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THE INFLUENCE OF THE DISCHARGE PORT SHAPE ON SCREW COMPRESSOR PERFORMANCE AND GAS PULSATIONS

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ABSTRACT

In a screw compressor, both pressure pulsations in the discharge chamber and performance are affected by the design of the discharge port. A function for calculating the real flow area of the discharge port has been included in two analytical models for evaluating this. The predicted results, thus obtained, showed that this function varies according to the shape of the discharge port in a manner that influences the amplitude of the gas pulsations and hence the noise generated in the discharge chamber. With the aid of the model, a port shape was designed to minimise these pulsations and thereby reduce noise levels by up to 5 decibels. However, reducing sound emissions by this means, slightly lowered the compressor efficiency. An experimental programme was then carried out to investigate this and good agreement was obtained between the predicted and measured values.

1. INTRODUCTION

In a screw compressor, the working chamber is connected to the suction and discharge chambers only periodically. This creates unsteady flow and variation of mass within both chambers. This creates pressure pulsations within them during both the suction and discharge processes, which create both vibration and noise. The amplitude of the gas pulsations in the compressor discharge chamber is much higher then that 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 with Fujiwara and Sakurai (1986). They measured gas pulsation, vibration, and noise in a screw compressor. After that Koai and Soedel (1990), developed an acoustic model in which they analysed flow pulsations in a twin screw compressor related them to its performance. More recently, Sangfors (1999) and Huagen et al (2004) developed mathematical models for the prediction of gas pulsations in screw compressor suction and discharge chambers.

All these authors explored the influence of various screw compressor design and operating parameters on gas pulsations in the compressor suction and discharge chambers. Some authors, such as Koai and Soedel (1990) recognized the influence of the compressor port area and recommended that the effect of the port shape on noise be further investigated. Mujić et al (2005) reported that changing the shape of a screw compressor discharge port alters gas pulsations in the discharge chamber, with reduced pulsation amplitude leading to lower overall noise levels. This led to the further investigation of this effect by Mujić et al (2007).

2. MATHEMATICAL MODELLING OF THE DISCHARGE FLOW PROCESS

Two mathematical models have been applied to analyse the screw compressor discharge process.

The first is a thermodynamic chamber model. The model provides averaged values of fluid properties in terms of pressure and temperature across a control volume, which can define the compressor suction, working or discharge chambers. The thermodynamic model consists of a set of differential equations for mass and energy conservation in the control volume. This system of differential equations is closed by a set of algebraic equations which describe internal leakage flow, flow associated with the suction or discharge processes, fluid injection and heat transfer. The relation between the fluid properties is derived from an equation of state, assuming either an ideal gas or a real fluid. The calculation time for their solution is minimised by the use of internal energy as the derived variable. Computation is then carried out through a series of iterative cycles until the solution converges. The pressure, together with any other required thermodynamic properties, is then derived directly from the internal energy calculation. This model provides reasonably accurate results which account for different discharge port shapes. The model described by Stosić and Hanjalić (1997) is fast, does not require extensive computer resources. More details of the mathematical modelling of the screw compressor discharge process, including the introduction of the true shape of the discharge port into the thermodynamic model are given by Mujić et al (2007).

The second is an integrated model which consists of a 3-D CFD model of the discharge chamber together with a thermodynamic model for the rest of the compressor fluid domains. This model accounts for a more complex discharge chamber geometry and therefore it is more accurate than the first one. However, due to the complexity of the 3-D CFD model, it requires longer calculation time than the chamber model. The integrated model is more accurate than the chamber model of Mujić et al (2007), and requires less computer resources and a shorter computational time than the full 3-D CFD model of a screw compressor described by Kovačević et al (2003). StarCCM+ commercial solver has been used for solving flow field quantities inside the discharge chamber. User subroutines, which enable transfer of mass and energy between 3-D CFD and thermodynamic chamber model, have been specially developed for this purpose and are included in the solver. This integrated model of the screw compressor discharge process is described in more detail by Kovačević et al (2007).

