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Capacity Control of Rotary Type Compressors for Automotive Air-Conditioners

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CAPACITY CONTROL OF ROTARY TYPE COMPRESSORS
FOR AUTOMOTIVE AIR-CONDITIONERS

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ABSTRACT

This paper presents recent developments relating to the capacity control of rotary type compressors. The basic design theory and its practical application, as outlined in this paper, will provide effective capacity control in rotary type compressors[1].

Because of their light weight, small size and low level of noise, we have seen a recent increase in the use of rotary type compressors in automotive air-conditioners. The major drawback to the use of rotary type compressors has been the fact that refrigerating capacity increases in proportion to an increase in engine speed. When running at a high speed the cooling capacity is increased to such an extent that there is an overcooling of the compartment which causes an overload of the condenser. A solution is needed that eliminates the major drawbacks without sacrificing the light weight, small size and low level of noise of rotary type compressors.

We focused our attention on the speed dependent refrigerant flow in the suction stroke. The ultimate purpose of our research was to develop a compressor which would combine the characteristics of little loss of refrigerating capacity at low speed operation and effective suppression of refrigerating capacity at high speed operation. The result of our research has made it possible to eliminate the factors which have previously hindered the realization of capacity control in compressors, and the final rotary type compressor design turned out to have ideal capacity control characteristics.

INTRODUCTION

As previously stated, in order to achieve capacity control in a compressor for automotive air-conditioners, the following two characteristics are necessary:

(1) Little loss of refrigerating capacity during low speed operation.

(2) Effective suppression of refrigerating capacity during high speed operation.

In order to incorporate the above mentioned characteristics in a rotary type compressor, several different solutions had been proposed. These included open and shut suction ports using change valves, the use of a planetary gear transmission and so forth; however, all of these involved great structural complexity. Although no detailed research, especially from the standpoint of capacity control, has been conducted regarding reciprocating gas compressors with suction valves and rotary type compressors, it is known that because of the difference of their suction characteristics, reciprocating gas compressors produce a suppression of refrigerating capacity during high speed operation.

In our research we concentrated on the flow of suction chamber refrigerant in the suction stroke of the rotary type compressor. The purpose being to design a compressor which would satisfy the two previously mentioned necessary conditions of little loss of refrigerating capacity during low speed operation and effective suppression of refrigerating capacity during high speed operation. In our analysis we used a first order, nonlinear differential equation, which indicates a thermodynamic equilibrium in the suction chamber during suction stroke. An insight into the type of hinderances on capacity control in compressors can be gained by finding the dependence of pressure drop characteristics on the rotation speed by analysing characteristics of different combinations of vanes and effective flow area of suction passage. We further studied the influence of an effective flow area of suction port in suction stroke, which is one of the hinderances indicated above. From our results we were able to construct a model compressor incorporating ideal capacity control characteristics. We introduced a refrigerating capacity control parameter (K_2) showing the relationship between the compressor's construction and the characteristics of its pressure drop. [See Fig.5, and Eq. (13)] When an effective flow area is constantly maintained during suction stroke, the refrigerating capacity control effect will be equal for all compressors, if they have been constructed to exhibit an equal value of K_2.

From this result we can determine the construction of a compressor which satisfies the previously mentioned two necessary conditions in air-conditioners for vehicles in which engine speed varies from 800 to 8,000 rpm. In addition, power consumption is effectively suppressed and it also resulted in a decrease of the operation cycle of clutch during high speed operation. All of this made it easier to design the air-conditioner.

When discussing the coefficient of power including power consumption, it is necessary to not only study the suction stroke but also the compression and discharge stroke[2]. At this time, however, we will limit our analysis to suction stroke and deal with the necessary characteristics of a compressor which effectively satisfies the two previously mentioned characteristics.

MATHEMATICAL MODELING FOR ANALYSIS

This section deals with an analysis of pressure characteristics of the chamber in the suction stroke of a compressor. The energy equation can be transformed into a dimensionless form by the introduction of an approximate function to a volumetric curve of compressor's chamber. From this equation, the capacity control parameter K_2, which is very useful when designing capacity controlled compressors can be seen.

[1] Basic Equation

The pressure characteristics of the compressor's chamber during the suction stroke can be expressed by the following energy equation[4], when Ps is constant.

$$\frac{Cp}{A}GT_A dt - PadVa + dQ = d\left(\frac{Cv}{A}\gamma aVaTa\right) \quad \text{..........} \quad (1)$$

In equation (1) above, the first term of the left member represents the thermal energy brought into the suction chamber through the suction port, while the second term represents the work done by the refrigerant, third term represents the thermal energy delivered from the outside through the wall of the compressor. The right part of the equation, represents the increment of the internal energy of the whole system.

