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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), 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. 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. 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
Fig. 3 Principle model (section B-B)
mentioned return ports close.

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.

* 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(P_s - P_a) + A_2(P_H - P_m) + F_i \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots 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Slider
The slider movement equation is expressed with the following formula.

\[ k_s x_s = \lambda_s (F_m - P_s) - P_s i - \mu F_s \frac{dx_s}{dt} - \frac{dx_s}{dt} + M_s \frac{d^2 x_s}{dt^2} \]  

(5)

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

\[ P_s + T_0 \frac{dP_s}{dt} = K_s x_s \]  

(6)

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

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

Analytical results
Fig. 6 shows an example of analytical results. This analysis concerned transition response assuming that suction pressure \( P_s \) 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.

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Fig. 5 Block diagram of Analytical model

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 $P_s$ 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 $P_m$ of (b), in contrast to (a), the suction pressure reached the setting, then immediately rose, and after fluctuating for 20s stabilized at 0.15MPa. 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.

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 diagrams, 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 COMRESSOR

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 \( \eta_v \) given as follows:

\[ \eta_v = \eta_v \times N_s \times \eta_v = \text{const.} \]

Accompanying this was a similar trend of reduction in torque \( T_r \).

At this time, the return port degree of openness was different from the volumetric efficiency \( \eta_v \) reduction curve to raise speed, and clearly increased in direct proportion to the rotation speed.

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 \( P_s = 0.15 \)MPa was reached. And, \( N > 1800 \) rpm suction pressure is fixed to constant by capacity control. In this case, discharge pressure \( P_d \) 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.
Moreover, the refrigerating capacity becomes fixed after the start of capacity control. As for those results, the torque $T_r$ started a drop at $N \geq 1800$ 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 clutch 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)

**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 experimental one and the methods we employed can establish outstanding control response and stability characteristics.

**NOMENCLATURE**

- $A_1$: Pressure reception area of Diaphragm
- $A_2$: Pressure reception area of Steel ball
- $A_3$: Pressure reception area of Slider
- $C$: Flow rate coefficient
- $d_1$: Diameter of outlet hole from pressure control valve
- $d_2$: Diameter of outlet hole from pressure control chamber
- $F_{i1}$: Sum of initial spring strengths of spring (1) and (2)
- $F_s$: Side pressure applied to slider
- $g$: Gravitational acceleration
- $K_o$: Gain constant of Refrigerating cycle
- $K_{13}$: Spring constant of spring (1)
- $K_{13}$: Spring constant of spring (2)
- $K_{13}$: Spring constant of spring (3)
- $M_s$: Mass of Slider
- $N$: Compressor rotation speed
- $N_o$: Compressor rotation speed at control start
- $P_a$: Atmospheric pressure
- $\phi_c$: Critical pressure ratio


Pd : Compressor discharge pressure
Pu : High pressure
Pim : Pressure of pressure control chamber
Ps : Suction pressure of Compressor
Qe : Refrigerating capacity
Q1 : In-flow rate to pressure control chamber
Q2 : Out-flow rate from pressure control chamber
R' : Flow resistance of slider side gap
To : Response time coefficient of Refrigerating cycle
Tr : Compressor required torque
V : Volume of cylinder chamber
Vth : Maximum theoretical volume of cylinder chamber
X1 : Lift of steel ball
X3 : Displacement of Slider
\( e \) : Compression ratio
\( \gamma_h, \gamma_m \) : Gas specific weight
\( \mu \) : Friction coefficient
K : Specific heat ratio
\( \eta_v \) : Volumetric efficiency of Compressor
\( \eta_{ve} \) : Volumetric efficiency of Compressor at control start

REFERENCE