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REDUCTION OF NOISE AND OVER-COMPRESSION LOSS
BY SCROLL COMPRESSOR
WITH MODIFIED DISCHARGE CHECK VALVE

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ABSTRACT
In this report, we will explain our development of an optimum discharge check valve structure used in scroll compressors, based on a simplified model calculation and an experiment. We found that the behavior of check valve in the calculation analysis was consistent with actual valve behaviors if only we apply damping coefficient and adhesive strength at the face of valve seat/valve retainer as experiment constants, even though the model is a simple one that uses the pressure change equation derived from the first law of thermodynamics, the compressive one-dimensional flow equation and the valve motion equation. Based on this method, we have accomplished a chattering noise reduction and over-compression loss reduction by modifying the shapes of scroll wrap, discharge port, valve and valve retainer, without losing the fundamental function of preventing reverse rotation on compressor shutdown.

INTRODUCTION
For scroll compressors, a check valve is generally installed at the discharge port to prevent reverse rotation on operation shutdown, but it is well known (ref.1) that the valve makes noise called chattering under a certain operation condition away from the design built-in pressure ratio.

Based on this theory, up to the present our company adopted a non-chattering design by adding a limitation to the size of channel area (valve port) and to the allowable lift. When we changed the scroll wrap (ref. 2), which was expected to ensure high efficiency and high reliability while the present check valve structure remained unchanged, however, noise was newly confirmed in an area where chattering did not occur before. For the purpose of preventing chattering and improving efficiency, we have created a simple analysis tool that can evaluate the characteristics of valve behavior, and performed measurements of valve lift and pressure of compressor internal plenum using an actual machine. Here, we will present, with analysis results, an outline of the simple analysis method, measures and their effects, to attain the previously stated purpose.
CHECK VALVE STRUCTURE

Conventional Check Valve Structure

Fig. 1 shows a conventional check valve structure. Gas from the discharge port located at the fixed scroll head discharges to the high-pressure plenum through the hub plenum, the valve port of the partition plate, and the check valve. The valve has a flat plate shape without preload of springs, etc. and a free structure, while the valve retainer has a hole to enhance the valve closing function during shutdown.

Scroll Wrap Shape for High Efficiency and High Reliability

Fig. 2 shows cross-sectional images of scroll wraps close to the discharge area. A conventional scroll type has uniform thickness wrap, however, the new scroll type has a non-uniform thickness wrap, where the thickness increases as it approaches the scroll center. This can increase rigidity at the scroll center area where high pressure is added, and can not only increase reliability but can also decrease flank side leak due to wrap deformation abatement and radial direction leak due to seal length elongation. Furthermore, by modifying the shape of the internal wrap to multi-circular arcs, which can increase the channel area after start of discharge, we can decrease over-compression. For both the conventional wrap and the new wrap, specifications are the same for cylinder volume and the built-in volume ratio.

As stated above, an unchanged conventional check valve structure installed with this newly developed scroll wrap, particularly under high pressure ratio conditions, caused a new chattering noise.
SIMPLE ANALYSIS MODEL

Gas condition change in an arbitrary plenum chamber of the compressor has an energy balance from the view of the first law of thermodynamics; and assuming an adiabatic change there, we can obtain the equation (1) of pressure change.

\[
\frac{dp}{dt} = \kappa \cdot \left( \frac{dm}{dt} m - \frac{dV}{dt} V \right) \tag{1}
\]

Where \( p \), \( m \), \( V \), \( \kappa \), and \( t \) represent pressure, mass, volume, specific heat ratio, and time respectively.

Assuming that gas flow between each plenum chamber is a compressive one-dimensional flow, then.

\[
\frac{dm}{dt} = C \cdot A \cdot \sqrt{2 \rho \kappa \frac{p_i}{p_i - 1} t} \left\{ \left( \frac{p_i}{p_o} \right)^{\kappa} - \left( \frac{p_o}{p_i} \right)^{\kappa} \right\} \tag{2}
\]

Where \( C \), \( A \), and \( \rho \) represents flow coefficient, channel area, and density respectively. The subscript \( i \) indicates upstream side and \( o \) indicates downstream side.

Now we can express valve movement as an equation (3).

\[
Mv \cdot \frac{d^2h}{dt^2} = Cp \cdot A_{port} (p_o - p_c) + Cd \cdot A_{port} \frac{p_o}{2} \left( \frac{dm_{hi}}{dt} \right) / A_{port} + \left( \frac{dh}{dt} \right)^2 - Cd' \cdot Av \left( \frac{dh}{dt} \right)^2 + Cf \cdot \frac{dh}{dt} + \text{Mv} \cdot g - Ca \cdot Aa \tag{3}
\]

\[
Aa = Av - A_{port} (h \equiv 0)
\]

\[
Aa = Ar (h \equiv h_{max})
\]

Where \( \text{Mv} \): check valve mass; \( h \): valve lift; \( A_{port} \): valve port area; \( \text{Av} \): valve surface area; \( \text{Aa} \): valve contact area at face of valve retainer;

Also, each coefficient shows as follows.

