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NEW CAPACITY CONTROL IN VANE ROTARY TYPE COMPRESSOR FOR AUTOMOTIVE AIR CONDITIONERS

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ABSTRACT

We have developed a vane rotary type capacity control compressor equipped with a capacity control mechanism which works on a principle where, in the compressor, arc shaped slider move over a wide range through ring shaped groove and automatically delay the substantial of suction process ending position (starting position of compression process).

During development, we performed theoretical analysis of control response and stability (where elementary cycle characteristics where added to compressor characteristics including pressure control valves and capacity control mechanism), and experimental performance analysis of control principles, and reviewed system characteristics of the entire refrigerating cycle. The developed vane rotary type capacity control compressor has the following features.

1. Capacity control mechanism with wide range from 10 to 100%
2. Pressure control mechanism with quick response and high stability.
3. Simple structure with few added parts

INTRODUCTION

Required functions and topics relating to automotive air conditioners are as follows.

1. How quickly is the ideal temperature attained?
2. How is temperature controlled to the ideal under all automobile operating conditions?
3. How can be driven the system economically through all seasons?
4. How can be the system driven without damaging automobile performance (running, feeling)?

In recent years there has been remarkable development in technology relating to variation by the compressor itself of refrigeration capacity (i.e. capacity control),1)-5) and the technology shows promise of satisfying the above requirements in a general way. In particular, 2-step capacity control types and linear capacity control types (where incline angle of wobble plate is continuously varied) have been applied in reciprocating type compressors.

Since the oil shock in Japan, the trend in automobiles has been toward economy, and automotive air compressors have been shifting to the Rotary type due to its compactness, light-weight, quietness and high efficiency.

In this context, there is a strong demand for progress in capacity control technology based on rotary compressors.

In response to these problems, we have developed a capacity control compressor which uses arc shaped sliders which slide through ring shaped grooves, is not much different in size from conventional compressors and which has a structure where volume is theoretically continuously variable from 10 to 100%.

The structure, principles, characteristics and effects of the new capacity control compressor for automotive air conditioning which we have developed are described below.

STRUCTURE AND PRINCIPLES

Fig. 1 is a partial section diagram indicating the overall structure of the capacity control compressor. The basis is a sliding vane type rotary compressor composed of a rotor which rotates in a round cylinder and vanes arranged freely from the rotor.

The key mechanical parts for capacity control are: a mechanical plate sandwiched between the front plate and cylinder, a arc shaped slider and spring which slide on the inside, and a pressure control valve fastened at the bottom of the rear plate. This is a simple structure with additional parts.

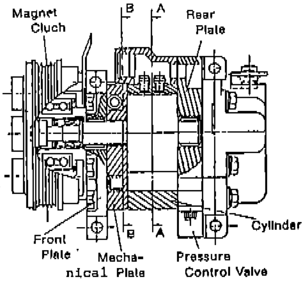


Fig. 1 Over all structure

Fig. 2 is a section diagram for Fig. 1 A-A which indicates the compression section. The diagram shows return ports arranged over a wide range which can be opened in the chamber surrounded by the rotor, cylinder and vanes, mainly in the compression stroke, and outlet port for return gas opened in the intake stroke. Return ports have a wide angle range and wide area so that cylinder volume can theoretically be added in the range 10 - 100%.

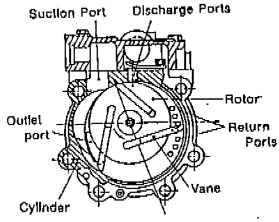


Fig. 2 Section diagram (A-A)

Fig. 3 is a principle model diagram for the Fig. 1 B-B section diagram which indicates the internal structure of the mechanical plate mentioned above.

Inside the mechanical plate, there is a ring shaped guide groove centered the same as the cylinder. A arc shaped slider with an attached spring is placed so that it can slide in the direction in which the above mentioned return ports close.

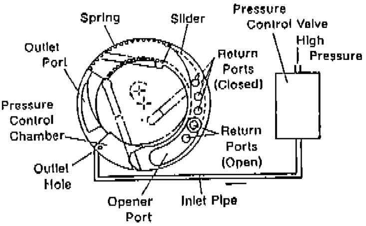


Fig. 3 Principle model (section B-B)

Pressure in the pressure control chamber rises according to the flow rate of gas sent from the pressure control valve, and the flow rate from the gap between the slider and guide groove, and the outlet hole, at the same time the slider moves to a position of balance with spring force. Then holes open in sequence starting from the hole on the near side in the rotor rotation direction of the return ports. The actual end of suction process in the cylinder chamber or the compression start position is delayed. The slider holds the opener port and has a structure which allows return gas inside to pass. The return gas further passes through the guide groove with spring and returns to the cylinder chamber of suction process from the outlet port.

