

Pressure Measurement and CFD Analysis on the Snubber of Reciprocating Hydrogen Compressor

Gyeong-Hwan Lee¹, Wanda Ali Akbar¹, Hyo-Min Jeong²
and Han-Shik Chung²

¹Graduate School, Department of Mechanical and Precision Engineering,
Gyeongsang National University,
445 Inpyeong-dong, Gyeongsang-namdo 650-160, Korea

²School of Mechanical and Aerospace Engineering, the Institute of Marine
Industry,
Gyeongsang National University, 445 Inpyeong-dong, Gyeongsang-namdo 650-
160, Korea
saba9543@naver.com

ABSTRACT: The study about hydrogen fuel which consist of hydrogen extraction and reforming process, fuel cell equipment, and receptacle are flourishing all over the world. Currently, Korean hydrogen station is still underdeveloping. And also the most important part which is hydrogen compressor has not been developed. Snubber is one of the most important parts in hydrogen compressing system. It has installed reciprocating hydrogen compressor. One of these components is snubber which has function to reduce pulsation waveform and to remove the impurities in the hydrogen gas. Therefore, a snubber has a buffer. It is an inclined plate, attached inside snubber. The best snubber is the one which has the least pressure loss and the most pressure amplitude reduction. Furthermore, investigating geometric optimum size of with-buffer snubber has been done by numerical analysis approach.

Keywords: Hydrogen Gas, Reciprocating Compressor, Pulsation Pressure, Fluctuation, Snubber, Buffer

NOMENCLATURE

P_{TL}	total pressure loss [Pa=N/m ²]
P_F	pressure pulsation [Pa]
P_{mean}	mean pressure value [Pa]

1. INTRODUCTION

The use of hydrogen as a transport fuel has been investigated for a few decades, but in the past 10 years the number of research and

pilot projects has escalated. The recent hikes in the price of oil have also added impetus to the movement towards hydrogen and other alternatives fuels [1]. A methodology for the integrated production planning and reactive scheduling in the optimization of a hydrogen supply network was proposed by SA. Heever et al in [2]. In this network's model, compressing system takes an important part of whole system. Generally,

hydrogen compressor is reciprocating two-stage type. The schematic drawing of the real Hydrogen compressing system is shown by Figure 1. From that figure, the snubbers applied to each compressor stage can be seen clearly. Here, four units of snubber are used for each inlet and outlet for the first and second stage.

Technically reciprocating compressor type has higher pressure ratio than the rotating one. Therefore this type is used in hydrogen handling both for storing and transferring. Special character of pressure produced by this compressor is pulsation or fluctuation. This phenomenon has a lot of disadvantages not only for the gas itself but also for equipments relating to the system. For this occasion the snubber was designed and used [3]. In order to damp the pressure fluctuation a flat plate is inserted inside the snubber. This plate is called buffer. The installation of a buffer inside a snubber can be seen in the cutting view in Figure 2. An experiment to find out the effect of buffer presence in a snubber had already been conducted [4]. This resulted a fact that snubber with buffer has better performance. As addition, the design of pulsation and vibration control for hydrogen compressor system should consider requirements in the API Standard 618 book [5].

For several parameters in low pressure range, hydrogen gas has same character with the atmosphere air. Especially to observe the pressure from physical approach (without considering the chemical character), pressured air can be used to represent hydrogen gas. Modeling hydrogen compressor by using air compressor unit brought a good agreement with the CFD simulation. After that, the study was focus on the numerical analysis to achieve the optimum dimension for snubber. This paper explains about the effect of varying buffer angle to the pressure characteristics. Three models group, distinguished with the volume, are compared..

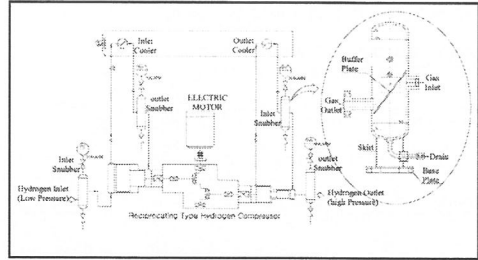


Figure 1: Schematic diagram of a Hydrogen gas Compressing System.