\( \text{Cp} \): pressure coefficient and function of non-dimensional number (\( h/D_{port} \)). 
\( \text{Cd} \) and \( \text{Cd}' \): drag coefficient and both a function of Reynolds Number. 
\( \text{Cf} \): damping coefficient. 
\( \text{Ca} \): adhesive coefficient, assuming that valve lift \( h \) acts only when less than a certain value near valve seat or near valve retainer. \( \text{Cf} \) and \( \text{Ca} \) are experimental constants, which are incorporated with experimental values.

Fig.3 Analysis model
Fig. 3 shows the analysis model. For simplification, we have considered 5 chambers after the discharge process, namely the A-discharge pocket, B-discharge pocket, Middle pocket, Hub plenum, and High pressure plenum; assuming that there is no leak from the suction process until the compression process.

Where the A-discharge pocket is a plenum chamber consisting of a fixed scroll interior wall and an orbiting scroll exterior wall; the B-discharge pocket is a plenum chamber consisting of a fixed scroll exterior wall and an orbiting scroll interior wall; and the Middle pocket is defined as below. The Middle pocket in this report, as shown in Fig. 4, is a plenum chamber of the virtual nearest distance, formed between the scroll top end and the facing scroll interior wall. When adopting the definition above, as the volume of the Middle pocket becomes extremely small with the proceeding rotation from the start of discharge, and it becomes difficult to have a distinction among it and the A/B chambers. On that condition, we can assume the Middle pocket has the same status (pressure, density) as the A/B chambers. We therefore regarded A, B, and the Middle pocket as one chamber in the latter half of rotation and we analyzed accordingly. Each channel area of gas among chambers is shown in Fig. 5.

Comparison of Channel Area
Fig. 6 shows the differences of effects (new and conventional) of the scroll wraps, in regard to each channel area change.

The angle of rotation was indicated on the basis of the start of discharge process and we made the channel area, which is divided by the discharge port area, a non-dimension. The channel area of the new scroll wrap is designed to be larger than that of a conventional scroll wrap, especially due to the effects of \( A_{AH} / A_{BH} \), which are the path from the flank side to the Middle pocket. This contributes, as stated before, to over-compression abatement under the rated condition close to the design built-in pressure ratio.

EXPERIMENTAL RESULT AND ANALYSIS RESULT

Experimental Equipment
Fig. 7 shows the location of sensors installed inside of compressor. Valve lift is measured by an eddy current type gap sensor installed on the top of valve retainer, and piezo-electricity type pressure sensors are installed in discharge pockets, a hub plenum, and a high pressure plenum. Besides, an accelerometer is installed at the body shell to evaluate compressor vibration at shutdown.

![Fig. 4 Middle Pocket](image)

![Fig. 5 Each channel area](image)

![Fig. 6 Comparison of Channel Area](image)

![Fig. 7 Sensors installed in a Scroll Compressor](image)
Experiment Result and Analysis Result

Fig. 8 shows the comparison with the experiment results and analysis results both at rated condition and high-pressure ratio condition, using conventional check valve system. We set crank angle on the basis of start of discharge, and plotted valve lift against the allowable lift value and pressure difference between hub plenum and high-pressure plenum.

Regarding the conventional scroll wrap, at a rated condition (operation pressure ratio 3.7) valve is moving in a floated state without contacting with valve seat and valve retainer, however, at high pressure ratio (6.0), due to the increment of reverse flow rate, valve displacement drops on an average, and shows cyclic behavior with a slight contact with valve seat. Collision velocity on this time is around 0.2 m/s, and we do not hear any abnormal sound.

On the contrary to above, regarding the new scroll wrap, valve is moving in more floated state, at a rated condition, compared with conventional one. And at high-pressure ratio, valve movement increases by the increment of reverse flow change due to the effect of discharge port channel enlargement. Valve shows cyclic behavior no longer, it shows unstable behavior and causes, what is called, chattering.

The experimental result, compared with analysis, have a same tendency though there is some absolute value difference in valve lift and in pressure difference. Under a high-pressure condition where the actual machine caused chattering, valve made a big movement at every 2 cycles. Behavior on analysis showed every 2 cycles, too and this proves that valve behavior can be said to be well simulated by analysis.

Fig. 8 Valve Lift and Pressure Difference
Check Valve Structure Modification

Fig. 9 shows a new check valve structure, which were developed to prevent a chattering under the operation of high-pressure ratio. In type C-specification, the valve is a ring shaped, and installed with internal valve seat, besides an allowable lift is reduced to small, and in type D-specification, a magnet was installed at valve retainer of conventional specification. The scroll wrap is a new type adopted with a non-uniform thickness and multi-circular arcs.

