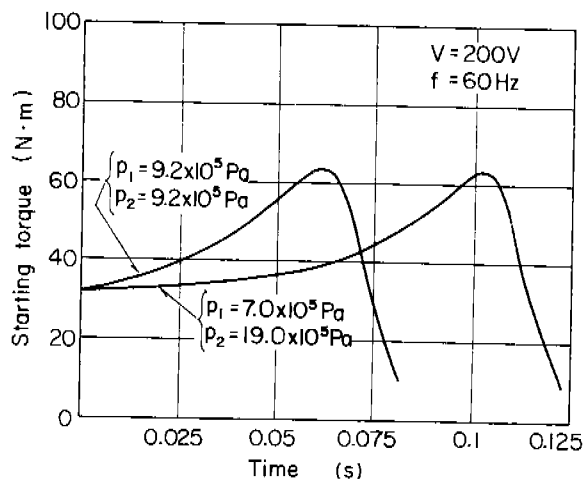loading.

Fig.9 Variations in starting torque
of induction motor.

Meanwhile, if the power source frequency is
changed from 60 Hz to 50 Hz under the same
power source voltage, the starting torque
is increased by about 1.6 times as large as
that in 60 Hz case. This leads to the
reduction in the start-up time, thus accom-
panying the maximum starting torque at 0.03
∿ 0.05 s.

Torques by Pressure of Refrigerant and
Inertia of Crank Mechanism. Fig. 10 shows
the theoretical result of the torque Ts
acting to the compressor body under a normal
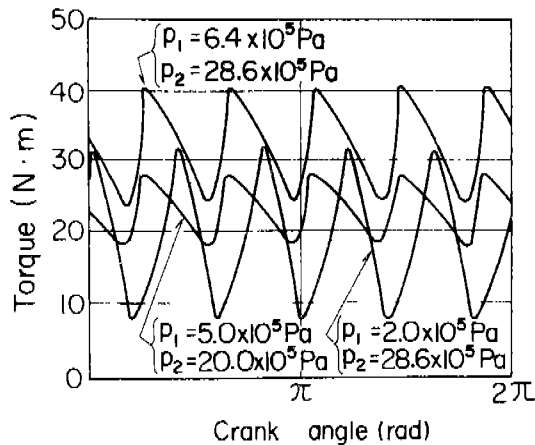rotating speed of 3600 rpm. As can be seen

Fig.10 Torques acting to compressor body.

in Fig. 10, very complicated torques act to the compressor body. The torque acting to the compressor body is ruled with that caused by the compression action of the refrigerant. This means that the torque caused by the inertia of the crank mechanism could be ignored. The average value of this torque variation tends to be large when both suction and discharge pressures are high. Even if the discharge pressure is high, the average torque is comparatively small under the low suction pressure.

Rotating Speed of Crank Shaft

Rotating Speed during Start Operation. Fig. 11 shows the theoretical result of the angular speed of the crank shaft when the compressor is started under the power source with a voltage of 200 V and a frequency of 60 Hz. When motor start is undertaken under a suction pressure of 7.0 x 10^5 Pa, and a discharge pressure of 19.0 x 10^5 Pa, the rotating speed of the crank shaft reaches the normal revolution at about 0.12 s. Under no difference between these pressures, the start-up time of the compressor is shortened to about 0.08 s
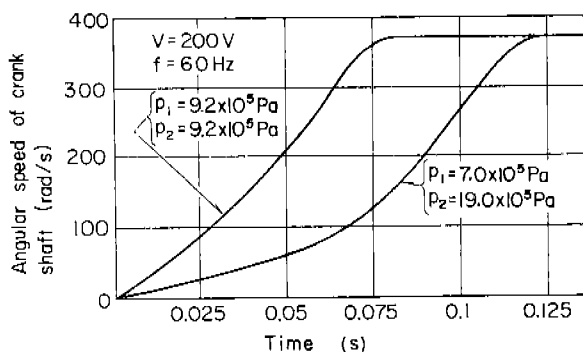

Fig.11 Relationship between angular speed of crank shaft and time during start operation.

because the starting torque is consumed only for the increase in the rotating speed of the crank shaft.

On the other hand, if motor start is undertaken at the same voltage and a power source frequency of 50 Hz, the starting torque of the motor will increase to about 1.6 times as large as that in 60 Hz case. This leads to the reduction of the start-up time, shortened to 0.05 ∿ 0.06 s.