3. APPLICATION

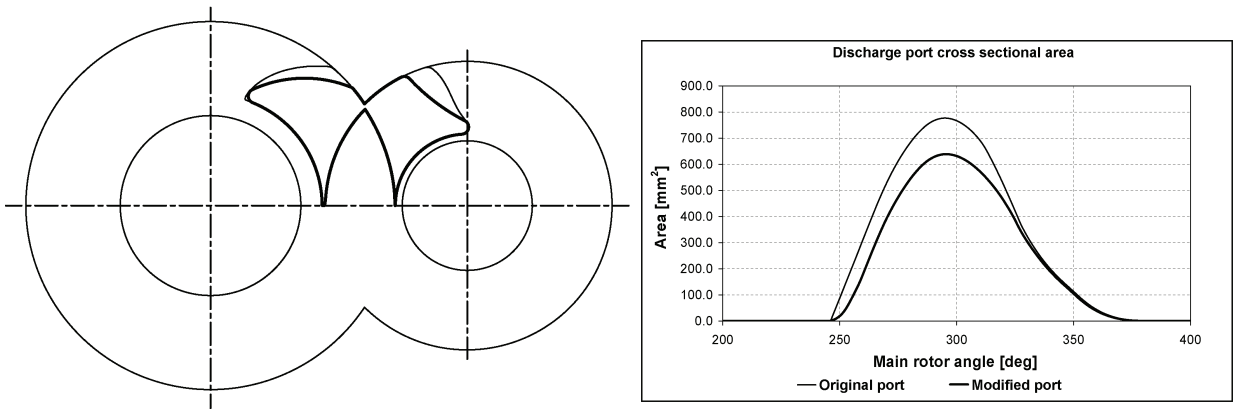
For the purpose of this investigation an oil flooded air screw compressor was used because it operates over a larger pressure range and develops higher gas pulsations than an oil free compressor. In addition an oil injected screw compressor does not need synchronizing gears, which affect the overall compressor noise level, and it may be driven at moderate speeds. Therefore the noise generated by the test compressor was mainly caused by the gas pulsations.

Both numerical and experimental results were obtained for two different port shapes in an industrial air compressor in order to evaluate their influence on the level of gas pulsations in the discharge port. Firstly, the pressure function in the compressor discharge chamber was calculated for the compressor with the original port by use of the integrated model. After experimental validation of the predicted results, a new discharge port shape was proposed. Due to its greater speed of calculation, the thermodynamic chamber model was used for establishing the new shape. Once an acceptable shape was obtained, the pressure function was predicted more accurately with the integrated model.

3.1. Modification of the discharge port and prediction of gas pulsations

The original discharge port was modified to reduce the gas pulsation amplitude by minimising the initial flow between the working and discharge chambers at the beginning of the discharge process, when the pressure difference between the chambers is the highest. As the built in volume ratio was not changed, the pressure difference at the start of discharge stays unchanged for both ports. To reduce flow losses, the original discharge port was designed to have the largest possible opening for any rotor position. This was achieved by aligning the port line to correspond with the rotor trailing edges. Such an approach produces an area function for the discharge port with a high starting gradient, as shown by the light line in Figure 1a. In that case the port has a large area when the pressure difference is greatest.

To avoid large initial flow, a new port shape was generated, as shown by the bold line in Figure 1a, where it can be seen clearly that the only change was in the curves that dictate the port opening. Thus both ports open at the same rotor position and only the starting gradient of the port area function is changed. Both ports therefore maintain the same internal volume ratio within the machines. The opening curves in this second case are generated by only three arcs. They are therefore simpler than the curves that define the old port. However, the modification of the port shape reduces the size of the port area. This difference increases from the very start of the discharge process and is greatest when the port is fully open. It then decreases as the port closes and finally disappears when the leading edges of the following rotor lobes cover the opening curves of the port completely. This reduction in the port area increases the flow loses.



a) **b)**

Figure 1 Shape and cross sectional area of two different discharge ports

A numerical grid of the discharge chamber was provided for the integrated model from a CAD model of the original shape. This was obtained directly from the CAD model by use of the STARCCM+ meshing features. The existing CAD model was then altered to generate numerical mesh of the discharge chamber with shape of the new port. The number of cells of the new fluid domain is almost the same as in the old. The numerical grid of the discharge chambers with two port versions is shown in Figure 2.

