The influence of port shape on gas pulsations in a screw compressor discharge chamber

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ABSTRACT

Gas pulsations in suction and discharge chambers are widely accepted to be a significant source of noise in screw compressors. This paper analyses the influence of both the compressor operating conditions and its geometric characteristics on the level of gas pulsations generated in its discharge chamber. The time dependency of the flow area in the discharge port is identified here as an important parameter influencing gas pulsations. It is shown how the gas pulsation amplitude in the discharge chamber can be reduced by optimization of its port shape.

NOMENCLATURE

A	area
dA_i	finite small area of the discharge port
φ	angular coordinate in cylindrical coordinate system
$\varphi_{i1}, \varphi_{i2}$	lower and upper boundary for finite small area
h_{in}	specific enthalpy flow in chamber
h_{out}	specific enthalpy flow from chamber
i	arbitrary position of the finite small area
m	fluid mass in chamber
$\dot{m}_{in}$	mass flow in chamber

$\dot{m}_{out}$	mass flow from chamber
n	number of finite divisions of the discharge port
p	fluid pressure in chamber
$\dot{Q}$	heat transfer between fluid and chamber surrounding
R	radial coordinate in cylindrical coordinate system
R_i , R_{i+1}	lower and upper boundary for finite small area
$R_{\max}$, $R_{\min}$	maximum and minimum radius of axial discharge port
ρ	fluid density
t	time
U	internal energy
v	fluid velocity
V	fluid volume in chamber
z	coordinate Z in cylindrical coordinate system
z_i , z_{i+1}	lower and upper boundary for finite small area

Subscripts

A	axial discharge port	M	female side of the discharge port
dc	discharge chamber	out	discharge chamber outlet
dp	discharge port	R	radial discharge port
F	female side of the discharge port	wc	working chamber

1. INTRODUCTION

The suction and discharge chambers of a screw compressor are connected to the working chambers only periodically. This creates unsteady flow, variation of mass and hence pressure pulsations in them during both the suction and discharge processes through the compressor ports. These create both vibration and noise. The amplitude of the gas pulsations in the compressor discharge chamber is much higher then those in the suction chamber. Therefore many authors consider the gas pulsations in a screw compressor discharge chamber to be the main source of noise.

Intensive research on gas pulsations in screw compressors started in 1986 when *Fujiwara and Sakurai* [0] first measured gas pulsation, vibration, and noise in a screw compressor. After that *Koai and Soedel* [5,6] 1990, developed an acoustic model in which they theoretically analyzed the low pulsation in a twin screw compressor and investigated how it was related to compressor performance. More recently *Sangfors* [12] in 1999, *Tanttari* 2000 [14] and *Huagen et al* 2004 [4] developed mathematical models for the prediction of gas pulsations in screw compressor ports.

All these authors explored the influence of various screw compressor working and design parameters upon gas pulsations in the compressor suction and discharge chambers. Some authors, such as *Koai and Soedel* [5] recognized the influence of the compressor port area and recommended that the effect of the port shape on noise be further investigated. *Mujic at all* 2005 [11] reported that changing the shape of a screw compressor discharge port leads to different gas pulsation levels in the discharge chamber. Reduction of pulsation amplitude leads to lower overall levels of compressor noise. This led to further investigation of this effect.

2. PARAMETERS WHICH INFLUENCE GAS PULSATIONS

In order to analyze the influence of the discharge port shape upon the gas pulsations in the discharge chamber and to quantify the advantage which optimization of the discharge port shape offers, the most important operating and design parameters which affect gas pulsations in compressor discharge chamber are discussed here.

2.1. Working conditions

Discharge pressure. Koai and *Soedel* [5] noticed that the gas pulsations as function of the discharge pressure have a minimum. This is also confirmed later by *Sangfors* [12]. According to *Huagen et aIl* [4], this minimum corresponds to the discharge pressure that matches the machine built-in volume ratio. *Koai and Soedel* [5] claim that this minimum does not correspond exactly to that pressure, while *Gavric* [2] consider that it coincides with the small undercompression.