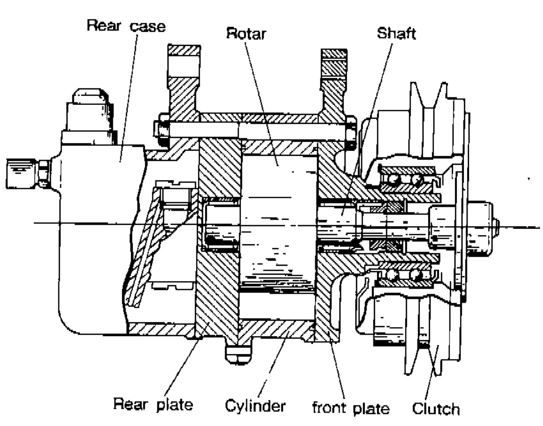

Fig.1 Construction of rotary type compressor

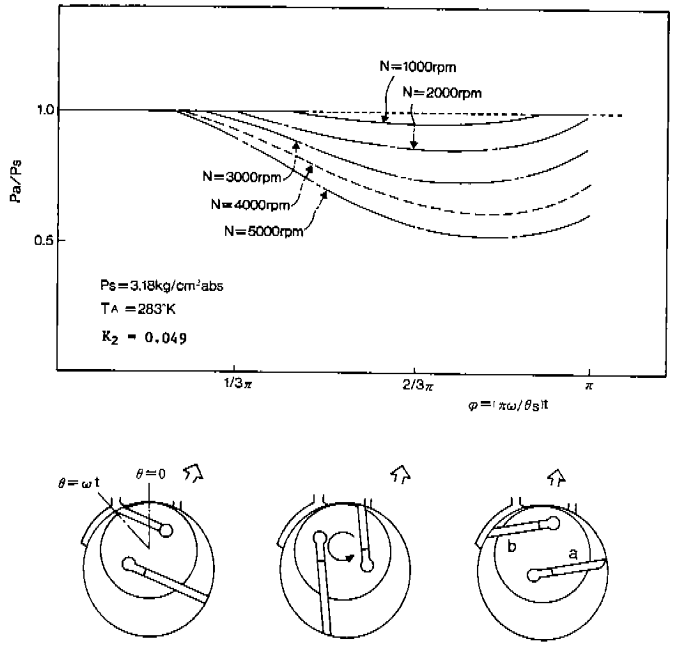

Fig.3 Pressure characteristics in suction chamber

It is assumed that there is no leakage of refrigerant from the high pressure side into the suction chamber during suction stroke. Assuming here that the refrigerant follows the Law of Ideal Gas, and that the suction stroke is adiabatic, the following equation is derived when $\gamma a = Pa/RTa$ and $dQ/dt = 0$, and using the relationship expressed by $\frac{1}{R} = \frac{A}{Cp} + \frac{1}{\kappa R}$

$$G = \frac{1}{RT_A}\frac{dVa}{dt} \cdot Pa + \frac{Va}{\kappa RT_A}\frac{dPa}{dt} \quad \text{..........} \quad (2)$$

The flow rate of the refrigerant passing through the suction port is assumed to be an isentropic, ideal flow, the heat transfer and dispation not being taken into consideration.

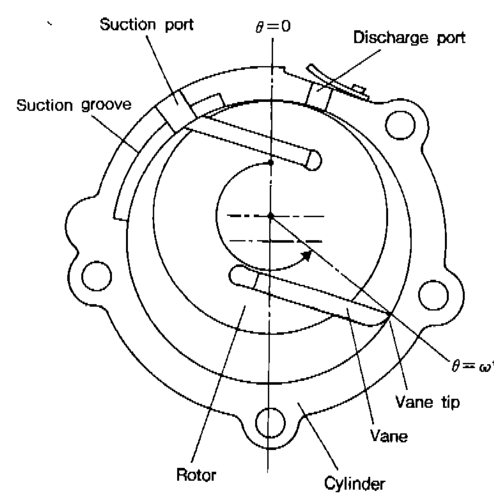

Fig.2 2 vane rotary type compressor(A)

$$G = a\sqrt{2g\gamma_A Ps\frac{\kappa}{\kappa-1}\left[\left(\frac{Pa}{Ps}\right)^{\frac{2}{\kappa}} - \left(\frac{Pa}{Ps}\right)^{\frac{\kappa+1}{\kappa}}\right]} \quad \text{.........} \quad (3)$$

The characteristic of the pressure Pa in the suction chamber is obtained by solving Eqs. (2) and (3 or 4) in combination. However, the following equation is derived when $Pa/Ps < \left(\frac{2}{\kappa+1}\right)^{\frac{2}{\kappa-1}}$

$$G = a\sqrt{2g\frac{\kappa}{\kappa-1}Ps\gamma_A\left(\frac{2}{\kappa+1}\right)^{\frac{2}{\kappa-1}}} \quad \text{.........} \quad (4)$$

Represent m by $m = Rr/Rc$, the volume $Va(\theta)$ of the chamber is obtained from[5]

$$V(\theta) = \frac{BRc^2}{2}\{(1-m^2)\theta + \frac{(1-m)^2}{2}\sin 2\theta - (1-m)\sin\theta$$

$$\times\sqrt{1-(1-m)^2\sin^2\theta} - \sin^{-1}[(1-m)\sin\theta]\}$$

$$+ \Delta V(\theta) \quad \text{.........} \quad (5)$$

Wherein, on condition of $0 < \theta < \pi$, $Va(\theta) = V(\theta)$ and on condition of $\pi < \theta < \theta s$, $Va(\theta) = V(\theta) - v(\theta-\pi)$.