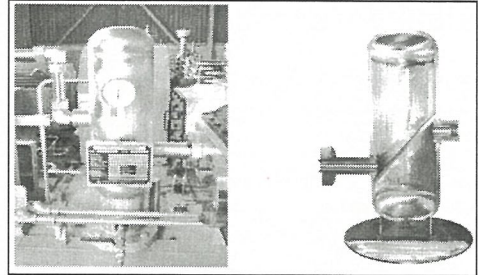


Figure 2: Detailed and Cutting View of a Snubber

2. TEST SETUP AND METHOD

2.1 MATHEMATICAL FORMULATION

The purpose of deriving mathematical model in CFD is to find the appropriate relation between each fluid flow properties. The three dimensional, compressible, turbulent flow of the gas in the test problems is governed by partial differential equations that express the principles of conservation of mass, momentum and energy. In this paper, the turbulent model chosen was $k-\epsilon$ with high Reynolds number. All of these equations below were taken from Star CD's, the CFD code used, Methodology book [6].

The continuity equation

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho U)}{\partial x} + \frac{\partial(\rho V)}{\partial y} + \frac{\partial(\rho W)}{\partial z} = 0$$

x-direction momentum equation

$$\frac{\partial(\rho U)}{\partial t} + \frac{\partial(\rho U^2)}{\partial x} + \frac{\partial(\rho UV)}{\partial y} + \frac{\partial(\rho UW)}{\partial z} - \frac{\partial}{\partial x} \left(\mu_{eff} \frac{\partial U}{\partial x} \right) - \frac{\partial}{\partial y} \left(\mu_{eff} \frac{\partial U}{\partial y} \right) - \frac{\partial}{\partial z} \left(\mu_{eff} \frac{\partial U}{\partial z} \right) = -\frac{\partial P}{\partial x} + S^U$$

y-direction momentum equation

$$\frac{\partial(\rho V)}{\partial t} + \frac{\partial(\rho UV)}{\partial x} + \frac{\partial(\rho V^2)}{\partial y} + \frac{\partial(\rho V'W)}{\partial z} - \frac{\partial}{\partial x} \left(\mu_{eff} \frac{\partial V}{\partial x} \right) - \frac{\partial}{\partial y} \left(\mu_{eff} \frac{\partial V}{\partial y} \right) - \frac{\partial}{\partial z} \left(\mu_{eff} \frac{\partial V}{\partial z} \right) = -\frac{\partial P}{\partial y} + S^V$$

z-direction momentum equation

$$\frac{\partial(\rho W)}{\partial t} + \frac{\partial(\rho UW)}{\partial x} + \frac{\partial(\rho VW)}{\partial y} + \frac{\partial(\rho W^2)}{\partial z} - \frac{\partial}{\partial x} \left(\mu_{eff} \frac{\partial W}{\partial x} \right) - \frac{\partial}{\partial y} \left(\mu_{eff} \frac{\partial W}{\partial y} \right) - \frac{\partial}{\partial z} \left(\mu_{eff} \frac{\partial W}{\partial z} \right) = -\frac{\partial P}{\partial z} + S^W$$

Turbulence kinetic energy equation

$$\frac{\partial(\rho k)}{\partial t} + \frac{\partial(\rho U k)}{\partial x} + \frac{\partial(\rho V k)}{\partial y} + \frac{\partial(\rho W k)}{\partial z} - \frac{\partial}{\partial x} \left(\left(\frac{\mu_t}{\sigma_k} + \mu \right) \frac{\partial k}{\partial x} \right) - \frac{\partial}{\partial y} \left(\left(\frac{\mu_t}{\sigma_k} + \mu \right) \frac{\partial k}{\partial y} \right) - \frac{\partial}{\partial z} \left(\left(\frac{\mu_t}{\sigma_k} + \mu \right) \frac{\partial k}{\partial z} \right) = G_k - \rho \epsilon$$

