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Numerical Investigation of Parameters Affecting Discharge Line Pressure Pulsations in Hermetic Reciprocating Compressors

Numerical Investigation of Parameters Affecting Discharge Line Pressure Pulsations in Hermetic Reciprocating Compressors

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Abstract. In hermetic reciprocating compressors, it is possible to improve the sound quality of the compressor as well as to minimize the discharge line flow losses of the compressor by optimization of the pressure pulsations in the discharge line. Pressure pulsation induced vibrations significantly affect both the sound quality and the vibration level of the entire cooling system. This study is prepared to minimize the pressure pulsations in the discharge line of the compressor by comparing and determining the effects of alternative designs on pressure pulsations with the numerical results of base and alternative designs formed by differentiation of crankcase orifice diameter and discharge muffler channel diameter from the discharge line design parameters. In this paper, the design parameters that affect the discharge line pressure pulsations and flow losses were investigated with a transient CFD analysis methodology, which was developed and confirmed by experimental measurements in previous study. At the end of the study, the geometric characteristics of the different design parameters of the discharge line such as orifice diameters are presented with transient numerical analysis.

1. Introduction

Compressor is the main component that determines many performance criteria, especially efficiency and noise level of the refrigeration system. Efficiency, acoustic noise level, compactness, service lifetime and price are the five fundamental requirements expected from hermetic reciprocating compressors design [1]. Pressure calculation in the design phase of a hermetic reciprocating compressor, is important since pressure pulsation affects the performance and noise level of a compressor. Unsteady flow causes pressure pulsation due to the reciprocating action of the piston and by self-acting valves that open and close depending on the pressure difference [2]. When the pressure pulsations in the flow lines of the household type hermetic reciprocating compressors are examined, it has been observed that they affect valve behaviours, structural parts and performance parameters. It is stated that if the amplitudes of the pressure oscillations are greater than the pressure drop in the valve, reverse flows will occur. In such cases, the impact speed of the valve on the seat increased and the valve was observed to perform more fluttering [3]. Jacobs [4] investigated the relationship between the movement of the suction valve and the pressure pulsations in the suction manifold. In his study, pressure pulsation in compressor flow lines was described as one of the loss item for compressor

performance. It was indicated that pressure pulsations were affect the movement of valve and forced to reopen after closing the suction valve. Therefore, it has been determined that there was a loss of 23% in compressor capacity and 12% in efficiency. On the other hand, high amplitude vibrations that may occur in the discharge line cause resonance formation in the pipe system and cause breakage, which affects the reliability of the household type hermetic compressors [5]. These vibrations in the discharge line are transmitted to both the compressor housing and other parts of the cooling system, such as the condenser, causing vibration and vibration-induced noise generation [6].

Within the scope of this study, pressure pulsations at discharge line of a hermetic reciprocating compressor were investigated numerically. In this paper, discharge line orifice diameters and channel diameter between the discharge mufflers, which are design parameters, that affect the discharge line pressure pulsations and flow losses were investigated with a transient CFD analysis methodology, which was developed and confirmed by experimental measurements in previous study [7]. Numerical analysis of the discharge line has been carried out with a commercial CFD software package. In the analysis, piston motion and the compression work are modelled as moving mesh structure.

In the study, dynamic mesh modelling is used instead of FSI which is an expensive solution method in terms of time and resources. The discharge valve motion as fully opened to limiter and closed is automatically carried out based on the pressure difference in the cylinder and discharge manifold. For the base and alternative design parameters pressure pulsations in the discharge plenum, which is the starting point of the discharge line, were analyzed at different operating speeds of the variable speed compressor under the ASHRAE LPB pressure condition. At the end of the study, alternative designs were compared with the base design.

2. Numerical Analysis

The developed numerical model allows both the numerical modeling of the pressure pulsations and the visualization of the flow, allowing for the creation of the most appropriate flow line design. Numerical analyses were carried out with the 3D discharge line geometry. The compressor used in the study has 145 kcal/h capacity at ASHRAE conditions. The inspected flow volume in analysis, is delimited between the cylinder bore where the fluid is pressurized and the compressor discharge pipe. In the numerical analysis unsteady and incompressible turbulent flow models have been examined. For this purpose, three-dimensional mass, momentum and energy conservation equations and turbulence model and ideal gas equations have been numerically solved. Analyses were performed at 2100, 3000 and 4500 rpm and used necessary boundary conditions and thermophysical properties.