Here, the position when the vane tip passes the axial seal is given as $\theta = 0$, the angle when the suction stroke ends is given as $\theta = \theta s$. $\Delta V(\theta)$ is a compensation term designed to compensate for the eccentric arrangement of the vanes with respect to the center of the rotor, and is usually of an order of 1 to 2%. Fig.3 shows the pressure characteristic in the suction chamber as obtained by Runge Kutta Gill method using the conditions shown in Table 1, with an initial condition of $t = 0$ and $Pa = Ps$, with a parameter of the rotation speed. Referring to Fig.3, the pressure Pa in the suction chamber has reached the upstream suction pressure $Ps = 3.18$ kg/cm^2 abs, before the suction stroke has been completed, i.e. at a position of $\theta = 260°$, when the rotation speed is low ($N = 1{,}000$ rpm). Therefore, no loss of pressure in the chamber takes place when the suction stroke is completed. As the rotation speed is increased, at the time of completion of the suction stroke ($\theta = 280°$ or $\psi = \pi$), the pressure drop is gradually increased. For instance, at a rotation speed of $N = 5{,}000$ rpm, the pressure drop ΔP in relation to the upstream suction pressure Ps is 1.25 kg/cm^2. This reduction in weight of the refrigerant, results in a remarkable reduction in refrigerating capacity.

Parameters	Symbols	2 vane rotary(A)	2 vane rotary(B)
number of vanes	n	2	2
theoretical displacement	Vth	85.5cc/rev	94cc/rev
angular vane tip position end for completing suction stroke	θs	280°	280°
Cylinder height	B	40mm	40mm
Cylinder radius	Rc	33mm	36mm
Rotor radius	Rr	26mm	29mm

Table. 1 Specifications and dimensions of compressors

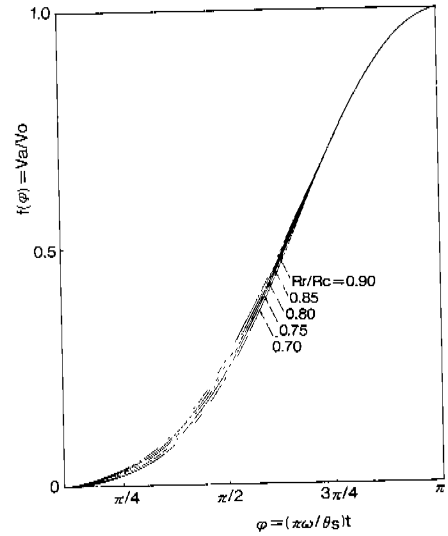

Fig.4 Volumetric curves

Instead of using equation (5), from which the volume Va of the chamber can be obtained, we now propose obtaining the correlation between the parameters and the efficiency of the performance capacity control by introducing the following approximation to equation (3 or 4) and (2). Let us suppose that Vo is the maximum suction volume and by using $\psi = \Omega t = (\pi\omega/\theta s)t$, the angle θ is transformed into ψ, where ψ varies from 0 to π. When we consider this condition, $f(0) = 0$, and $f'(0) = 0$ at $t = 0$, and $f(\pi) = 1$ and $f'(\pi) = 0$ at $t = \theta s/\omega$ at which the suction stroke has been completed, the approximate function $[f(\psi)]$ is defined. At this time the chamber volume Va is given by the following equation:

$$Va(\psi) \doteqdot Vo\cdot f(\psi) \quad \text{.........} \quad (6)$$

In Eq. (6) above, Vo and $f(\psi)$ represents the function such as Rotor radius (Rr), Cylinder radius (Rc), and eccentricity ($\varepsilon = Rc - Rr$) but $f(\psi)$ changes very slightly by Rr, Rc and ε. The smaller Rr/Rc is, the smaller compressor can be, under a fixed displacement. But, because of the large pressure gap, between the front and rear sides of the vane, there is a limit to the strength of vending stress in the vane. Although it is usually fixed with $0.7 < Rr/Rc < 0.9$, the difference in $f(\psi)$ is quite insignificant, as you can see from Fig.4.