Rotating Speed during Stop Operation. Fig. 12 shows the theoretical result of the angular speed of the crank shaft when the power source is turned off suddenly under a normal rotating speed of 3600 rpm. As can be seen in Fig. 10, very complicated torques act to the compressor body until the rotating speed of the crank shaft reaches standstill. During this period, the angular speed of the crank shaft decreases almost constantly with increasing the time as shown in Fig. 12. Fig. 12 also indicates that the time to get the rotating speed of zero is about 0.11 s under a suction pressure of 6.4 x 10^5 Pa, and a discharge pressure of 28.6 x 10^5 Pa. It is about 0.19 s under a suction pressure of 2.0 x 10^5 Pa and a discharge pressure of 28.6 x 10^5 Pa. The time required for stop operation is almost in proportion to the average torque obtained from the torque variations shown in Fig. 10. If the compressor is under the normal rotating speed at a power source frequency of 50 Hz, the decreasing tendency in the rotating speed of the crank shaft is quantitatively similar to that in 60 Hz case.
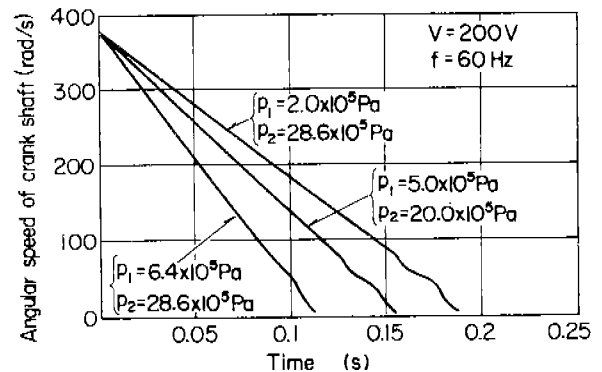

Fig.12 Relationship between angular speed of crank shaft and time during stop operation.

Transient Stress Produced in Internal Suspension Spring

Comparison between Theoretical and Experimental Results. Figs. 13 ∿ 17 show the comparison between the theoretical and experimental results on the shear stress

produced in the internal suspension spring over a wide range of test conditions. In these figures, the variations of the shear stresses are indicated on the base that shear stresses before motor start or shut off are zero. From these results, it is clear that the theoretical results are comparatively in good agreement with the experimental results. This means that the calculation is successful to simulate the generation mechanism of the shear stress produced in the internal suspension spring. The theoretical result also shows that the effect of the damping on the shear stress is comparatively small. This leads to the conclusion that the damping coefficient in Eqs. (14) and (15) can be neglected in the practical spring design.

Transient Stress during Start Operation.
When the compressor is started up under a suction pressure of 7.0×10^5 Pa, a discharge pressure of 19.0×10^5 Pa, a voltage of 200 V, and a frequency of 60 Hz, the shear stress produced in the suspension spring is in the pulsating fluctuation within the period of 0.12 s as shown in Fig. 13. When the motor is started up, both the torques caused by the motor start and the compression action of the refrigerant act to the compressor body in a step manner. This phenomenon induces the pulsating fluctuation shown in Fig. 13 in the stress variations during this period. When the rotating speed of the crank shaft reaches the normal rotating speed, they become in the equilibrium condition, accompanying no torque apparently acting to the compressor body. This phenomenon produces the completely reversed fluctuation of the shear stress because of the free vibration of the compressor body. On the other hand, if there is no pressure difference between the suction and discharge pressures, the time to get the normal rotating speed of the crank shaft is shortened to about 0.08 s. During this period, a large transient
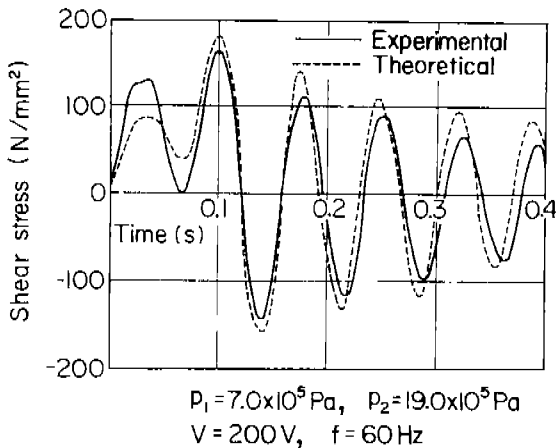
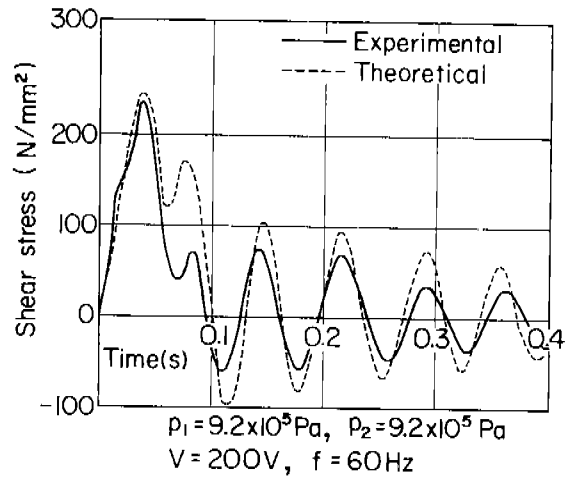shear stress is caused in the internal suspendion spring as shown in Fig. 14.