turbulence dissipation rate equation

$$\frac{\partial(\rho \epsilon)}{\partial t} + \frac{\partial(\rho U \epsilon)}{\partial x} + \frac{\partial(\rho V \epsilon)}{\partial y} + \frac{\partial(\rho W \epsilon)}{\partial z} - \frac{\partial}{\partial x} \left(\left(\frac{\mu_t}{\sigma_\epsilon} + \mu \right) \frac{\partial \epsilon}{\partial x} \right) - \frac{\partial}{\partial y} \left(\left(\frac{\mu_t}{\sigma_\epsilon} + \mu \right) \frac{\partial \epsilon}{\partial y} \right) - \frac{\partial}{\partial z} \left(\left(\frac{\mu_t}{\sigma_\epsilon} + \mu \right) \frac{\partial \epsilon}{\partial z} \right) = C_{1\epsilon} \frac{\epsilon}{k} G_k - C_{2\epsilon} \rho \frac{\epsilon^2}{k}$$

$$G_k = \mu_t \cdot \left[\left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \right)^2 + \left(\frac{\partial v}{\partial z} + \frac{\partial w}{\partial y} \right)^2 + \left(\frac{\partial w}{\partial x} + \frac{\partial u}{\partial z} \right)^2 \right]$$

Pressure Characteristics Analysis

The main functions of the snubber are to reduce the pulsation and (in the same time) to keep the pressure magnitude [4]. For that reason, three following equations are derived.

$$\Delta P = P_{in} - P_{out}, \quad P_{TL} = \frac{\Delta P}{P_{in}} \times 100\% \quad (1)$$

$$\text{And } P_F = \sum \frac{\Delta P}{2 \times P_{mean}} \times 100\% \quad (2)$$

In these equations, ΔP , P_{in} , P_{out} , P_{TL} , P_F and P_{mean} are pressure difference, inlet pressure, outlet pressure, total pressure loss, pressure pulsation and mean pressure value, respectively. Reducing the pulsation means reducing P_F value, while keep the pressure

magnitude means keep P_{TL} value as low as possible.

2.2 COMPUTATIONAL METHODOLOGY

The computational code used was Star CD (Version 3.24), which solve the full 3D time dependent Navier-Stokes, continuity and energy equations using the finite volume method. This commercial code is widely used in the numerical simulation of different flow conditions in various complex geometries and was chosen in this study because it is proven capability and validity. The turbulent flow in this investigation is considered to be transient, incompressible, viscous, Newtonian and isotropic.

The numerical solution involves splitting the geometry into many sub-volumes and then integrating the differential equations over these volumes to produce a set of coupled algebraic equations for the velocity components, and the pressure at the centre of each volume. The solver guesses the pressure field and then solves the discretised form of the momentum equations to find new values of the pressure and velocity components. This process continues, in iterative manner, until the convergence criterion is satisfied. In this study, simulation was started by verifying the numerical result to the experimental result. It has been done and gave good agreement. Afterward the simulation series for different snubber model was conducted (section 4).

Modeling and Grid System

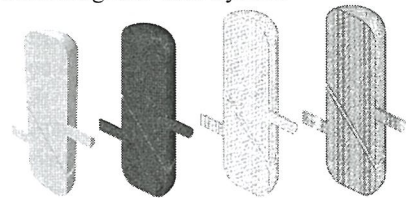


Figure 3: Computational model: (a) 3D design, (b) surface mesh in pro-Surf, (c) volume mesh and (d) boundary condition.

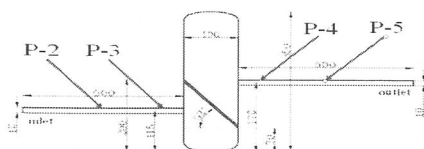


Figure 4: Dimension of computational model with observed point positions

Boundary Condition and verification of the Computational Method

In order to solve the partial differential equations described in section 2, appropriate boundary conditions must be declared for all boundaries of the computational domain. The boundary conditions associated with the computational domain as shown in figure 3(d) are inflow, outflow and symmetry plane. In order to reduce the cell number and the calculation time also, the model was made in a half type. This is regarding the symmetrical shape of the snubber. Therefore the symmetry boundary condition was applied on the cross section marked by blue color in figure 3(d).