At discretization stage, tetrahedral and hex mesh is generated for discharge line. The smallest and the biggest element sizes are $5.5 \cdot 10^{-5}$ m and $1 \cdot 10^{-3}$ m respectively. The total number of elements is approximately 800,000 that is enough for mesh independency. Both the 3D geometry and mesh geometry of cylinder bore and discharge line are shown in Figure 1.

Opening and closing of the discharge valve is modeled by converting the flow line to wall and fluid interior boundary conditions. These changes in boundary conditions have been automated from dynamic mesh events depending on the pressure difference between the cylinder bore and the discharge plenum. The valve flow area is determined by the maximum clearance that is allowed by the discharge valve limiter.

In the analysis, isobutane (R600a) is used as a refrigerant and the fluid properties other than the density have been implemented as constant at mean temperature of operating speed condition. Density is obtained from the ideal gas equation depending on the temperature. The refrigerant is assumed to be pure without oil. Table 1 shows the refrigerant properties for a single operating condition.
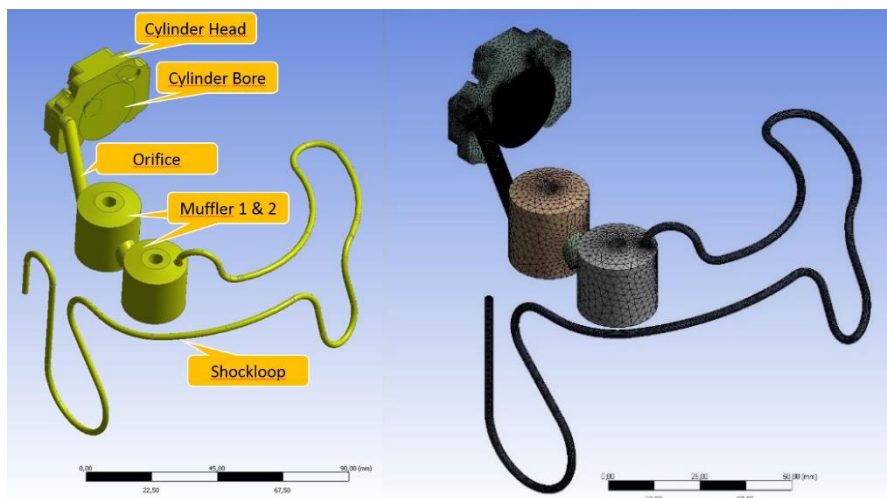
Figure 1. 3D mesh geometry of discharge line.

Table 1. The refrigerant properties.

| Properties	Isobutane (R600a)
Density (kg/m3)	Ideal Gas
Specific Heat Cp (j/kg-K)	2151.4
Thermal Conductivity (W/mK)	0.02586
Dynamic Viscosity (kg/ms)	9.43017e-6
Molecular Weight (kg/kgmol)	58.1222

For the boundary conditions of discharge tube pressure outlet condition was set. Due to the suction flow line is neglected in the simulation, there is no fluid inlet section in the numerical model. An initial cylinder pressure that is obtained from ASHRAE LBP inlet pressure conditions was set at the end of the suction phase of each cycle. Wall temperatures of cylinder and other discharge line areas are assumed to be constant and are the temperature values obtained by experimental measurements in steady state condition. The defined boundary conditions are summarized in Table 2.

Table 2. Boundary conditions.

Zone	Boundary Condition
Discharge Valve	Wall (Closed) / Interior (Fully Opened)
Piston	Mowing Wall
Shockloop Out	Pressure Outlet @ASHRAE
Cylinder	Wall
Discharge Line	Wall

Between the time steps, 1° crank angle is used in analysis. For the compressor model examined, the average fluid velocities calculated in the operating conditions of ASHRAE at different operating speeds and the Realizable k-Epsilon turbulence model was used according to the Re values. The turbulence parameters, such as turbulence intensity, were calculated and specified at appropriate zones. SIMPLE algorithm was used for the coupling of pressure-velocity equation pairs in analyzes using pressure-based solver.