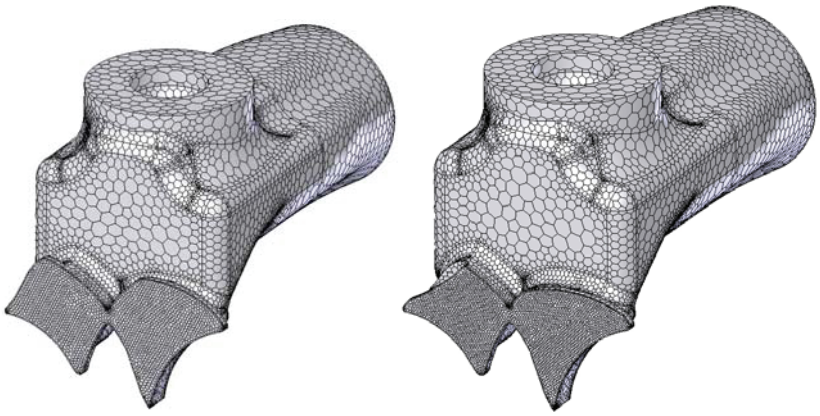
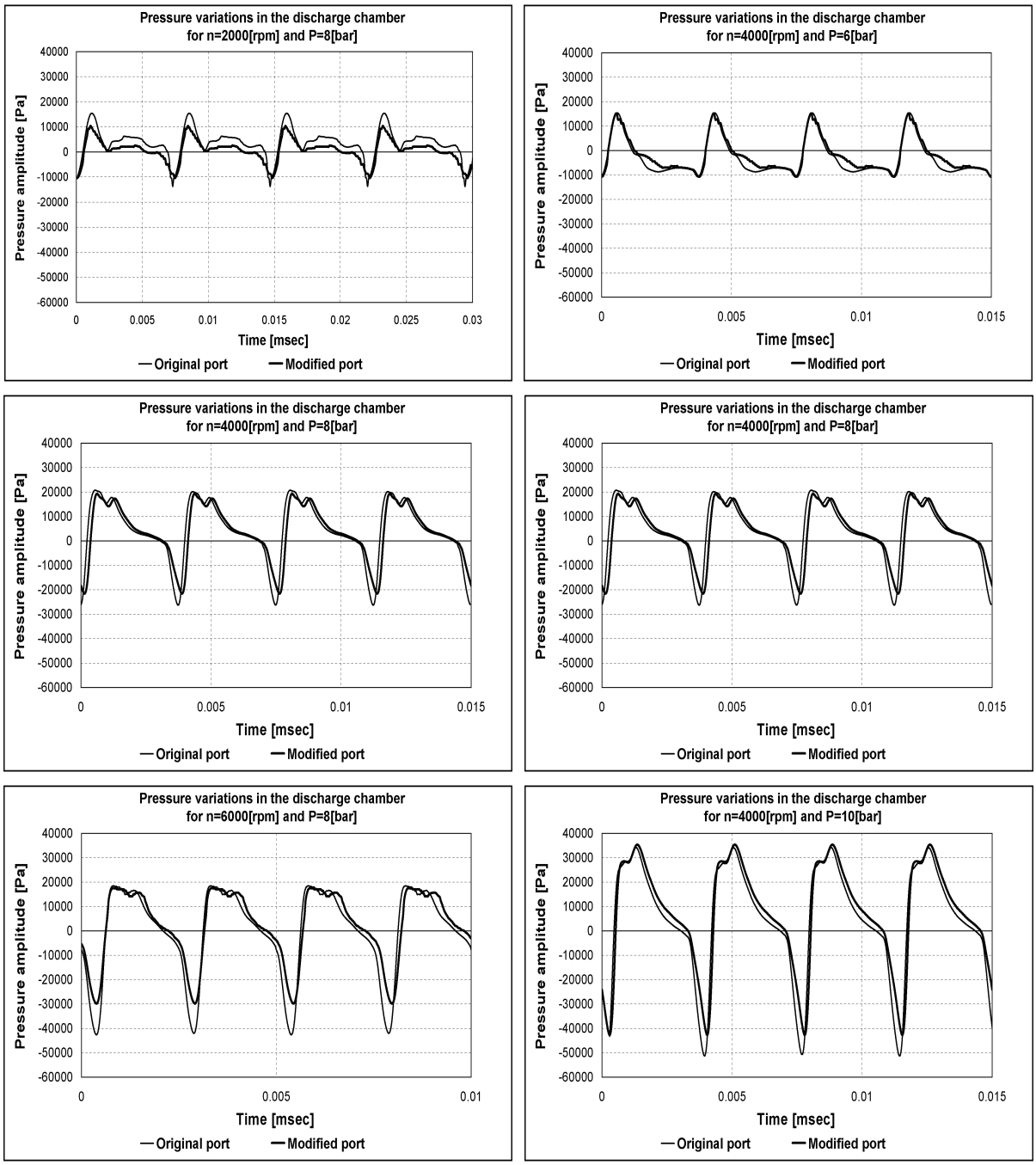