The inflow was set to the pressure value taken from measured data form the experiment. The data is transient and waving with maximum and minimum limit. The pressure value is different for each motor frequency. Figure 4 shows the inflow data as the boundary condition for motor frequency 20, 40 and 60 Hz. Boundary condition for the outflow was set to the atmospheric pressure value, 101.325 kPa. The dimension of the model is depicted in Figure 4. In this figure the four points where the sensors were placed also are pointed. So the corresponding location in the CFD model must be the observing objects. The simulation was run for three simulations, according to different motor frequency. The mean pressure for the measuring points was compared. The result data both from experiment and CFD simulation is listed in table 1. From the data the comparison graph

can be plotted as figure 8,9 and 10. All of those three graphs show the same trend between CFD and experimental result.

3. SIMULATION FOR VARIOUS SNUBBER TYPES

The appropriate modeling and simulation setting described in previous discussion are the basic to extend the investigation for various dimensions of snubber. The purpose is to find the optimum design of it. In designing a snubber several dimension such as snubber height (H), snubber diameter (D), buffer width (W) and buffer angle (θ) must be defined. Figure 11 illustrates all of dimensions mentioned above. The simulation run in this paper were occupying three groups of snubber volume, they are $H/D = 3.23$, $H/D = 3.82$ and $H/D = 4.48$. The volume size for each models are 0.0124 mm^3 , 0.0147 mm^3 and 0.0170 mm^3 respectively. The complete dimension parameters are listed in table 1. Each model requires specific grid system shown in figure 12. Like as the simulation to get verified data, the trimmed-hexagonal cell also was used. The half-symmetry model was used to reduce the cell number. Therefore the calculation job will be reduced. Special treatment was applied to the area near the wall. In these areas the mesh size was made much smaller, so the accurate calculation should be made. Figure 12(d) shows the detailed view of cross section of the model.