Representing η by $\eta = Pa/Ps$, the following equation is the transliteration of Eq. (2) using Eq. (6).

$$G = \frac{Ps\Omega Vo}{RT_A}\{f'(\psi)\cdot\eta + \frac{f(\psi)}{\kappa}\frac{d\eta}{d\psi}\} \quad \text{..........} \quad (7)$$

Eq. (8) is derived from the transliterated of Eq. (3) as follows:

$$G = a\sqrt{Ps \cdot \gamma_A 2g \cdot \frac{\kappa}{\kappa-1}\left[\eta^{\frac{2}{\kappa}} - \eta^{\frac{\kappa+1}{\kappa}}\right]} \quad \text{.......... (8)}$$

Thus, the following Eq. (9) is derived by combining both Eqs. (7) and (8).

$$K_1 g(\eta) = f'(\psi)\eta + \frac{f(\psi)}{\kappa}\frac{d\eta}{d\psi} \quad \text{.......... (9)}$$

But the following Eq. (10) should be noted.

$$g(\eta) = \sqrt{\frac{\kappa}{\kappa-1}\left(\eta^{\frac{2}{\kappa}} - \eta^{\frac{\kappa+1}{\kappa}}\right)} \quad \text{.......... (10)}$$

The effective suction area changes in general by the angle (ψ), so K_1 becomes the fuction of ψ. But, the following dimensionless variable is derived, if assuming (a) is constant in the suction stroke, which will make K_1 constant. In Eq. (9) above, K_1 is a dimensionless form expressed by the following Eq. (11)

$$K_1 = \frac{a\theta s}{Vo \cdot \pi \cdot \omega} \cdot \sqrt{2gRT_A} \quad \text{.......... (11)}$$

In the case of a sliding, vane type rotary compressor, the theoretical displacement Vth is given by $Vth = n \times Vo$, where n represents the number of vanes, so that Eq. (11) can be written as follows.

$$K_1 = \frac{a\theta sn}{Vth \cdot \pi \cdot \omega} \cdot \sqrt{2gRT_A} \quad \text{.......... (12)}$$

In Eq. (9) above, K_1 is a constant which is determined by the kind of the refrigerant. Therefore, under the condition of K_1 being constant, the solution of Eq. (9), i.e. $\eta = \eta(\psi)$ can always be directly determined. This means that the drop of pressure in the chamber at the moment of the completion of the suction stroke is equal for all compressors which are constructed to have an equal value of K_1, and the control of refrigerating capacity is effected at an equal rate with respect to a given refrigerating capacity Q (Kcal) which is obtained when no refrigerating capacity control is effected.

In order to evaluate the effect on the refrigerating capacity control of the compressor, a parameter K_2 which depends on the dimensions of the compressor is introduced and is defined by the following equation.

$$K_2 = \frac{na\theta s}{Vth} \quad \text{.......... (13)}$$

Upstream temperature T_A and pressure Ps normally are controlled by an expansion valve in order to keep superheat $\Delta T = 10$ deg in the refrigerating cycle of car air-conditioners using refrigerant R-12. K_2 is determined by the displacement of a compressor, the number of vanes and so on, assuming T_A, g, R and κ of Eqs. (9) and (12) are fixed, and becomes an important evaluation index in determining the effectiveness of capacity control in relation to compressors. Hereafter, K_2 is named as the capacity control parameter.

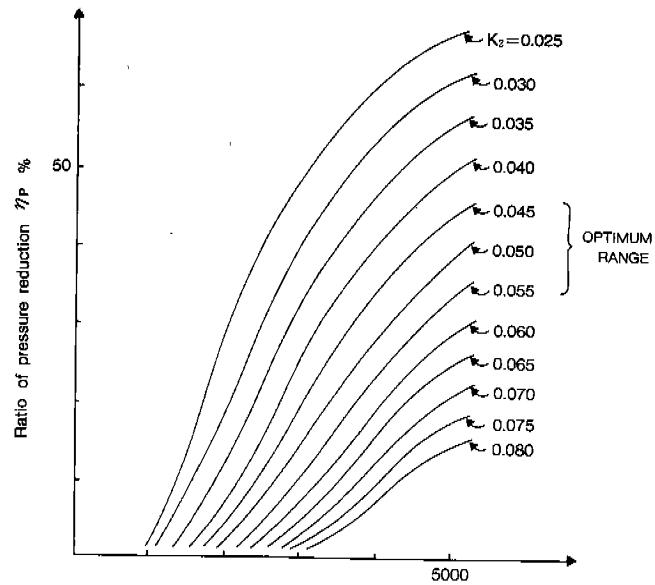

Fig.5 Chapacity control characteristics under the effective flow area is constantly maintained during suction stroke

[2] The ratio of reduction of refrigerating capacity and the ratio of pressure drop

For the purpose of evaluating the effectiveness of capacity control, the ratio of reduction of refrigerating capacity (η_Q) and the ratio of pressure drop (η_P) are defined as follows: Assuming [Was: the total mass of refrigerant, Pas: the pressure, Tas: the temperature] in the chamber when suction stroke is completed and (Wth) as the theoretical total mass of refrigerant, $Was = Pas \cdot Vo/R \cdot Tas$, $Wth = Ps \cdot Vo/R \cdot T_A$, and the rate of reduction of refrigerating capacity (η_Q) is derived by the following equation:

$$\eta_Q = \left(1 - \frac{Was}{Wth}\right) \times 100$$

$$= \left(1 - \frac{Pas}{Ps} \cdot \frac{T_A}{Tas}\right) \times 100 \quad \text{.......... (14)}$$