Figure 2 Numerical mesh of the discharge chamber with original and modified ports

In both cases, the numerical grid consists of about 80 000 cells. The time step length corresponds to the time required for rotation of a male rotor by an angle of 1º and is dependent of the compressor speed. Convergence was achieved after four discharge cycles. The calculation time was about 8 hours, using a computer with a Pentium Dual Core processor with a working frequency 1.8 GHz and 2 GB of RAM. Comparison of results obtained with the original and modified ports is presented in Figure 3. The predicted results are shown by a bold line for the modified

port and a light line for the original port. They show a lower level of gas pulsations with the new port for all calculated cases. As expected, the difference between the old and new port pressure pulsation amplitude increases at higher speeds and pressure differences. Thus it was confirmed analytically that it is possible to reduce gas pulsations by changing the discharge port shape



a) Outlet pressure 8 [bar] b) Compressor speed 400[rpm]

Figure 3 Comparison of results predicted by the integrated model for original and modified discharge ports

3.2. Experimental investigation of pressure pulsations

An ELGI E102 compressor was used for experimental investigation of the effects shown by the analysis. This was installed in an air compressor test rig at City University, which conforms with CAGI and PNEUROP standards. Two sets of measurements were taken. One covered compressor speeds from 2000 to 6000 [rpm] at a discharge pressure of 8 [bar] while the other covered discharge pressures from 5 to 12 [bar] at a speed of 4000 [rpm].

The discharge chamber pressure was measured by use of an Endevco model 8530C piezo-electric pressure sensor to obtain a pressure function. The sound pressure level (SPL) around the test compressor was measured by sound pressure level indicators. An SJK HML 323 Sound Level Meter was used to obtain a correlation between the pressure function within the discharge chamber and the overall compressor noise. Measurements of the pressure, temperature, driving torque, air flow and compressor speed were also taken.