Fig.14 Shear stress of internal suspension spring during start operation.

When the compressor is started up under the power source with a voltage of 200 V and a frequency of 50 Hz, the starting torque of the motor is about 1.6 times as large as that under a power frequency of 60 Hz. This induces the larger stress than that in the internal suspension spring under a power frequency of 50 Hz.

Transient Stress during Stop Operation.
Comparing Figs. 15 ∿ 17 with Fig. 12, it is clear that the stress fluctuation caused in the internal suspension spring during stop operation correctly corresponds to the time to get the rotating speed of zero. Namely, if the power source is turned off under a normal rotating speed of 3600 rpm, a suction pressure of 5.0×10^5 Pa, and a discharge pressure of 20.0×10^5 Pa, the torque shown in Fig. 10 acts to the compressor body within the period of 0.16 s until the compressor comes to standstill. This torque

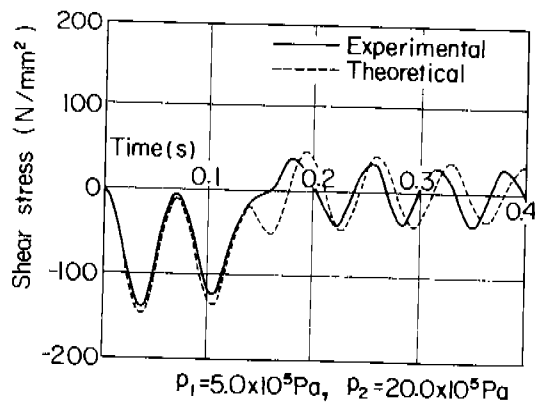Fig.13 Shear stress of internal suspension spring during start operation.

Fig.15 Shear stress of internal suspension spring during stop operation.

induces the transient vibration in the compressor body and the shear stress of the internal suspension spring with the pulsating fluctuation. After the rotating speed of the crank shaft becomes zero, the vibration of the compressor body turns to the free vibration because of no external exciting torque acting to the compressor body. At the same time, the shear stress of the internal suspension spring also turns to the completely reversed fluctuation.

In the case when both the suction and discharge pressures are high, the rotating speed of the crank shaft becomes zero immediately after the second peak appears in the stress fluctuation. Thereafter, the compressor body turns to the free vibration with the large amplitude. This phenomenon is accompanied by the large stress variation, as indicated in Fig. 16. Meanwhile, if the suction pressure is low and the discharge pressure is high, the time to get standstill in the crank shaft rotation increases to about 0.19 s. In this case, three peaks appear in the stress fluctuation during this period.
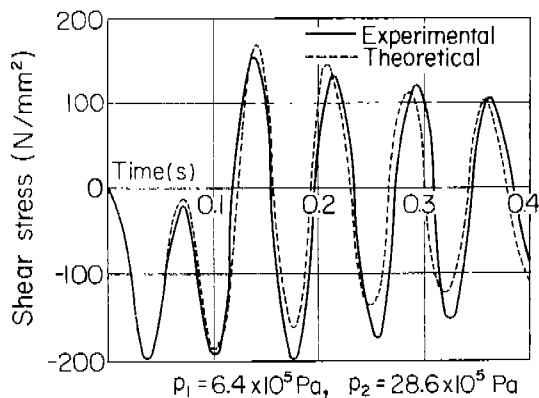

Fig.16 Shear stress of internal suspension spring during stop operation.
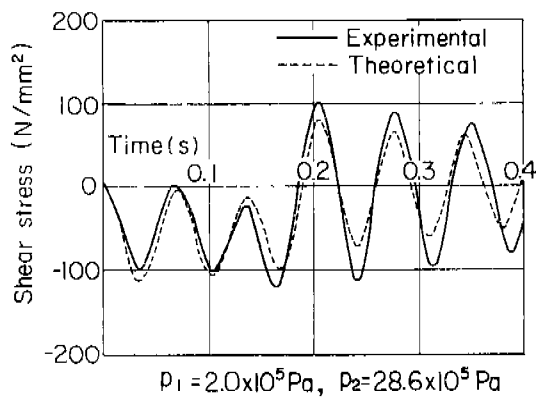

Fig.17 Shear stress of internal suspension spring during stop operation.