ANALYSIS OF DYNAMIC BEHAVIOR

Theoretical analysis

Dynamic characteristic of the refrigerating cycle in a capacity control compressor are an important research topic for understanding the stability and response of capacity control mechanisms. We have performed theoretical analysis by modeling the dynamic behavior of control mechanisms (including the entire refrigerating cycle) which use pressure control valves to control capacity so that suction pressure becomes constant.

Fig. 4 is a model diagram for the refrigerating cycle system of the capacity control compressor. Formulas for relationships are introduced below while explaining the action of each component.

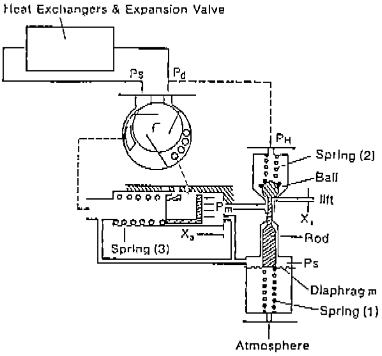


Fig. 4 Analytical model

* Pressure control valve
When compressor suction pressure goes below a certain setting, the steel ball is pushed up by the rod, producing lift x_1. That is,

$$-(k_1+k_2)X_1 = A_1(Ps-Pa) + A_2(P_H - Pm) + Fi \quad \ldots\ldots\ldots\ldots\ldots\ldots (1)$$

The flow rate dQ_1/dt produced by the steel by the steel ball life x_1 comes from the nozzle formula.

$$dQ_1/dt = C\pi d_2 X_1 \sqrt{2gK/(K-1)P_H \gamma_H \{(Pm/P_H)^{2/K} - (Pm/P_H)^{(K+1)/K}\}} \quad \ldots\ldots (2)$$

when $Pm/P_H < Pc$ then $Pm/P_H = \{2/(k+1)\}^{K/(K-1)}$

* Pressure control chamber
For the gap between slider and guide groove and the flow rate dQ_2/dt from the outlet hole:

$$dQ_2/dt = (1/R_3 + C\pi d_3^2/4)\sqrt{2gK/(K-1)Pm\gamma m\{(Ps/Pm)^{2/K} - (Ps/Pm)^{(K+1)/K}\}} \ldots (3)$$

when $Ps/Pm < Pc$ then $Ps/Pm = \{2/(K+1)\}^{K/(K-1)}$

Therefore, pressure control chamber pressure Pm is:

$$\gamma m A_3 dx_3/dt + A_3 X_3 \gamma m \, dPm/dt/Pm = dQ_1/dt - dQ_2/dt \quad \ldots\ldots\ldots (4)$$

Here, x_3 is amount of slider variation.

* Slider

The slider movement equation is expressed with the following formula.

$$k_3x_3 = A_3(P_m-P_s) - F_3i - \mu F_3 dx_3/dt/|dx_3/dt| + M_3d^2X_3/dt^2 \quad \ldots\ldots (5)$$

* Refrigerating cycle

To simplify the refrigerating cycle model, we assume first order lag system. That is,

$$P_s + T_0\, dP_s/dt = K_0X_3 \quad \ldots\ldots (6)$$

Solving the above simultaneous equations (1) - (6) enables us to obtain the restoration transition response for intake pressure Ps of the control system when suction pressure Ps has changed due to external causes.

Fig. 5 shows this feedback control system as a block diagram.

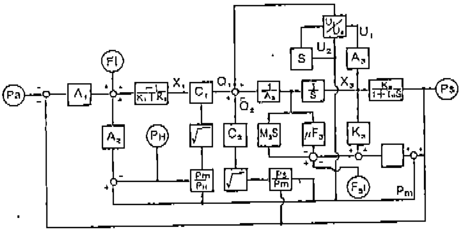
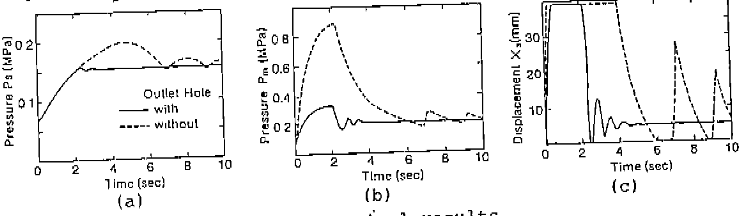