Fig.10 shows the comparison of the experimental result and analysis result for above two specifications. We find that both in C- and in D-specification, the valve is working stable at close to allowable valve lift status, even at the operation of high-pressure ratio where B-specification caused a chattering. Naturally, there is no collision velocity.

In C-specification, the valve port channel area was reduced in comparison with a conventional specification, and as a result it made the effect of pressure fluctuation small and slow in movement, and in D-specification, the improvement was considered to be done with stability by magnetic force.

Fig. 11 shows each chamber’s Pressure-Time behavior and an example of P-V diagram converted from this (P-θ). We can observe that a large over-compression loss in B discharge pocket on A-specification was reduced on D-specification.

<table>
<thead>
<tr>
<th>Table.1 Over-compression Loss and Under-compression Loss</th>
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<td>Scroll Wrap</td>
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Fig.9 New Check Valve Parts

Fig.10 Valve Lift and Pressure Difference
Table 1 shows an analysis comparison of over-compression loss and under-compression loss for specification A to D. Under-compression loss that was originally small in B pocket on A-specified was increased a little due to scroll wrap change. We, however, attained that loss abatement of 7% on C-specified and about 10% on D-specified at a total loss, against on conventional A-specified.

Characteristic of Shutdown

Most important function of check valves is, as a matter of course, preventing reverse rotation on shutdown. Fig. 13 shows a typical example of reverse rotation phenomenon. This is a data based on the specification aimed at valve operation stability by installing a spring as shown Fig. 12 and is plots of valve lift, vibration acceleration, and Hub plenum, High pressure plenum behavior at compressor shutdown. Further, time is expressed as non-dimensional number on the base of the valve close time at rated condition under A-specification.

The valve of full open status during operation closes at t time later after current stops. During this Δt, as gas causes reverse flow from a high pressure plenum into a low pressure plenum, therefore, the compressor makes reverse rotation. When we look at acceleration behavior, we can observe violent vibration increment due to reverse rotation after vibration stops once. At this moment, it accompanies also violent noise by gas expansion. Besides, we can observe from this figure that the compressor continues rotating with inertial force and that Hub plenum pressure is dragged down momentarily to vacuum status.

Δt time from current stop until check valve closes is one of the evaluation indexes, to know the degree of reverse rotation phenomenon. Fig. 14 shows the result of Δt time measured on C and D-specified developed this time and on conventional A-specified. Δt time is prescribed by operation differential pressure ΔP (discharge pressure-suction pressure) before stoppage. Namely, we can say that the bigger operational differential pressure is, the sooner the check valve reacts. When compared with each specification, we can say that C-specified is almost same as A-specified, and Δt time is large for D-specified, especially at low differential pressure region. At
low differential pressure condition, valve shutdown time $\Delta t$ is large indeed, but at the low differential pressure, the energy of reverse rotation becomes surely smaller. Fig. 15 shows shutdown characteristic of low differential pressure ($\Delta P=0.55 \text{ MPa}$) on D-specification. If differential pressure before shutdown is small, acceleration at reverse rotation is reduced as figure shows, even if $\Delta t$ is at same degree. Actually, the reverse rotation sound at $\Delta P=0.55 \text{ MPa}$, on D-specification, is hardly heard at human hearing level.

CONCLUSION

Regarding measures on preventing chattering and improvements in efficiency stated in this report, based on the results of both the simple analysis and the experiment, we have found following findings:

(1) The result of numerical model analysis which is combining a pressure change equation, a compressive one-dimensional flow equation, and a valve motion equation, applying damping coefficient and adhesion coefficient as experimental constant, have obtained a good coincidence with the check valve behaviors of actual machine operation.

(2) At an operation condition away from built-in pressure ratio, especially where at the high pressure ratio condition which have a large compression shortage, a valve suddenly shuts down due to low discharge pocket pressure, and collide with valve seat and cause noise.

(3) Even if in the same operation pressure ratio, as discharge pocket pressure differs due to the difference of channel area change, and this affects valve, too. The larger the channel area change is, the more valve velocity increases, therefore, it tends to cause noise easily.

(4) On the other hand, at close condition to built-in pressure ratio, the larger the channel area change is, the more we can decrease over-compression loss, compared with reverse flow loss increment.

(5) To prevent a chattering, the stabilizing method using a ring valve and magnet is effective.

(6) Reverse rotation characteristic at valve shutdown can be measured by operation differential pressure just before shutdown, the smaller differential pressure is, the more valve takes time to shut down. As the energy of reverse rotation becomes small, we hear few as reverse rotation noise.

As stated above, we have developed the compressor of a chattering noise reduction and an efficiency improvement, by adopting C-specification, without loosing shutdown function.

REFERENCES