Also, the ratio of pressure drop is derived by the following form:

$$\eta_P = \left(1 - \frac{Pas}{Ps}\right) \times 100 \quad \text{.......... (15)}$$

Above (η_Q) would be considered as almost equal to the control effectiveness of refrigerating capacity. Also $\eta_P \doteqdot \eta_Q$ when $T_A \doteqdot Tas$

[3] The design of capacity control

Fig.5 shows the result which is obtained by the resolution of Eqs. (2) and (3 or 4) in combination, under the condition of $T_A = 283$, giving $\Delta T = 10$ deg as a superheat, and using K_2 as a parameter. As will be seen from Fig.5, it is possible to effectively obtain a pressure drop only during high speed operation, by suitably selecting the parameters of the compressor. The pressure reduction characteristic in relation to the speed has a region which may be referred to as an "insensitive region". With the discovery of this region, it became clear that the possibility of the realization of an efficient capacity control rotary type compressor was possible. This is the most important finding of our research. Capacity control can easily be designed by using both capacity control parameter (K_2) and Fig.5, for example.

(1) The ratio of reduction of refrigerating capacity (pressure drop) is less than 3% at a speed of $N = 1,800$ rpm.

(2) The ratio of reduction of refrigerating capacity at a speed of $N = 3,600$ rpm is greater than 20%.

The range of parameter K_2, which meets both of the above conditions, is given as follows:

$$0.045 < K_2 < 0.055 \quad \text{..........} \ (16)$$

Thus, by selecting the parameter a, θs, n and Vth of the compressor to meet the above range (16), it is possible to obtain a compressor which can self-control its refrigerating capacity and fulfill the above two conditions.

THE PATTERN OF THE EFFECTIVE FLOW AREA AND THE CAPACITY CONTROL CHARACTERISTICS

This section deals with capacity control characteristics when the effective flow area changes during the suction stroke. In the case of a 2 vane rotary compressor, as shown in Fig.2, the effective flow area of the suction groove, formed on the inner surface of the cylinder, determines the starting point of capacity control and the effective flow area of the suction port determines the slope of the refrigerating capacity drop ratio as related to rotational speed.

[1] Capacity control characteristic of a compressor with the suction groove formed on the cylinder

In the previous section only dealt with the case in which the effective flow area during suction stroke was constant. However, it can generally be said the effective flow area of suction passage of a sliding vane compressor, changes in accordance with the rotation angle in rotor, that is to say, with vane tip position. Therefore, capacity control parameter (K_2) is a function of angle (ψ). Here we will examine three cases to find out how the pattern of the effective flow area relates to capacity control characteristics.

(1) Change the former half of the effective flow area and fix the later half.

(2) Fix the former half of the effective flow area and change the later half.

(3) Change both the former and the later halves of the effective flow area.

The above mentioned three cases can be realized with the type of the compressor shown in Fig.2, in which the suction groove was formed on the inner surface of the cylinder wall. Defining a_1 as the effective flow area of suction port and a_2 as the

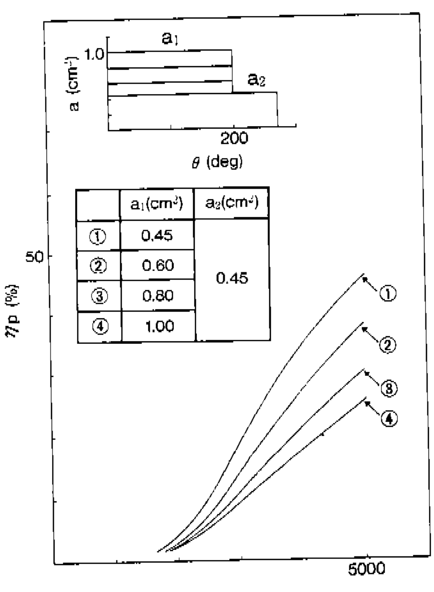

Fig.6 N-ηp characteristics of compressors(A) which effective flow area of former half (a_1) are changed and are fixed later half (a_2)

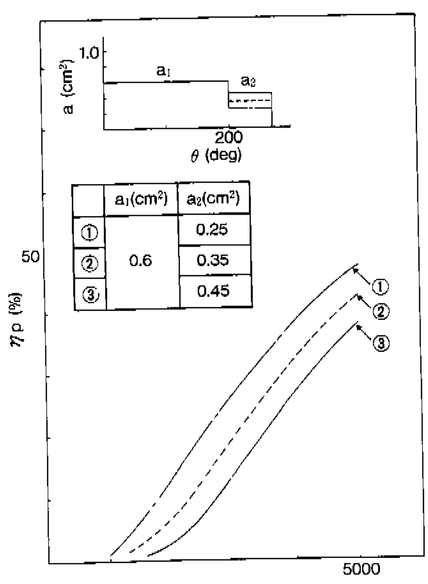

Fig.7 N-ηp characteristics of compressors(A) which effective flow area of former half(a_1) are fixed and are changed the later half (a_2)