Fig. 5 Block diagram of Analytical model

Analytical results

Fig. 6 shows an example of analytical results. This analysis concerned transition response assuming that suction pressure Ps decreased in a step-by-step fashion from 0.16MPa to 0.07MPa. To understand the size of the outlet path as a factor which could affect control stability, two cases were compared: with and without outlet hole. Fig. (a) gives suction pressure, (b) pressure control chamber pressure and (c) slider variation transition response characteristics. The presence of outlet hole is indicated with a solid line. In contrast to the stabilization in 3-4s while performing small hunting in this case, with no outlet hole it appears that unstable action will continue even though 10s elapse. Moreover, in both cases, an extremely rapid response was obtained.



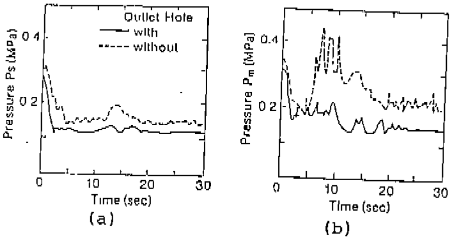
(a) (b) (c)

Fig. 6 Analytical results

Experimental analysis

Next we give results of measurement of dynamic characteristics through experiment.
Slider movement distance is determined by detecting the depth of the slit in the slider using an over-current type gap sensor integrated into the mechanical plate. A semi-conductor type pressure sensor is used for each pressure transition characteristic.

Figs. 7 (a) and (b) are the transition characteristics for suction pressure and pressure control chamber pressure at compressor start-up. Just as in theoretical analysis, we compared the difference in characteristics with and without outlet hole. With outlet hole (shown with solid line), suction pressure Ps in (a) suddenly decreased to the setting (0.12MPa) after start, and immediately control began, thus indicating high response. On the other hand, with the pressure control chamber pressure Pm of (b), in contrast to (a), the suction pressure reached the setting, then immediately rose, and after fluctuating for 20s stabilized at 0.15MPz. In contrast to this, without outlet hole (shown with dotted line), is both (a) (b) it required a long time to reach a stable state.
Next we describe results of measuring slider action, using Fig. 8.



(a)

(b)

Fig. 7 Experimental Results of pressure

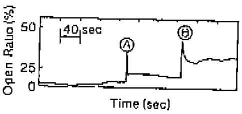


Fig. 8 Experimental Results of slider action

In this experiment, the fixed pressure setting of the pressure control valve which performs suction pressure stabilization was varied in a step-by-step fashion from 0.13 to 0.15 in (A) and from 0.15 to 0.18 in (B), to indicate slider transition response characteristics. As can be seen from the diagram, directly after changing the setting the slider opening suddenly enlarged and there was overshoot, but in a few seconds a stable value was apparently reached.

On the basis of these results, it is possible to make specifications for extremely rapid response and stability for the capacity control compressor we have developed.
Also, the results of theoretical analysis and experiment tend to agree.

PERFORMANCE OF COMPRESSOR

Fig. 9 shows variations due to the degree of openness of return ports for cylinder internal pressure of the capacity control compressor. In the diagram, there is a comparison in the return port full closed state with the PV diagram, and as the return ports gradually open, the substantial compression start point gradually moves, as can be seen by the thinning of the PV diagram.

Fig. 10 shows calorimeter characteristics using an suction pressure stabilized control valve. In this experiment, at 1000 rpm suction pressure was set to 0.16MPa with an expansion valve, and compressor speed was gradually increased in that state. Discharge pressure Pd was held constantly fixed at 1 MPa. The result was that suction pressure Ps became constant at 0.13MPa with N greater than or equal to 1200rpm at a volumetric efficiency η_v given as follows:

$$\eta_v = \eta_{vo} \times N_0/N \quad (Qe \propto \eta_v \cdot N = \text{const.})$$

Accompanying this was a similar trend of reduction in torque Tr. At this time, the return port degree of openness was different from the volumetric efficiency η_v reduction curve to raise speed, and clearly increased in direct proportion to the rotation speed.