Pressure difference. According to *Koai*, *Soedel*, *Sangfors* and *Gavric*, the pressure difference between the working and the discharge chamber at the start of the discharge process is the most important factor in the flow pulsations. Since the gas pressure in the working chamber at the end of compression is determined by the built in volume ratio of the machine, while the gas pressure in the discharge varies according to the instantaneous flow conditions, it follows that pressure difference cannot be defined as an independent parameter. However, its use for comparing compressors with different built in volume ratios, is more appropriate than the discharge pressure.

Compressor speed. S*angfors* [12] and *Huagen* [4] concluded that the amplitude of the pressure pulsations during the discharge process increases with the rotational speed.

Oil influence. Oil has an attenuating influence upon the noise generation process. According to *Sangfors* [12] this is significant at harmonics higher than the 3rd order but *Tanttari* [14] states that this is noticeable only above the 5th order.

2.2. Compressor design parameters

Compressor clearances. Reduction of leakage within the compressor, caused by smaller clearances, increases the noise generated in the discharge port. *Soedel* [5] and *Sangfors* [12] reported that for the same working conditions, changed compressor clearances result in a change in the working chamber pressure and fluid flow through the discharge port.

Discharge chamber length. According to *Sangfors* [12] the gas pulsations and generated SPL are affected by the discharge chamber length. This influence is significant and the sound pressure level expressed as a function of chamber length also has a minimum.

Number of rotor lobes. According to *Sangfors* [12], the number of rotor lobes influences the noise level. Rotors consisting of more lobes generally generate a lower sound pressure level in operation then those with fewer lobes.

2.3. Main factors affecting gas pulsations

Although there are many parameters which cause the pressure to vary in the discharge chamber only two of them are important. To identify them, it is necessary to consider the process in detail. Pressure variations are generated when unsteady flow occurs into or out of the chamber.

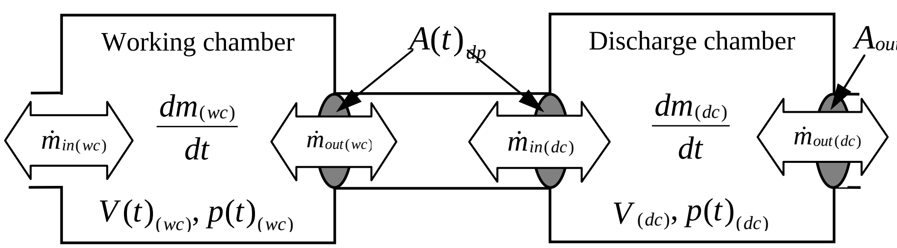

Figure 1 Screw compressor discharge system

The flows between the working and discharge chambers and out of the chamber are presented in Figure 1. These must satisfy the continuity equation for both discharge (inlet) and outlet conditions:

$$\dot{m}_{in(dc)} = \rho v A(t)_{dp} \quad \text{and} \quad \dot{m}_{out(dc)} = \rho v A_{out} \tag{1}$$

Equation 1 shows that the mass flow depends on the instant gas density and the velocity. It also depends on the size of the discharge chamber inlet and outlet areas. The size of the discharge chamber outlet opening A_{out} is constant while inlet area

$A(t)_{dp}$ varies with time. The inlet area $A(t)_{dp}$ is the actual cross sectional area of the discharge port exposed to the working chamber.

Changes in any of these parameters will affect the pressure difference between the working and discharge chambers and will result in pressure variation in the discharge chamber. The pressure difference between the working and discharge chambers occurs as a consequence of the screw compressor process.

The effect of changing the size of the discharge port and the outlet opening is of the same order of magnitude as those of changes in density or velocity. This implies that the gas flows and pressure variations can be altered by modifying the size of the inlet and outlet areas. The size of the outlet is constant and it is good to make it as large as possible. A larger area will cause lower pressure difference for the same flow between discharge chamber and pipe. That will the stabilize pressure in the discharge chamber.