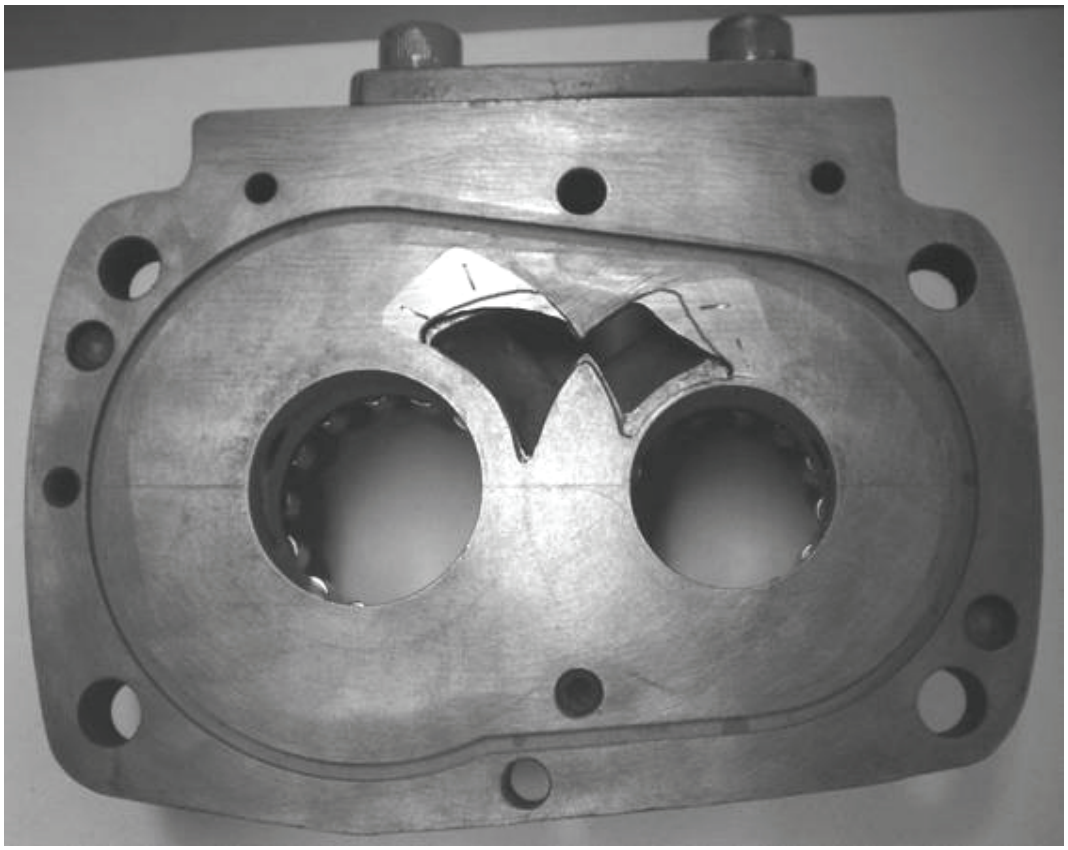
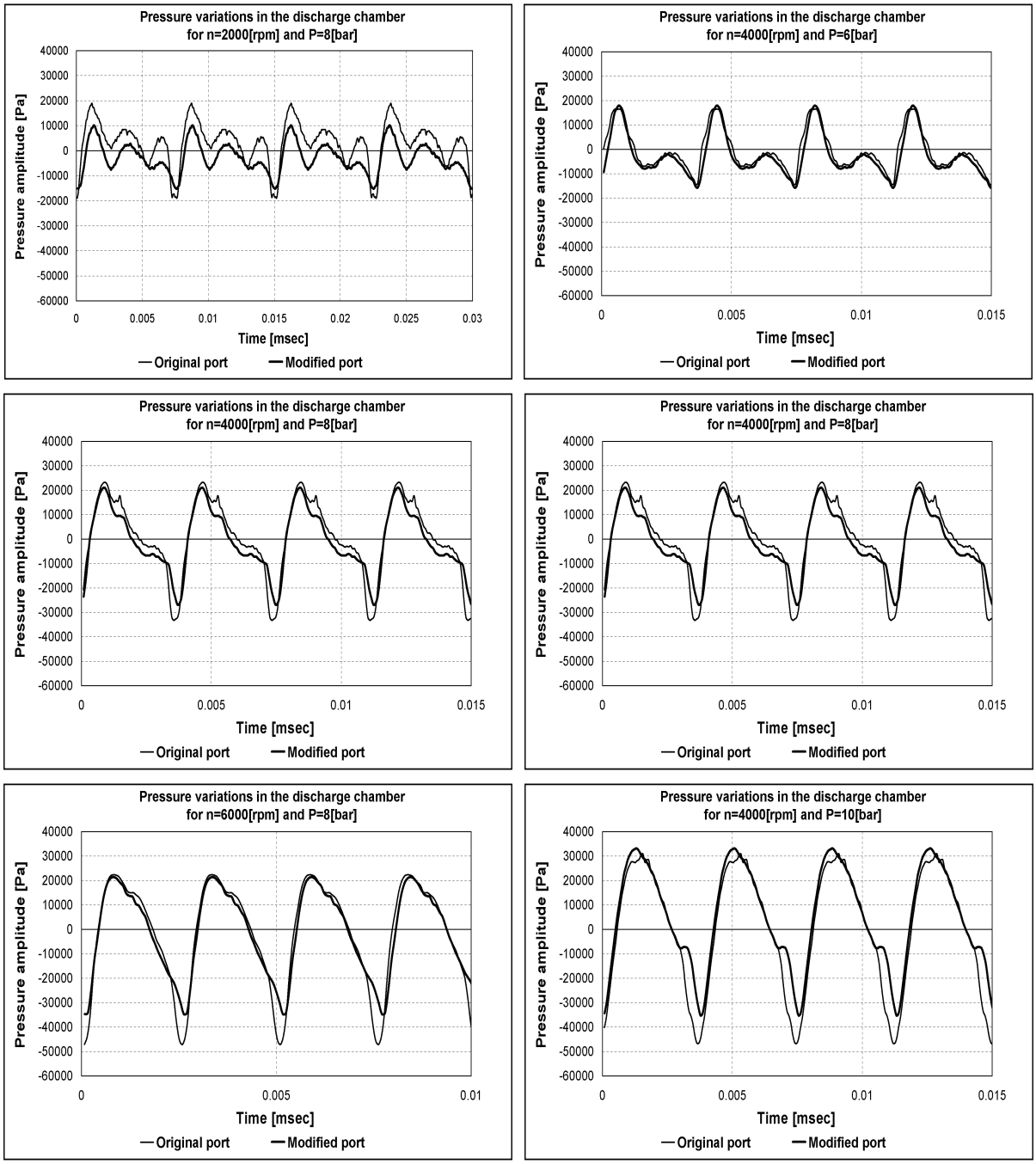