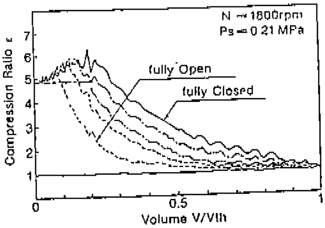


Fig. 9 Variations of PV diagram

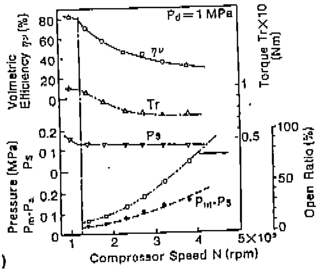


Fig. 10 Calorimeter characteristics

PERFORMANCE OF REFRIGERATING CYCLE

Finally we introduce in Fig. 11 an example of refrigerating cycle characteristics using the aforementioned capacity control compressor. Accompanying increases in compressor speed, suction pressure Ps gradually decreased, and at N = 1800 rpm the setting suction pressure Ps = 0.15MPa was reached. And, N > 1800 rpm suction pressure is fixed to constant by capacity control. In this case, discharge pressure Pa is shown by the rising curve up to 1800 rpm, but subsequently the circulation rate of refrigerant becomes constant, so the curve changes to dropping.

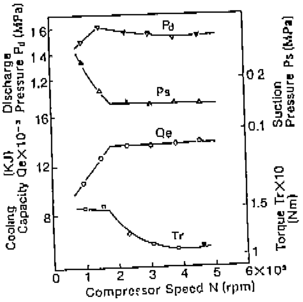


Fig. 11 Performance of Refrigerating cycle

Moreover, the refrigerating capacity becomes fixed after the start of capacity control. As for these results, the torque Tr started a drop at N ≥ 1800 rpm, and at N = 5000 rpm, in constant to the value of N = 1800 rpm, about a 35% reduction was reached.

Consequently, with a system employing a capacity control compressor there is no attachment and detachment of the electromagnetic cluth in the range of capacity control, so a stable value with no fluctuation is maintained, thus yielding comfortable temperature regulation and driving feeling (See Fig. 12)

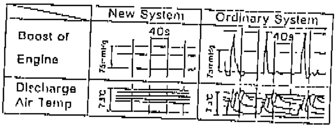


Fig. 12 Comparison of fluctuations

CONCLUSIONS

In the development a vane rotary type capacity control compressor equipped with a capacity control mechanism which engaged the method of automatically delaying the ending position of suction process, we studied dynamic behavior of capacity control mechanism and came to the following conclusions.

1. The control mechanism using a arc-shaped slider is simple, with few additional parts and enables capacity control over the wide range from 10 to 100%.
2. At the refrigerating cycle employed the capacity control compressor, with pressure control valve which stabilize suction pressure constant, refrigerating capacity becomes constant after the start of capacity control, and reduction of compressor torque is reached. Consequently yielding comfortable temperature regulation and driving feeling with no fluctuation in the range of capacity control.
3. Through analysis of capacity control compressor, results of theoretical analysis tend to agree with experimantal one and the methods we employed can establish outstanding control responce and stability characteristics.

NOMENCLATURE

A_1 : Pressure reception area of Diaphragm
A_2 : Pressure reception area of Steal ball
A_3 : Pressure reception area of Slider
C : Flow rate coefficient
d_2 : Diameter of outlet hole from pressure control valve
d_3 : Diameter of outlet hole from pressure control chamber
Fi : Sum of initial spring strengths of spring (1) and (2)
F_3i : Initial spring strength of spring (3)
F_3 : Side pressure applied to slider
g : Gravitational acceleration
Ko : Gain constant of Refrigerating cycle
k_1 : Spring constant of spring (1)
k_2 : Spring constant of spring (2)
k_3 : Spring constant of spring (3)
M_3 : Mass of Slider
N : Compressor rotation speed
No : Compressor rotation speed at control start
Pa : Atmospheric pressure
Pc : Critical pressure ratio

P_d : Compressor discharge pressure
P_H : High pressure
P_m : Pressure of pressure control chamber
P_s : Suction pressure of Compressor
Q_e : Refrigerating capacity
Q_1 : In-flow rate to pressure control chamber
Q_2 : Out-flow rate from pressure control chamber
R_3 : Flow resistance of slider side gap
T_o : Reponse time coefficient of Refrigerating cycle
T_r : Compressor required torque
V : Volume of cylinder chamber
V_{th} : Maximum theoretical volume of cylinder chamber
X_1 : Lift of steel ball
X_3 : Displacement of Slider
ε : Compression ratio
γ_H, γ_m : Gas specific weight
μ : Friction coefficient
κ : Specific heat ratio
η_v : Volumetric efficiency of Compressor
η_{v0} : Volumetric efficeincy of Compressor at control start

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