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CONTROLLING DISCHARGE VALVE CLOSING IMPACT IN SCROLL MACHINES

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ABSTRACT

This paper analyzes scroll wrap geometry with respect to volumetric outflow at the time of porting and it’s influence on valve reed position and closing impact. A relation including displacement rate at the time of porting is developed to show how geometry of the inner wrap and the compressor operating conditions influence discharge valve impact. Tradeoffs in selecting profiles for minimum impact and low clearance volume are analyzed.

NOMENCLATURE

\( \rho_d \) – density in clearance volume under valve
\( A_b \) – port area under valve
\( A_p \) – port area leading to intermediate chamber
\( A_v \) – area of valve opening
\( C \) – valve damping
\( c \) – speed of sound
\( C_c \) – critical damping
\( C_d \) – coefficient of contraction of vena contracta
\( c_v \) – correction in speed of sound for vena contracta
\( e \) – base of natural logarithms
\( H \) – scroll compressor wrap height
\( k \) – polytropic coefficient
\( K \) – valve stiffness
\( n \) – mass flow rate through the port
\( M_v \) – valve mass
\( P_d \) – pressure in discharge pipe
\( P_{dv} \) – pressure in control volume
\( R_s \) – length of vector normal to scroll wrap profile
\( R_x \) – length of vector tangential to scroll wrap profile
\( R_{or} \) – orbiting scroll orbit
\( t_0 \) – time at start of backflow
\( V \) – clearance volume
\( z \) – valve distance from the seat
\( z' \) – valve velocity
\( z'' \) – valve acceleration

INTRODUCTION

Scroll compressors operate with a constant built-in volume ratio and thus do not require dynamic valving with its associated flow losses. This accounts in part for their reputation of high efficiency. However, at high compression ratios excessive “undercompression” losses may easily eliminate performance advantages. Accordingly, scroll machines for these applications are available with a discharge valve which limits undercompression losses.

The apparent added reliability risk is small since only a discharge valve is added. Experience in reciprocating compressors demonstrates that suction valves are more prone to exhibit high stress due to oil “stiction” effects (Khalifa and Liu, 1996). However, the introduction of discharge valves in scroll compressors presents a new risk in the form of the sudden backflow and resulting closing impact of the valve reed on the valve seat that can take place at the time of discharge porting of the sealed compression chamber.

Valved Applications

The effect of undercompression in fixed volume ratio devices is well known (Lifson and Bush, 1995). The undercompression loss is a result of an underpressurized compression chamber being ported to the system high
High pressure vapor fills the chamber, only to be pushed back out during the discharge process. This irreversible mixing of vapor leads to high discharge temperature and high power.

Excessive discharge temperature may occur in heat pumps, refrigeration applications or, in the case of small transportation systems, where limited heat exchanger sizes drive the system operation to high pressure ratios compared to a conventional residential or commercial systems. High temperatures cause oil and refrigerant breakdown, generating acids or solid contaminants which lead to other, more spectacular failure modes. Traditionally, liquid refrigerant injection has effectively controlled discharge temperatures. It also carries a penalty in cost, efficiency, and system capacity, often in turn requiring larger compressors which also brings higher cost.

The energy which goes into heating of the discharge vapor represents an extra power draw and subsequent efficiency loss. This can translate into added operating expense or, in the case of transport refrigeration, a restriction on the number of units that can be operated from a mobile power source of limited electrical capacity.

Reduced energy consumption has been a driver in HVAC markets where system efficiency levels are increasingly regulated. As environmental concerns expand to envelop the refrigeration and transportation industries, energy consumption and Total Environmental Warming Impact (TEWI) become important considerations as well.

Valve Impact

In valved applications, opening impact, against the valve stop or retainer, and closing impact, against the valve seat, are typical reliability concerns. Both can lead to premature fatigue and induced breakage of valve elements. In the case of the scroll, opening impact is generally not a concern. Valve stop design to limit stress has become quite advanced in the development of reciprocating and rotary compressors, where opening speeds are high compared to the slow pressure ramp of the scroll. However, closing impacts in scroll compressor can be quite high.

During the major part of the discharge cycle, where flow rates are high, the valve reed is held against the stop by the dynamic flow pressure. However, as the scroll compressor discharge chamber reaches the end of the discharge process, the flow rates drop and the valve starts to close. At this time the distance of the valve reed from the seat is mainly controlled by the discharge flow rate and valve stiffness.