Figure 4 Modified shape of the discharge port

To validate the predicted results, experiments on both the original and modified ports were carried out. The shapes of the discharge port in the discharge housing of the compressor are shown in Figure 4. The original port was removed by machining two shallow slots. Brass inserts were used to fill the machined slots and to create the new port shape.

The measured values of the pressure functions in the discharge chamber are compared in Figure 5. These are shown for the original and modified port by light and bold lines respectively. It can be seen that the gas pulsation amplitudes are reduced across the whole range of the working conditions.

The Sound Pressure Level (SPL) is calculated using the root means square (RMS) amplitude obtained from experimental results. It is shown in Figure 6. The light line presents results for the original shape of the port, while the bold line presents results obtained after the port was modified. It can be seen that the amplitude of the gas pulsations is reduced across the whole range of the working conditions.



a) Outlet pressure 8 [bar] b) Compressor speed 400[rpm]

Figure 5 Comparison of experimental data for original and modified discharge ports

Figure 6 shows the reduction of SPL for different outlet pressures. This reduction varies from 1[dB] when the outlet pressure is low to almost 5[dB] for higher outlet pressures. The lowest level of gas pulsations and lowest reduction in SPL is expected to correspond to the outlet pressure being equal to the pressure at the end of the compression. The pressure at the end of the compression for this size of the discharge ports is around 6 [bar]. So for the outlet pressures close to built in pressure the gas pulsation amplitudes are lowest and due to that reduction in SPL is also lowest. By increasing the outlet pressure the pressure difference also increases which generates higher gas

pulsations. In such conditions the affect of the modified port is higher achieving higher reduction in gas pulsations and the SPL, as shown in Figure 5 and Figure 6.