Comparing Figs. 15 ∿ 17 with Fig. 10, it is clear that the stress amplitude of the internal suspension spring is effectively ruled with the average level of torque obtained from these fluctuation torques rather than the maximum value of the torque fluctuation. In other word, if the suction and discharge pressures are high, the shear stress of the internal suspension spring is also large. Meanwhile, if the suction pressure is low, it is not necessarily large even though the discharge pressure is high. The maximum stress value of the internal suspension spring amounts to about 1.90 ∿ 1.95 times as large as that obtained under the assumption that the average value of the fluctuating torque acts to the compressor body in a step manner.

CONCLUSIONS

An approximate analysis was conducted on the transient stress produced in the internal suspension spring of the multi-cylinder hermetic refrigerating compressor during start and stop operations. At the same time, experiments were carried out on the hermetic refrigeration compressor with five cylinders, and the transient stresses of the internal suspension springs were measured over a wide range of start and stop conditions. Comparing the theoretical results with the experimental results, the stress generation mechanism was discussed. The results obtained are as follows:

(1) When the compressor is started up, the difference between the starting torque of the motor and the torque by the compresion action of the refrigerant acts to the compressor body in a step manner. When the power source is turned off, only the torque by the latter acts to the compressor body in a step manner. These abrupt torque variations induce the transient vibration to the compressor body, accompanying the transient stress in the internal suspension spring.

(2) The transient stress produced in the internal suspension spring has the close corelation with the time to get the normal rotating speed or standstill of the crank shaft. During these periods, the stresses of the internal suspension spring are in the pulsating fluctuation. After the rotating speed of the crank shaft reaches the normal rotating speed or zero, the transient vibration of compressor body turns to the free vibration, accompanying the stress variations of the internal suspension spring with the completely reversed fluctuation.

(3) The stress produced in the internal

suspension spring is almost ruled with the average value of the fluctuating torque rather than the maximum value of the torque which acts to the compressor body. This transient stress amounts to 1.90 ∿ 1.95 times as large as the static stress obtained under the assumption that the average torque acts to the compressor body in a step manner.

(4) When the compressor is started up under the same power source voltage, the larger stress is produced in the internal suspension spring under a power frequency of 50 Hz than that under a power source frequency of 60 Hz. When the power source is turned off, however, the power source frequency does not influence the stress of the internal suspension spring.

From this study, it is confirmed that the current spring design is acceptable. This means that the design concept described above is in accordance with practical experience for the internal suspension spring. The design of the internal suspension spring has been largely a matter of trial and error up to date. Through this investigation, however, it is considered to be made possible to determine quantitatively the dimensions of the internal suspension spring.

This study has been conducted for a comparatively large hermetic compressor. In order to establish the general concept of the spring design, further investigations are needed to conduct for a small hermetic compressor with multi freedom for the vibration of the compressor body.

REFERENCES

(1) Tanaka, G., "On the Suspension Spring of the Refrigerating Compressor", Refrigeration (Reito, Japan), Vol. 42, No. 472, Jan. 1967, pp.34-45.

(2) Soumerai, H. and T. Kusuda, "Digital Computer Applied to Compressor Design Analysis", ASHRAE Journal, Vol. 1, No. 7, July 1959, pp.43-49.

(3) Matsushima, M. and A. Yokoyama, "Studies on Torque Characteristics of Refrigerating Hermetic Compressor", Hitachi Hyoron, Vol. 45, No. 5, May 1963, pp.115-120.

(4) Watari, A., "Dynamics of Crank Mechanisms", Kyoritsu, Tokyo, 1954, pp.111-123.

(5) Electric Association, "Induction Machine and AC Commutator Machine", Electric Engineering Handbook, 1st ed. 1967, p.602.

(6) Wahl, A.M., "General Design Consideration for Herical Compression or Extensional Springs", Mechanical Springs, 2nd ed. McGraw-hill, New York, 1963, pp.55-74.