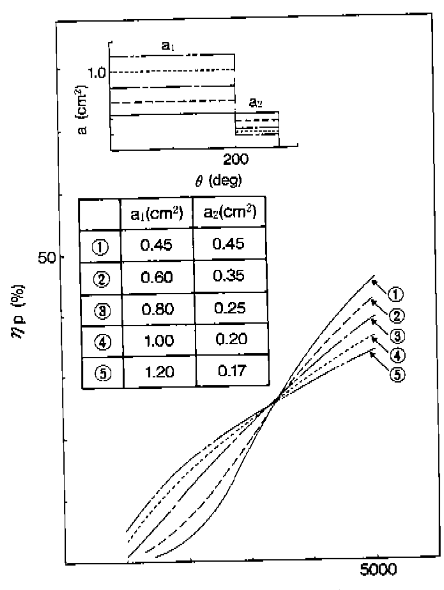

Fig.8 N-ηp characteristics of compressors(A) which are changed former(a_1) and later(a_2) halves of the effective flow area.

effective flow area of groove, the path of the refrigerant, which flows from the supplying side to the suction chamber, is shifted over by action of the suction stroke at $\theta = 200°$. In other words, $a = a_1$ before vane (b), following after vane (a), reaches the suction port and $a = a_2$ after vane (b) passes the suction port (see Fig.3).

In the case of the above mentioned (1), when the area of suction port (a_1) is changed, the ratio of pressure drop only changes during high speed operation and not during low speed operation. The effective flow area of the suction port has a great influence on controlling the operation of the refrigerating capacity during high speed operation, but not on the loss of refrigerating capacity during low speed operation ($N = 1,000 \sim 2,000$ rpm).

In the case of the above mentioned (2), we can easily see that refrigerating capacity is mainly determined by the effective suction area of the suction groove (a_2) during low speed operation.

The above mentioned (3) is a combination of (1) and (2), which shows that the less the grade difference of the former and the later half of the effective flow area, the bigger the slope in the ratio of pressure drop against the rotation speed becomes. It is also evident that a compressor in which the effective flow area, during the suction stroke, remains constant, can satisfy the conditions under which capacity control effectively works, i.e. small loss of refrigerating capacity during low speed operation and effective suppression of refrigerating capacity during high speed operation. In short, in the case of a two vane rotary compressor as shown in Fig.2, the suction area should be set to an adequate value and at the same time both the suction port and the suction groove should be formed so as to obtain a value of $a_2 \gg a_1$.

THE EXPERIMENTS

First, I would like to deal with the results of the refrigerating capacity (Q) and the volumetric efficiency (ηv) of a compressor having a constant effective flow area during the suction stroke, and in this way lead to a comparison of the refrigerating capacity drop ratio (η_Q) obtained from (ηv) with a theoretical pressure drop ratio (η_P).
A comparison of the theoretical and experimental results of pressure characteristics in the suction chamber, in the case in which the effective flow area of the suction stroke changes in two steps will follow.

[1] Experimental method

For the purpose of observing refrigerating capacity characteristics when the effective flow area of suction port is selected, we experimented with a compressor having a constant effective flow area during the suction stroke, and incorporated the results into our theory. Fig.11 shows the construction of our model compressor, which had a ring shaped spacer in the suction passage. The model also had a suction groove on the inner surface of

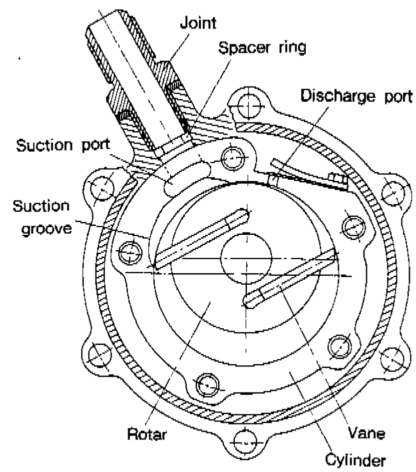

Fig.11 2 vane rotary type compressor (B) having spacer ring in suction passage

the cylinder wall which was deep and large enough so that the effective flow area during the suction stroke could be determined only by the spacer. With this model, the dispersion of characteristics caused when re-assembling a compressor can be avoided because the effective flow area can be changed by only changing the suction spacer(3).

Using the relation between the mass flow rate of air in each rotation position of rotor and the pressure ratio of the suction passage in the front and rear sections, we designed our compressor and calculated the effective flow area of the suction passage. Fig.12 illustrates the experimental method to measure the effective flow area, and Fig. 13 is the experimental result itself.