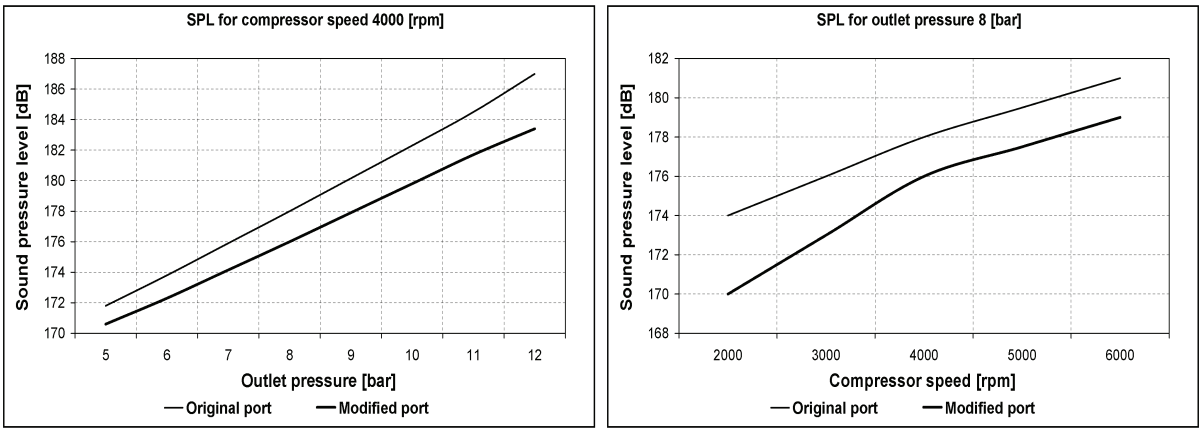


Figure 6 Comparison of calculated SPL inside the discharge chamber for original and modified discharge ports

The achieved reduction in SPL for different compressor speeds is also presented in Figure 6. Generally, increase in speed increases the amplitude of gas pulsations achieving higher reduction at higher compressor speeds. However, as it is shown in Figure 6, the highest reduction in SPL of 4 [dB] has been achieved at the lowest compressor speed of 2000 [rpm]. This is due to the large influence of reflected waves on the pressure function for such compressor working conditions as is shown in the first diagram in Figure 5. The modified port also reduced the amplitude of the reflected waves, resulting in a higher reduction of SPL than for compressor speeds where these are not significant.

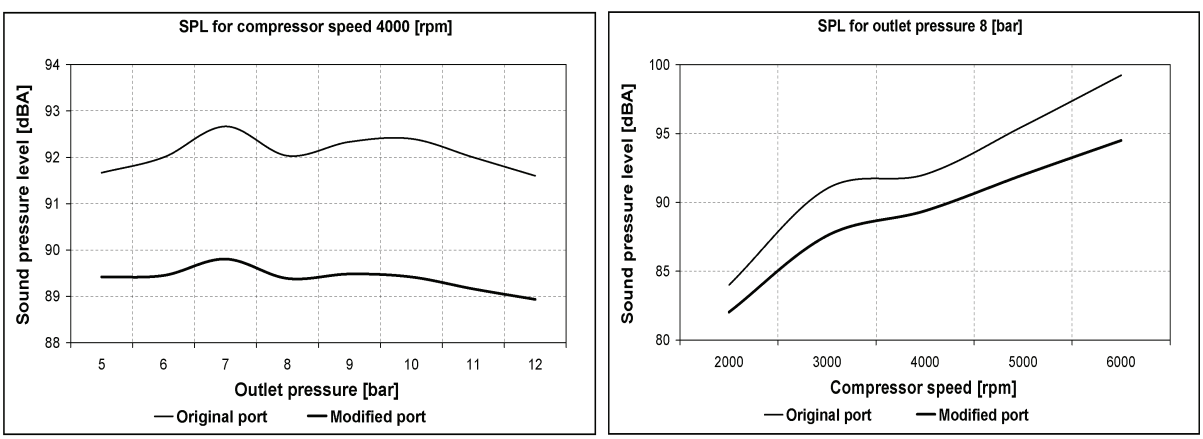


Figure 7 Comparison of measured SPL around compressor for original and modified discharge ports