At the end of the discharge process, porting occurs where the next set of sealed compression chambers is opened to the discharge passage. If the pressure in the intermediate compression chamber is below the discharge pressure, then discharge vapor rushes into the intermediate chamber and the valve closes under the influence of the reversed flow. The closing valve reed impacts against the seat, with an impact velocity related to the pressure differential across the valve (the driving force), and the distance of the reed from the valve seat at the time of pressure reversal (the distance over which it can accelerate). This closing impact can lead to noise and, more importantly, fatigue and premature failure of the valve reed at high pressure ratio or high discharge pressure conditions.

Closing impacts are not much of an issue in traditional reciprocating and rotary compressors. In a reciprocating machine, where the piston speed becomes zero at the end of the discharge process, the valve reed is allowed to settle on the seat before the gradual reversal of pressure begins as the piston starts accelerating downward again. In a rotary compressor, where there is more of a sudden reversal of pressure, the reversal occurs when the displacement rate is very low and the valve reed, if not seated, is very close to the seat.

Design Responses

Particular to scroll compressors, one design variation, reminiscent of some rotary compressor applications, is to place a pulse chamber near the valve to reduce the magnitude of the reverse flow. If this is combined with a resonator feature, it can also contribute to reduction of valve noise and flutter from resonances in the discharge chamber and port (Motegi, et al., 1997). However, such chambers require careful design and are often sensitive to
changes in the acoustic velocity in the refrigerant due to changes in discharge temperature. The increase in clearance volume also penalizes performance.

In another method a small leak passage just before porting is introduced on the working portion of the scroll (Fujitani, et al., 1994). The controlled leak serves to better equalize pressure between the compression chamber and discharge prior to porting, thus reducing the initial pressure difference. While this is a valid approach, it results in additional leakage losses (reminiscent of undercompression losses) and is sensitive to operating conditions. The machining or fabrication of scroll components is also somewhat more complex.

High pressure ratio scroll wraps for valved applications are typically designed for minimum clearance volume. The resulting wrap design may often have difficulty with hydraulic hammer, or so-called vapor lock, at the end of the process where the port is covered by the scroll wrap (Cailliat, 1988). A common solution is an undercut near the center of the wrap or other relief as shown in Figure 1. This design feature inherently introduces a non-zero mass flow rate at porting which results in the valve being positioned away from its seat at the time backflow begins.

Figure 1. Scroll Set Geometry Near Porting and Valve Closure

WRAP DESIGN AND ITS EFFECT ON VALVE IMPACT

A number of geometric curves have been used in the past to form the working surfaces of scroll machines. The most common profile adapted by the industry for high pressure ratio refrigeration applications combines the use of a standard involute of a circle for the outer wrap portions and offset circles for the inner portions.

In a scroll compressor the intermeshing wraps of the fixed and orbiting scroll form a series of pairs of crescent shaped pockets. As the orbiting scroll moves through its orbit, the contact points between wrap walls of the fixed and orbiting scroll move continuously from an initial point at the outer periphery toward a terminal point at the inner end of the working scroll wrap surface. Bush and Beagle (1992) describe the position of the contact point using two mutually perpendicular vectors $R_s$ and $R_g$. The volumetric flow rate through the port after the valve is open, neglecting flow losses, can be approximated by the rate of volume change of the scroll compression pocket.

$$\frac{dV(t)}{dt} = 2 R_s(t) H R_{wr} \omega \tag{1}$$

For one form of the offset circle geometry, $R_s$ may become zero at the end of the discharge process and thus the valve velocity is inherently low, at most. However, this is the design which often requires the scroll wrap cutout or other relief at the very inner portion of the involute to avoid hydraulic lock as shown in Figure 1. However, this also introduces increased clearance volume. More importantly, it can reduce valve reliability by increasing the impact velocity on the valve seat because the presence of the cutout or relief introduces a finite value of $R_s$ at the wrap end. This in turn causes the valve reed to be located away from the seat at the time of porting, which may result in high valve impact velocity as it closes under the influence of the backflow of refrigerant through the valve.