The effect of the discharge line design parameters on the pressure pulsations is the channel diameter between the exhaust mufflers and the orifice diameter. The positions of these parameters are shown in Figure 2 and the values are given in Table 3.
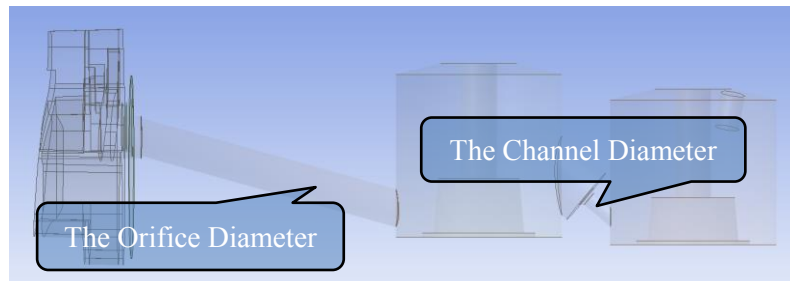

Figure 2. Design parameters changed in alternative geometries.

Table 3. Value of design parameters.

| Design Parameter	Base	A	B	C	D
The Orifice Diameter	X	**X-0,8**	**X+0,8**	X	X
The Channel Diameter	Y	Y	Y	**Y/2**	**2Y**