However, the pressure difference and mass flow between the working and discharge chambers cause variations of pressure. The influence of the cross sectional area of the discharge port upon gas pulsations in the discharge chamber has been reported by *Mujic at al* [11]. It is shown that various shapes of the discharge port produce different gas pulsations in the compressor discharge chamber.

3. MATHEMATICAL MODELLING OF THE DISCHARGE FLOW PROCESS

The 1-D model of screw compressor discharge process is based on a model described by *Stosic and Hanjalic* [13]. Features which improve analysis of the discharge process have been introduced. The first is the simultaneous connection of more than one compressor working chamber with the discharge chamber. This allows calculation of flow when two working chambers are connected as shown in Figure 2. It can be seen from Figure 2, that when one working chamber starts its discharge the previous chamber has still not finished. Also, before the discharge of one chamber ends, the discharge of the next chamber starts. This affects the mass and energy inflow into discharge chamber. The equations for mass and energy conservation have been modified to account for this phenomenon.

The equations for conservation of internal energy for compressor are given for the working chamber (2) and the discharge chamber (3) as follows:

$$\frac{dU_{(wc)}}{dt} = \dot{m}_{in(wc)}h_{in(wc)} - \dot{m}_{out(wc)}h_{out(wc)} + \dot{Q} - \left(p\frac{dV}{dt}\right)_{(wc)}, \tag{2}$$

$$\frac{dU_{(dc)}}{dt} = \dot{m}_{in(dc)}h_{in(dc)} - \dot{m}_{out(dc)}h_{out(dc)} + \dot{Q} \tag{3}$$

The enthalpy inflow into the discharge chamber consists of flows from more then one working chambers. Due to that equation for the energy inflow into discharge chamber takes into account outflows from *n* working chambers:

$$\dot{m}_{in(dc)} h_{in(dc)} = \sum_{i=1}^{n} \dot{m}_{out(wc)i} \cdot h_{out(wc)i} \tag{4}$$

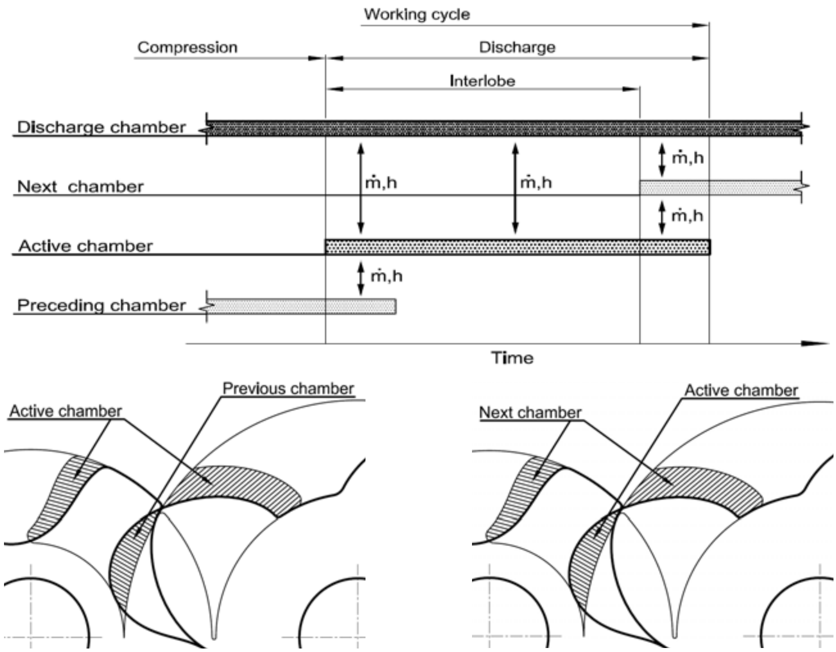

Figure 2 Screw compressor discharge process

The equations of mass conservation for the working chamber (5) and discharge chamber (6) are:

$$\frac{dm_{(wc)}}{dt} = \dot{m}_{in(wc)} - \dot{m}_{out(wc)}, \tag{5}$$

$$\frac{dm_{(dc)}}{dt} = \dot{m}_{in(dc)} - \dot{m}_{out(dc)} \tag{6}$$

Here mass inflow into discharge chamber also consists of outflows from the *n* working chambers which are connected to discharge chamber at the same time:

$$\dot{m}_{in(dc)} = \sum_{i=1}^{n} \dot{m}_{out(wc)i} \tag{7}$$

Each of the mass flow rates in the above equation satisfies the continuity equation:

$$\dot{m}_{out(wc)i} = \left(\rho v A\right)_{(wc)i} \tag{8}$$

The area A is the cross sectional area of the discharge port connected to the working chamber *i*. This area is of complex shape and changes in time. Since the accuracy of the simulation depends on the size and gradient of that area, it is necessary to avoid any approximation and to accurately calculate the size of the discharge port area for any arbitrary rotor position and any shape of the discharge port and rotor profiles. Also to analyze the influence of different shapes of discharge ports on gas pulsations, it is necessary to quantify their difference in the 1-D model. This can be done providing the 1-D model with the actual size of the discharge port area of different ports.

In order to satisfy these requirements a numerical integration method has been developed. A finite number of small areas dA_i across axial and/or radial discharge port with respect of the rotor position is shown in
Figure 3. These are bounded by the edges of both the discharge port and the rotors and vary both in space and time.

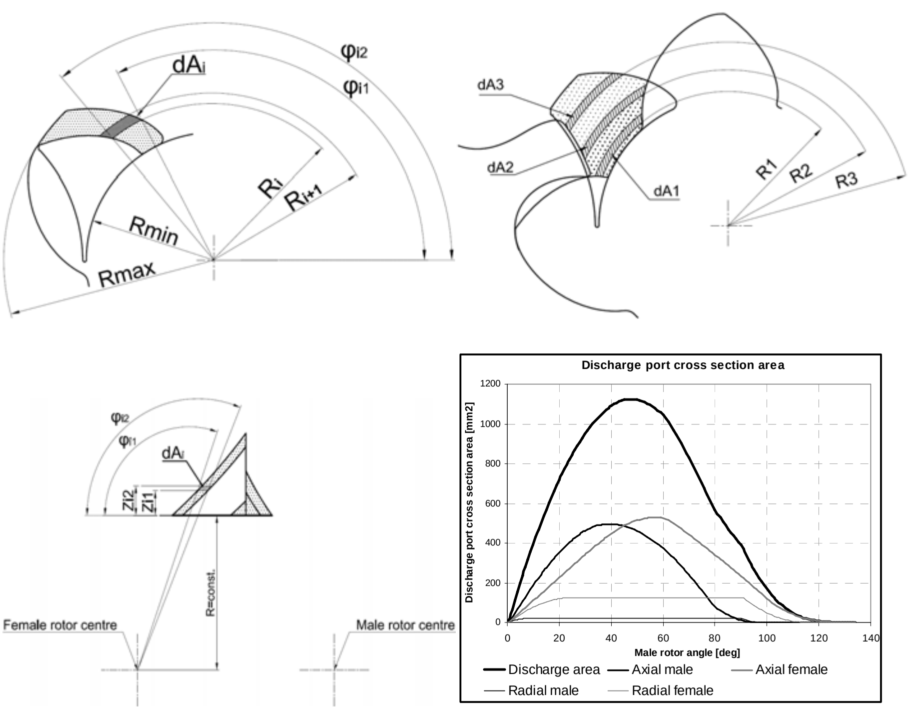

Figure 3 Principle of the discharge port area calculation

The size of the discharge port for any arbitrary rotor position is calculated as:

$$A_{dpi} = \sum_{i=1}^{n_M}\left(\int_{R_{MAi}}^{R_{MAi+1}} dr \int_{\varphi_{1MAi}}^{\varphi_{2MAi}} R_i \cdot d\varphi\right) + \sum_{i=1}^{n_F}\left(\int_{R_{FAi}}^{R_{FAi+1}} dr \int_{\varphi_{1FAi}}^{\varphi_{2FAi}} R_i \cdot d\varphi\right)$$
$$+\sum_{i=1}^{n_{MR}}\left(\int_{z_{MARi}}^{z_{MRi+1}} dz \int_{\varphi_{1MRi}}^{\varphi_{2MRi}} R_M \cdot d\varphi\right) + \sum_{i=1}^{n_{FR}}\left(\int_{z_{FRi}}^{z_{FRi+1}} dz \int_{\varphi_{1FRi}}^{\varphi_{2FRi}} R_F \cdot d\varphi\right) \quad (9)$$

Two different shapes of the screw compressor discharge port are shown in Figure 4, the original one, as a light line and the new one, as a bold line. Although they are different in shape, the ports keep the same built in volume ratio. The new shape is proposed to reduce the gas pulsations in the discharge chamber by avoiding sudden port opening and thereby reduce sudden flows in the cases of *under* or *over* compression.

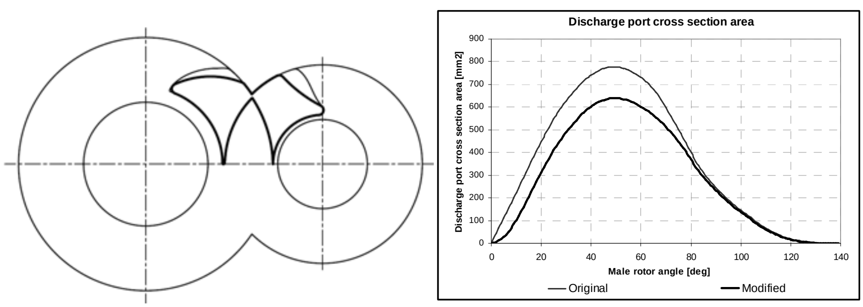

Figure 4 Shape and cross sectional area of two different discharge ports

A comparison of the estimated and measured pressure pulsations in the discharge chamber are shown for the original port in Figure 5 and for the new discharge port shape in Figure 6. In both cases, the experimental results are shown as light lines, and the simulated results by bold lines.

The measurements were carried out on an industrial compressor with a 4/5 lobe configuration and a male rotor diameter of 120 [mm]. Measuring points cover a range of compressor speeds from 2000 rpm to 4000 rpm at a discharge pressure of 8 bar. Another set of points covers discharge pressures from 5 bar to 12 bar at the same speed. The pressure pulsations in the discharge chamber were measured with a piezzo electric pressure transducer. The only difference between the two sets of tests was in the discharge ports and all other parameters were maintained identical.