The SPL measured around the compressor is shown in Figure 7. This does not correspond with the SPL diagrams of Figure 6. The sound level meter used for these measurements registered the highest SPL. Therefore, the results presented in Figure 7 are for different frequencies and are influenced by other sources of noise in the compressor system. However, even in this case, modification of the discharge port shows a noticeable reduction of noise across the entire measuring range. It can be seen that the overall noise in the compressor environment is attenuated by about 3dB at 4000rpm over the whole pressure range, while there is a noticeable noise reduction across the compressor speed range, varying from 2 [dB] at the lowest speed up to 5 [dB] at the maximum speed for an outlet pressure of 8bar. The use of the model therefore represents a good step towards overall noise reduction.

The modified port requires more power input to the compressor for the same working conditions than the original port, which was anticipated because of the smaller flow area. The specific power for the two versions is presented in

Figure 8. In the case of the modified port, the losses increase with the compressor speed when a higher fluid flow through the port is present. At higher outlet pressures it is expected that the modified port will reduce back flow which will potentially improve the compressor performance. However, this has not been captured by the experiment.

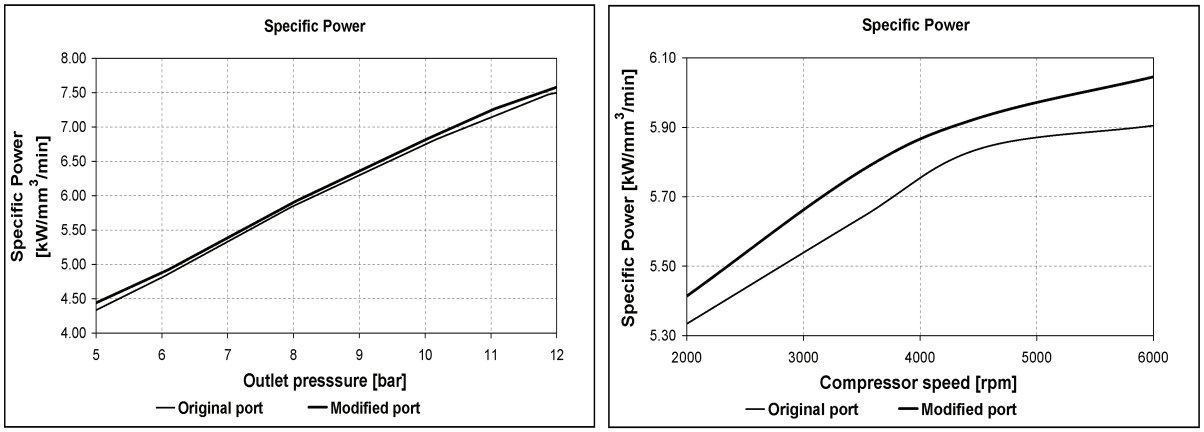


Figure 8 Compressor performance for original and modified discharge port

4. CONCLUSION

Both the integrated model and the test results gave good agreement over a wide range of speeds and pressures in a compressor with two different port shapes and have shown that gas pulsations in a screw compressor discharge port and, consequently, the generated noise can be reduced by making appropriate changes to the size and shape of the discharge port.

The given example confirms that, by this means, it is possible to reduce level of gas pulsations in the discharge chamber by up to 5 [dB]. Unfortunately, this reduction is accompanied by a drop in compressor performance of up to 2%. Therefore, this method of reducing noise is most useful for compressor applications where lower noise levels take precedence over compressor power consumption.

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