3. Results

The numerical analysis results of the effect of the alternative orifice diameters (Design-A and Design-B) on the discharge plenum pressure pulsations are given in Figures 3, 4 and 5, for the operating speeds of 2100, 3000 and 4500 rpm respectively. The Design-A of the smaller orifice diameter compared to the base design has been shown to increase the amplitude of pressure pulsation in the discharge plenum at all operating speeds, while the bigger orifice diameter Design-B reduces the amplitude at 2100 and 4500 rpm. It was seen that the first peak with decisive effect on amplitude decreased with the increase of the orifice diameter.
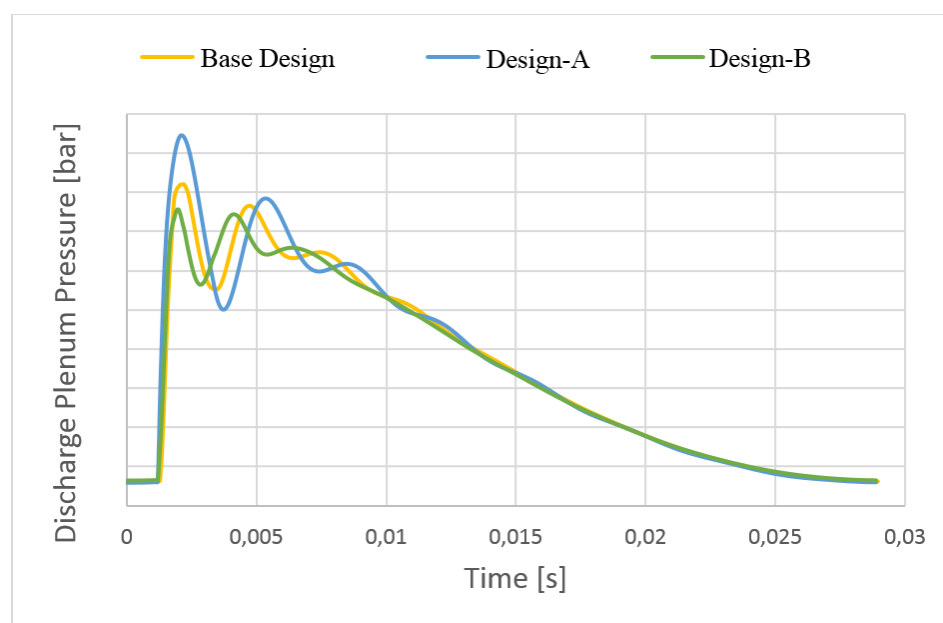
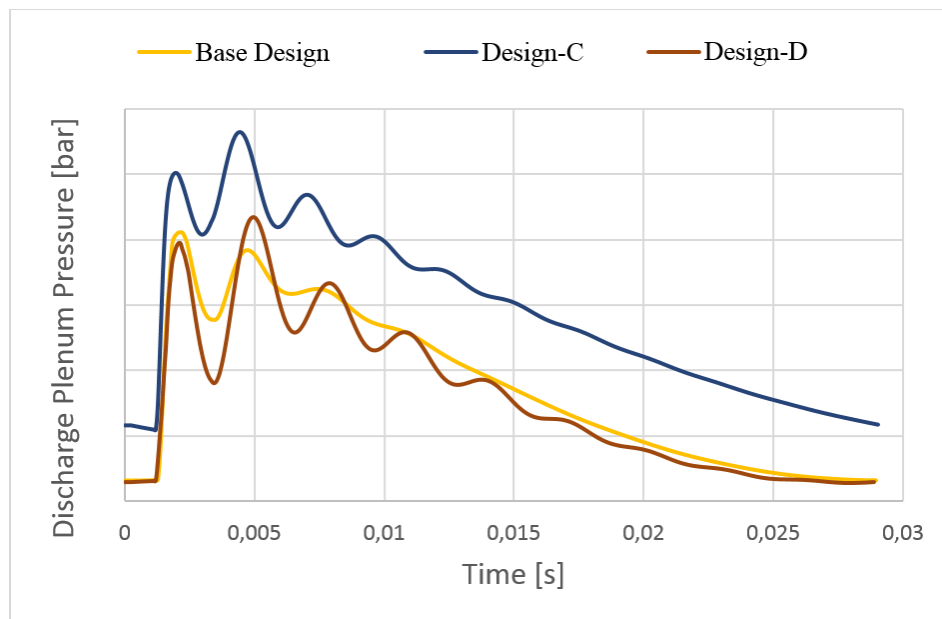**Figure 3.** The effect of orifice diameter on pressure pulsation (2100 rpm)
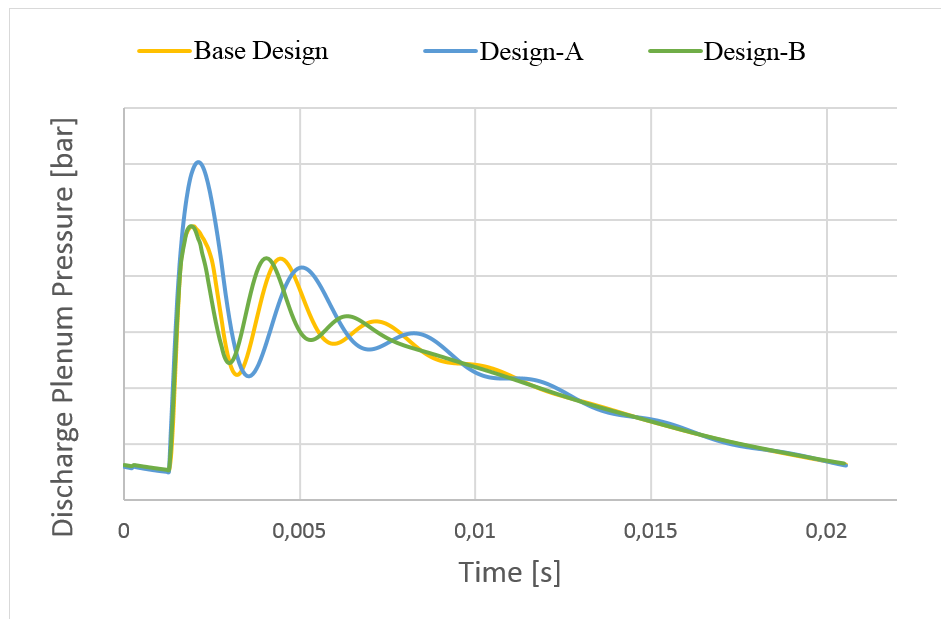
Figure 4. The effect of orifice diameter on pressure pulsation (3000 rpm)
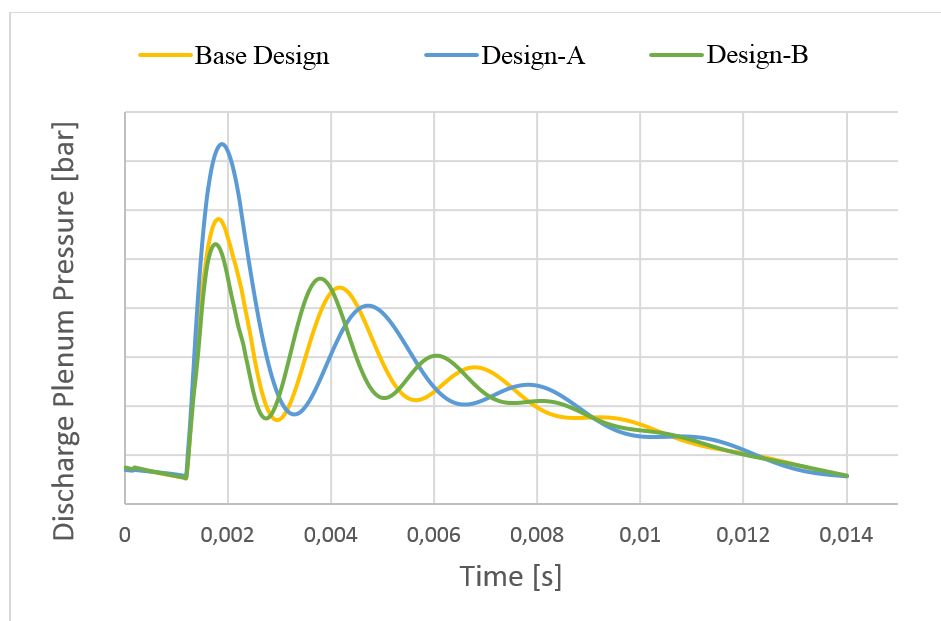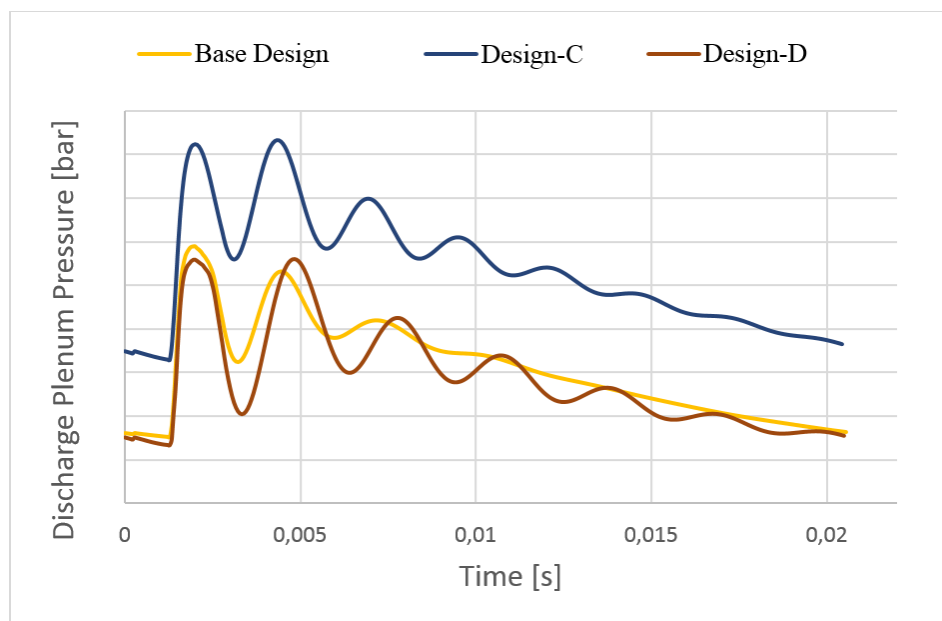