Valve Governing Equations

Pressure changes very rapidly under the valve reed at the time of porting. The analysis of the valve reed behavior between the time the reverse flow is initiated and the reed valve is closed must consider unsteady effects.
Compressibility effects also exist since flow is expanding from a high pressure region of the discharge pocket into a much lower pressure region of the compression pocket. It is not uncommon to have pressure ratios of 5 or higher between the discharge and intermediate pocket with a shock wave propagating between these two pockets.

![Diagram of valve behavior](image)

The valve behavior is further complicated by the fact that both the area between scroll members and the gap between the orbiting scroll tip and the port edge, through which the flow is passing, is rapidly increasing. At the same time the inflow of vapor into the volume through the valve is becoming more restricted as the valve moves downward toward the seat. As more refrigerant is carried out of the volume than carried in, the pressure in the volume decreases rapidly causing the downward accelerated movement of the valve reed and its ultimate impact on the seat. The initial distance of the valve reed from its seat at porting plays a crucial role in determining the final valve impact velocity. At the limit, there would be no impact at all if the valve is already positioned on the seat prior to the initiation of the reverse flow. On the other hand, if the valve is positioned relatively far away from the seat just prior to porting, then the valve has a greater distance over which it can be accelerated before impacting on the seat. This distance is controlled by the amount of lift provided by the velocity and density of vapor through the valve opening just prior to porting. The definition of common terms used in the analysis and pictorial behavior of the reverse backflow is shown in Figure 2.

We use a lumped parameter approach in defining the valve stiffness, mass, and location from the seat. We are not interested in higher order responses at this time. We assume that the pressure in the intermediate compression pocket is held constant, since the closing valve time is very short compared to a full discharge cycle and any increase in the intermediate compression chamber during this closing time is small and can be neglected. We also assume negligible fluid momentum effects for valve position and motion. The time required to accelerate fluid flow is very short compared to the time required to accelerate the much heavier valve element toward closing.

We begin by assuming that the flow leaving the clearance volume through the port is choked, and variations in the speed of sound due to slight changes in vapor temperature are neglected. This seems reasonable, as discharge pressure exceeds the intermediate chamber pressure into which the refrigerant flow is expanding by a factor of at least 3:1 or more for most of the refrigeration applications in valved scroll compressors. Also, as will be seen in the analysis, the valve impacts on the seat well before the pressure in the volume under the reed valve has a chance to equalize with or even approach discharge pressure. With this assumption in mind, we can write

\[
\dot{m}(t) = A_p(t) \cdot \rho_{vr}(t) \cdot c_v \cdot c_v
\]  

Where coefficient \(c_v\) accounts for creation of vena contracta and flow choking in the area smaller than \(A_p\).

We next assume that the mass flow leaving the clearance volume into the intermediate chamber is much greater than the flow entering the volume through the valve. The "driving" pressure drop through the port is much higher (flow through the port is choked) than the pressure drop through the valve. Also, the area for out-flowing vapor through the port is rapidly increasing as the port is uncovered by the wraps while the area through the valve is decreasing as the valve reed begins to approach the seat. Basically, the additional effect of apparent increase in the clearance volume due to backflow through the valve is neglected. We also discount the term \(dV(t)/dt\), as it affects the amount of mass getting through the valve (we still, of course, account for this term to determine the distance of the valve from its seat at time of porting). A more comprehensive analysis will be presented elsewhere by Lifson.
and Bush (1998) in which flow through the valve is treated in more detail.

Using the above assumptions, the conservation of momentum equation, combined with equation 2, may be written as:

\[ \frac{dP_{av}(t)}{dt} = -(A_p(t) \cdot c \cdot c_v/V) \cdot k \cdot P_{av}(t) \]  

(3)

Using the initial condition for \( P_{av}(t) \) to be equal to \( P_a \) and solving equation 3 the pressure inside the volume, \( P_{av}(t) \) is

\[ P_{av}(t) = P_a \cdot e^{-c \cdot c_v \cdot k \int_0^t A_p(t) \, dt} \]  

(4)

Neglecting inertia effects, we can write the second order differential equation describing the valve movement as:

\[ K \cdot z + C \cdot z' + M_v \cdot z'' = P_a (1 - e^{-c \cdot c_v \cdot k \int_0^t A_p(t) \, dt}) A_b \]  

(5)

where the right hand term represents the force acting on the valve reed. This ordinary differential equation can be solved by imposing the initial conditions of zero initial velocity and acceleration for the reed and an initial position \( z \) given by

\[ z(t_0) = c_p \cdot P_a \cdot A_b \cdot (dV(t_0)/dt)/(A_x(t_0) \cdot C_x)^2/(2K) \]  

(6)

which is based on the flow area under the open reed which is required to accommodate the outflow through the valve at the time of porting. We also assume a constant damping factor \( C \). This equation can be solved explicitly to arrive to the value of \( z \) and \( z' \) as a function of the scroll compressor operating characteristics and geometry.