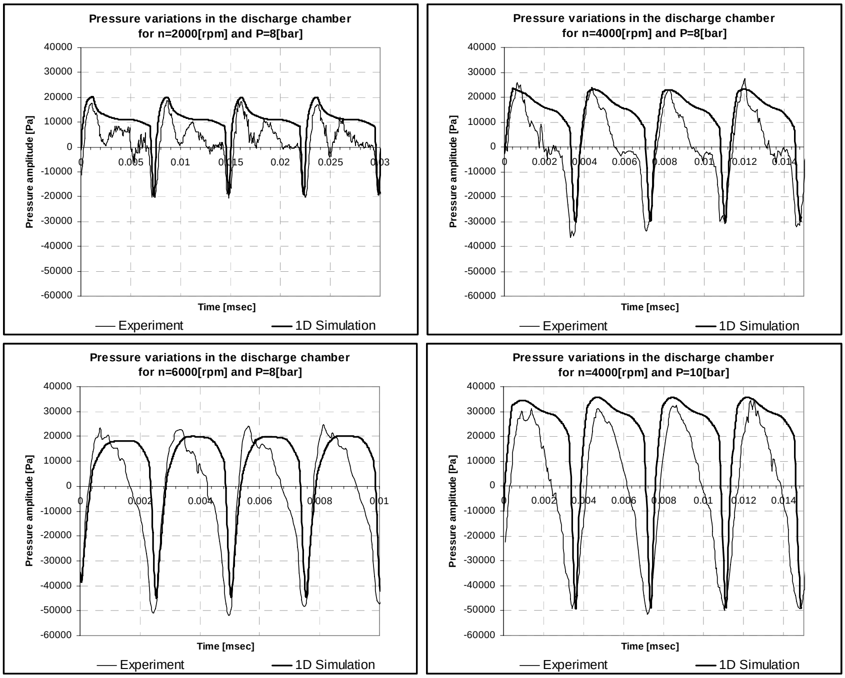

Figure 5 Comparison of experimental and results simulated by a one dimensional model for original discharge port

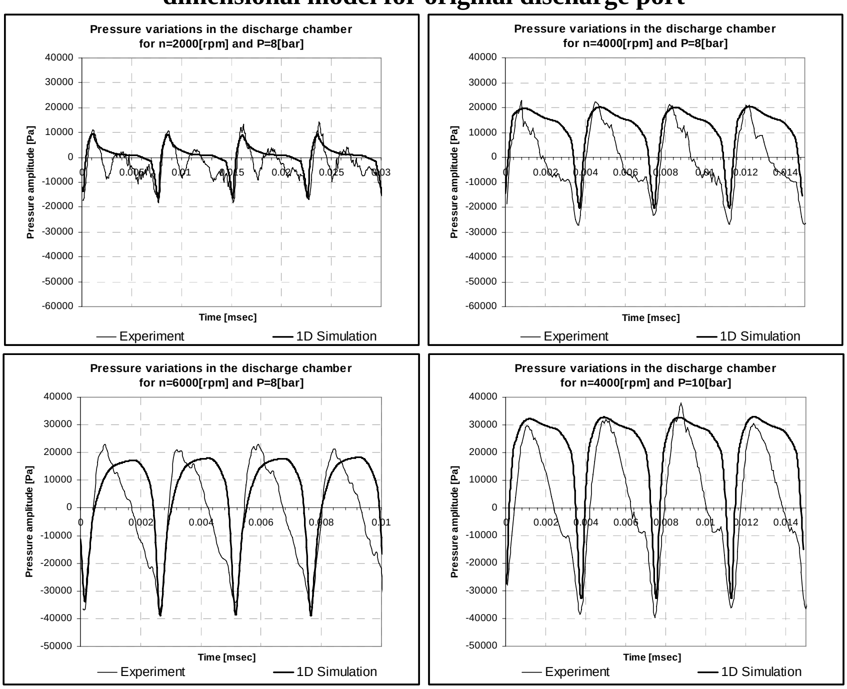

Figure 6 Comparison of experimental and results simulated by a one dimensional model for modified discharge port

The results presented in Figure 5 and 6 show that the two shapes of discharge port generate different amplitudes of gas pulsation in the discharge chamber, with smaller values from the modified shape.
It can be seen that the predicted results follow the experimental trends for all compressor speeds and discharge pressures. However, the higher harmonics are not well predicted because the complex geometry of the discharge chamber and compressor discharge system could not be accounted for by the simplified model. A full 3D CFD model which is able to account complex discharge chamber geometry is explained by *Kovacevic at all* [8]. This model has been employed for flow calculation in the discharge chamber and resulted in better agreement.

4. CONCLUSION

A simplified model was applied to calculate the flow in screw compressor discharge chambers in order to evaluate the influence of the discharge port size and shape upon gas pulsations. The results obtained with it were compared with test results on a compressor with two different port shapes. These agreed well and confirm that the shape and size of the discharge port determine level of gas pulsations in the discharge chamber. It follows that gas pulsations in screw compressor discharge port and consequently the generated noise can be reduced by appropriate size and shape of the discharge port.

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