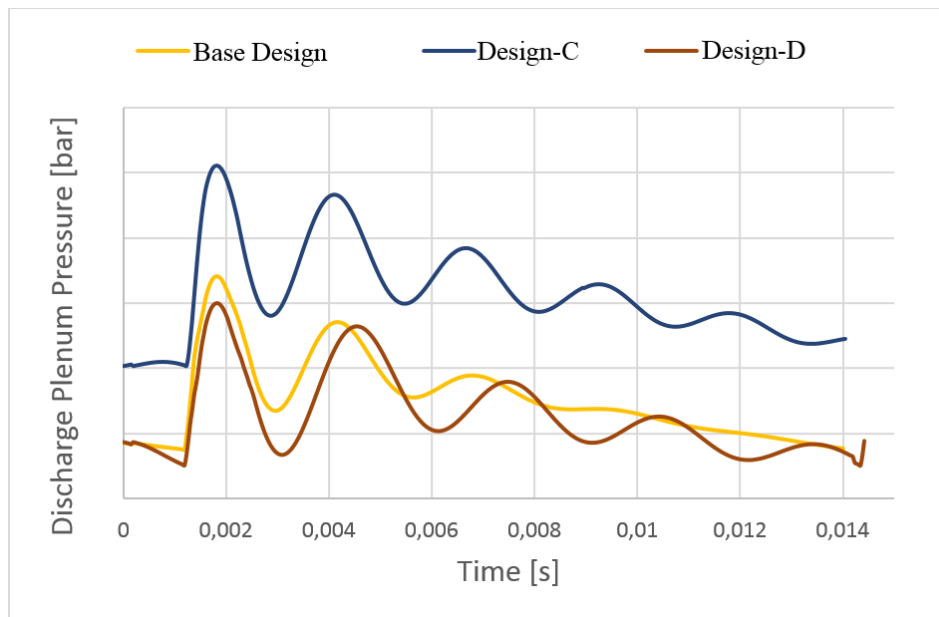**Figure 5.** The effect of orifice diameter on pressure pulsation (4500 rpm)