[2] Volumetric efficiency and the ratio of refrigerating capacity drop

Fig.14 shows the experimental results of refrigerating capacity (Q) and volumetric efficiency (ηv), varing the effective flow area of suction port by using different sized spacers. It is clear that the smaller (a), the effective flow area, is, the more (ηv), the volumetric efficiency, during high speed operation, the suppression amount of refrigerating capacity increases; nevertheless, a loss can be seen starting from a low speed operation of 1,000 rpm if (a) is under 0.40 cm^2. Volumetric efficiency at $N = 1,000$ rpm reaches $\eta v = 85\%$, when a is more than 0.40 cm^2. Therefore, the loss $\Delta\eta v = 100 - 85 = 15\%$ is assumed to be caused by the influence of factors other than pressure drop, such as internal leakage, internal heat transmission, lubricating oil mixed with the refrigerant and so on. Assuming a rotation speed that reaches $\eta v = 85\%$ as the starting point of capacity control, it is easy to discover that the starting point is around $N = 1,400 \sim 1,600$ rpm with $a = 0.56$ cm^2.

In order to separate the control effectiveness of refrigerating capacity by the pressure drop of the suction chamber from other losses, we arranged the experimental results, assuming $\eta_{vo} = 85\%$ as the standard characteristics and the difference from η_{vo} as the drop ratio of the refrigerating capacity ($\eta_Q = \eta_{vo} - \eta_v$). The result of comparing (η_p), the theoretically derived ratio of pressure drop, with (η_Q) is shown in Fig.15, gives us a good approximation of (η_p) and (η_Q). From this, it is clear that the method of capacity control using the parameter (K_2) shown in Fig.5, is of great practical application.

[3] Pressure characteristics in the suction chamber

The following deals with the experimental results of the pressure characteristics in the suction stroke. The results shown in Fig.16, indicate that the pressure Pa reaches at 1,000 rpm the supplying pressure ($Ps = 3.18$ kg/cm^2 abs) when the suction stroke is completed. However, pressure drop increases as rotation speed increases and drops to $Pa = 2.34$ kg/cm^2 abs, with $N = 4,000$ rpm. This is a good approximation of the theoretically derived value of $Pa = 2.23$ kg/cm^2 abs. A pressure transition occurs around $Va/Vo \simeq 0.75$, where the effective flow area changes suddenly from $a_1 = 0.6$ to $a_2 = 0.35$ cm^2, because the vane starts to run on the suction groove at this point. A strict comparison of the measured pressure in the suction chamber with the theoretical values is difficult because of the problem of calibration of instruments as well as the formerly mentioned error factors, which are mainly due to internal leakage. Even so, our results closely approximate the theoretically derived values.