**Analytical Results**

Table 1 summarizes analysis results for a given scroll design. It compares, on a normalized basis, the influence of various design and operating factors on calculated parameters, including in particular the valve impact velocity.

<table>
<thead>
<tr>
<th>Variable Parameters</th>
<th>Case A Baseline</th>
<th>Case B</th>
<th>Case C</th>
<th>Case D</th>
<th>Case E</th>
</tr>
</thead>
<tbody>
<tr>
<td>Clearance Volume Length, ( L )</td>
<td>1</td>
<td>2</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Discharge Pressure, ( P_d )</td>
<td>1</td>
<td>1</td>
<td>1.5</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Valve Damping, ( C/C_c )</td>
<td>10%</td>
<td>10%</td>
<td>10%</td>
<td>100%</td>
<td>10%</td>
</tr>
<tr>
<td>Flow Rate at Porting, ( dV/dt_0 )</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>2</td>
</tr>
<tr>
<td>Calculated Parameters</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Valve Impact Velocity, ( z'(t_{impact}) )</td>
<td>1</td>
<td>0.83</td>
<td>1.24</td>
<td>0.91</td>
<td>1.37</td>
</tr>
<tr>
<td>Distance from Seat at Porting, ( z(t_0) )</td>
<td>1</td>
<td>1.22</td>
<td>0.91</td>
<td>1.05</td>
<td>1.15</td>
</tr>
<tr>
<td>Time to Impact from Porting</td>
<td>1</td>
<td>1.22</td>
<td>0.91</td>
<td>1.05</td>
<td>1.15</td>
</tr>
<tr>
<td>Pressure, ( P_{av}/P_d ) at Valve Closure</td>
<td>1</td>
<td>1.13</td>
<td>1.04</td>
<td>0.98</td>
<td>0.92</td>
</tr>
</tbody>
</table>

Case A in the table is the baseline design. In cases B through E one parameter is changed at time to show its effect on computed results. For convenience, the changed parameter in each case is indicated by slightly larger and bold font.
The increase in the clearance volume length is beneficial as pressure in the volume decreases more gradually. However, the clearance volume increase results in additional performance penalty due to re-expansion and would alleviate the impact problem only partially. An increase in discharge pressure has significant detrimental effect on the valve closing impact as it both provides higher valve forces as well as inducing higher initial lift at porting. The increase in the valve damping is also beneficial. However, its effect becomes pronounced only at very high damping values which may be impractical.

Higher volumetric flow rate $dV/dt$ results in higher vapor velocity and higher valve lift at porting and is clearly detrimental to valve impact. The vapor velocity at porting is controlled by the geometry of the inner scroll profile. For example, if the volumetric flow rate at porting can be brought to zero, the valve will already be located on its seat, no reverse flow will be present at all, and the valve impact eliminated. This should be the design engineer’s ultimate goal while providing adequate port relief to avoid trapped volumes.

CONCLUSION

The geometry of the inner portion of the profile is the key in reducing the valve closing impact velocity at the end of the discharge process. Scroll wraps with geometries selected for zero or near zero volumetric flow at porting will essentially eliminate backflow (and the resulting initial valve lift) and will provide negligible valve impact velocity. Other options such as increasing valve damping or increasing clearance volume under the valve, in general, provide only partial alleviation of the valve impact and also introduce additional losses. Often, design constraints call for a compromise in selecting the most appropriate scroll wrap geometry by trading off volumetric flow at porting with compressor port size, relief volume between the scroll wraps (prevent vapor lock), size of the clearance volume, leakage control, etc.

High pressure ratio and especially high discharge pressure also significantly affect the severity of the valve impact. These application requirements are often beyond the design engineer’s control. As high pressure refrigerants such as R410A gain acceptance in the market, some existing valved scroll machines may need to be redesigned to withstand the associated increased valve impact velocity.

REFERENCES