The results of the numerical analysis, which examines the effect of the channel diameter between the two discharge mufflers on the discharge plenum pressure pulsations, are given in Figures 6, 7 and 8, for the operating speeds of 2100, 3000 and 4500 rpm respectively The Design-C, which has a smaller channel diameter than the basic design, has been found to increase the average cycle pressure with increasing cycle numbers at all operating speeds. Since the channel diameter between the mufflers is low, it is seen that the flow cannot be evacuated from the discharge line in the increasing number of cycles and the exhaust plenum pressures before the channel are gradually increased. According to the basic design, the number of oscillations per cycle has increased in both the C and D designs.

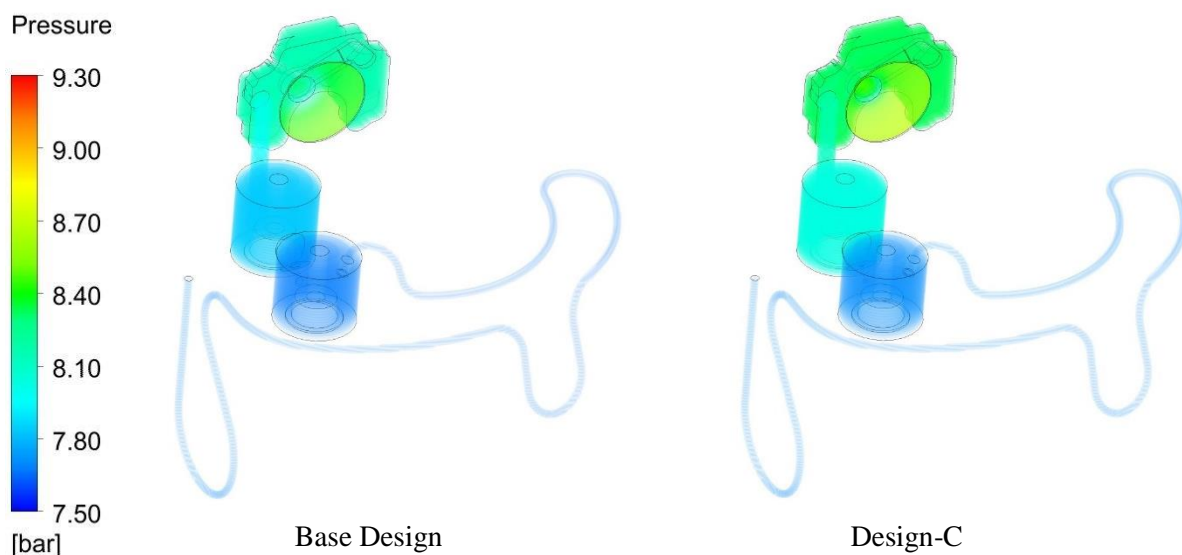**Figure 6.** The effect of channel diameter on pressure pulsation (2100 rpm)

Figure 7. The effect of channel diameter on pressure pulsation (3000 rpm)

6

Figure 8. The effect of channel diameter on pressure pulsation (4500 rpm)

The plenum pressure before the channel, that is between the discharge mufflers, is higher than the basic design is shown in Figure 9.

Figure 9. Discharge line pressure distribution (3000 rpm)

When the effect of the operating speed was examined, it was seen that the pressure pulsation amplitude increased with the decrease of the working speed. Accordingly, it is observed that the decreasing time of the pressure decreases with the increase of the working speed.

4. Conclusion

This paper presented a three-dimensional CFD analysis to investigate the effects of both orifice diameter and channel diameter between the discharge mufflers on pressure pulsations at discharge plenum of a variable speed hermetic reciprocating compressor. Within the scope of the study, the effects of alternative designs were investigated with the analysis model developed in the previous study [7].

According to the basic design, pressure pulsation amplitude at discharge plenum decreases with the increase of the orifice diameter. On the other hand, changing the diameter of the channel between the discharge mufflers does not have any advantage over the pressure pulsation amplitude. Based on the results of this study it will be extended that:

- The effect of alternative designs in the discharge flow line on pressure pulsations will be examined experimentally and compared with the numerical results.
- The movement of the discharge valve will be examined experimentally and compared with the opening and closing times in the analyses.

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