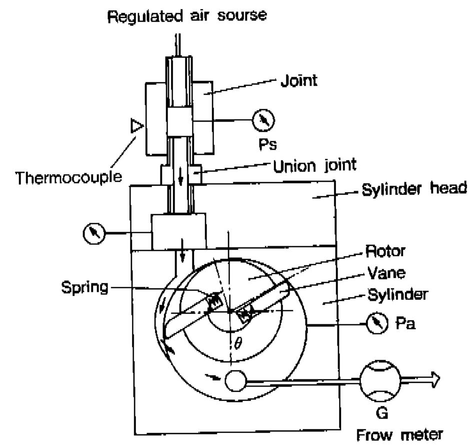

Fig.12 Details of measuring effective flow area

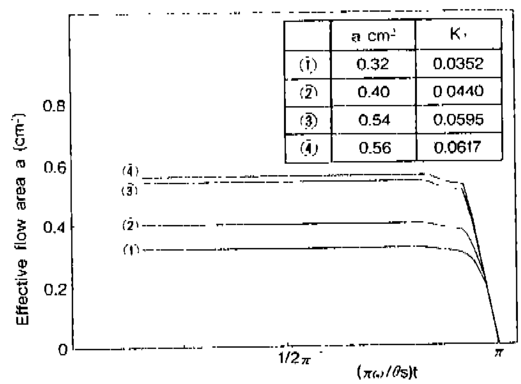

Fig.13 Measured effective flow area in suction stroke

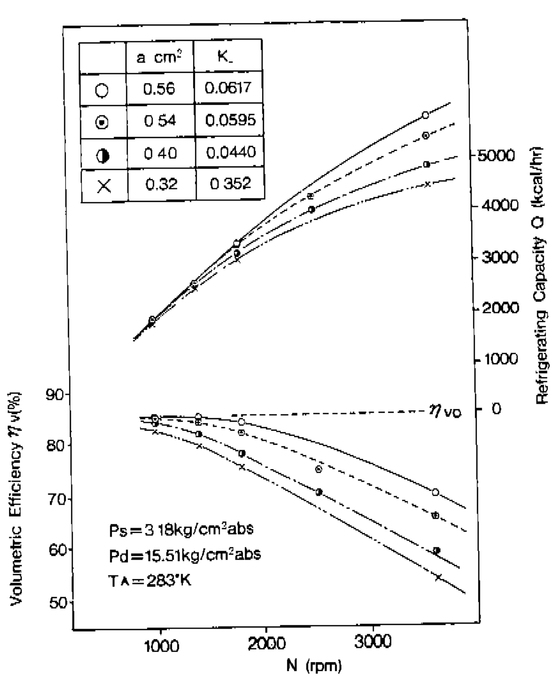

Fig.14 Measured refrigerating capacity and volumetric efficiency

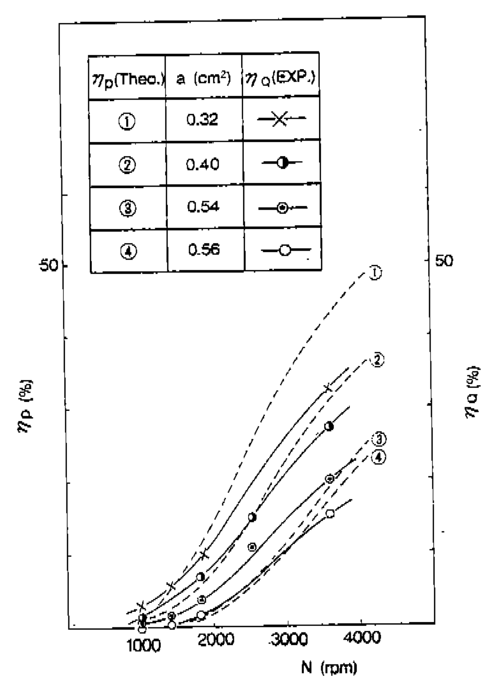

Fig.15 N-η_p and N-η_Q characteristics

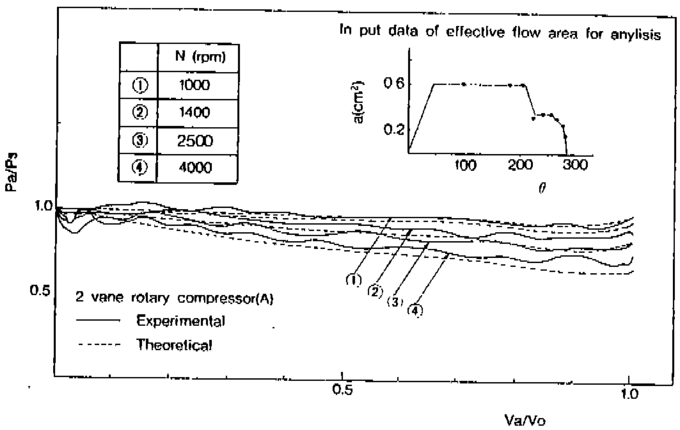

Fig.16 Pressure characteristics in suction chamber when rotation speed are changed.

CONCLUSION

By examining the refrigerant flow characteristics in the suction chambers under the suction stroke of a rotary type compressor, and investigating the influence of rotation speed, we were able to reach the following conclusions:

(1) We define "$K_2 = a\theta sn/Vth$" as the capacity control parameter for a rotary type compressor, in which the effective flow area in suction stroke is constant. In this condition a: effective flow area (cm^2), θs: angular vane tip position end for completing the suction stroke (radian), n: number of vanes, Vth: displacement (cc/rev).

A compressor with capacity control can be designed by using Fig.5 and K_2 which satisfies our two original necessary conditions of little loss of refrigerating capacity during low speed operation and effective suppression of refrigerating capacity during high speed operation.

(2) It is possible to select the effective flow area in the former half of the suction stroke and the later half of the suction stroke by constructing a compressor with a suction groove on the inner surface of the cylinder wall.

The effective flow area of the suction groove determines the starting point of capacity control and the effective flow area of the suction port determines the slope of the refrigerating capacity drop rate as related to rotational speed.

(3) Since the measured results of refrigerating capacity using a calorie meter and the experimental results of pressure characteristics contain errors caused mainly by internal leakages, it is difficult to strictly compare these results with the theoretical values.

However, it seems that our experimental results indicate that designing method of capacity control using our findings will be of practical use.

NOMENCLATURE

a	Effective flow area
a_1	Effective flow area of suction port
a_2	Effective flow area of suction groove
Cp	Specific heat at constant pressure
Cv	Specific heat at constant volume
g	Acceleration of gravity
G	Mass flow rate of refrigerant
K_1	See Eq. (12)
K_2	Capacity control parameter
N	Rotation speed
n	Number of vanes
Pa	Pressure in suction chamber
Ps	Upstream pressure of suction chamber
R	Gas constant
T_A	Upstream temperature of suction chamber
Ta	Temperature in suction chamber
Va	Chamber volume
Vo	Maximum volume of suction chamber
Vth	Theoretical displacement
η_p	The ratio of pressure drop
η_v	Volumetric efficiency
η_Q	The ratio of refrigerating capacity drop
θ	Angular vane tip position measured counter-clockwise from axial seal
θs	Angular vane tip position end for completing suction stroke
κ	Ratio of specific heats
ω	Angular velocity of rotor

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