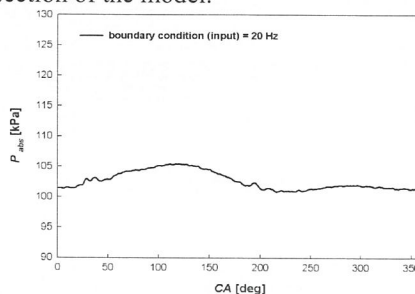


Figure 5: Inlet boundary condition at 20 Hz

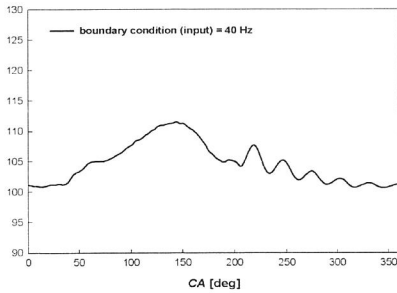


Figure 6: Inlet boundary condition at 40 Hz

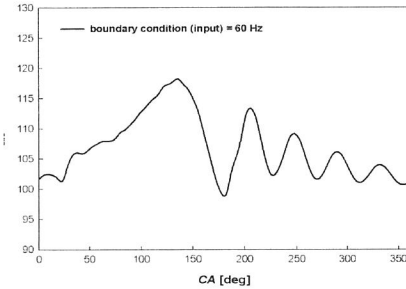


Figure 7: Inlet boundary condition at 60 Hz

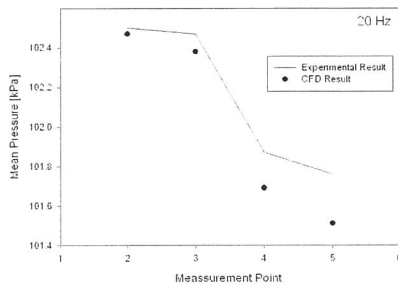


Figure 8: Comparison of numerical with experimental data at 20 Hz

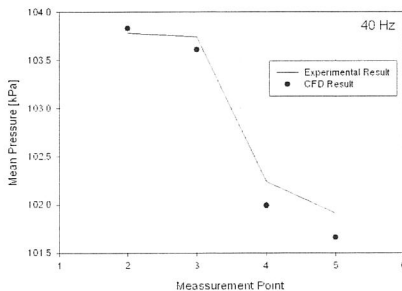


Figure 9: Comparison of numerical with experimental data at 40 Hz

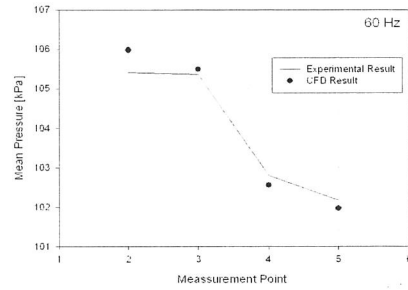


Figure 10: Comparison of numerical with experimental data at 60 Hz

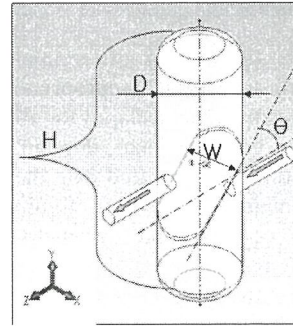


Figure 11: Modeling Notifications

This work was to solve the real problem occurred in the hydrogen gas station. Therefore in this simulation the real dimension and condition are use. The model with $H/D = 3.82$ with 0.0147 mm^3 is the one which is used in the industry field, in this paper this is named as standard model. Two other models are studied as the benchmarks. One model has volume bigger than the standard model and the other has smaller than the standard. To produce the comparable result to the real condition, so the boundary condition, material properties and numerical setting were adjusted to approach the reality. The inlet pressure for boundary condition was set to the high enough value representing the operational pressure in the real hydrogen gas plant. This mean pressure value was stated of 10 MPa with pulsation factor of 2 MPa. Figure 13 shows this inlet boundary condition. On the other hand the pressure outlet boundary condition was set of non pulsating 10 MPa.

Hydrogen gas properties applied into this computation were taken from Star CD material database. The computation was set with transient condition, compressible flow and using $k-\epsilon$ high Reynolds number turbulent model. Upwind scheme was chosen to execute the calculation for all variables, such as mass, U, V, W momentum, pressure and turbulent kinetic energy.

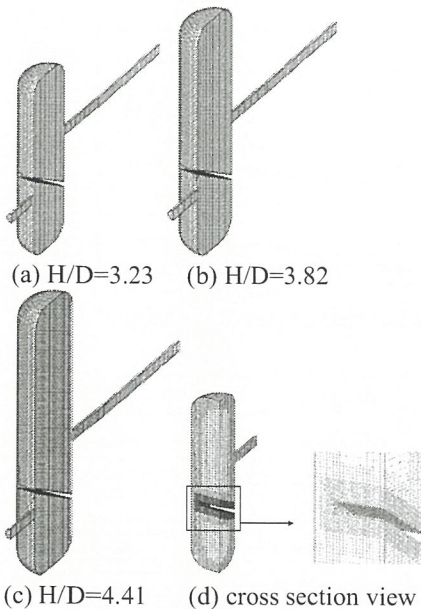


Figure 12: Grid Systems of the Models

The study is focused on pressure at point 1 and point 2 in the figure 14. These measured points, both at inlet and outlet pipe, are located 50 mm from the snubber wall. These points were chosen in order to analyze damping phenomena inside the snubber.

4. RESULT AND DISCUSSION

The simulation result the mean pressure value at point 1 and point 2. The mean pressure at point 1 is shown by figure 15, meanwhile at point 2 by figure 16. At point 1, inlet pipe, the mean pressure decrease as the increasing snubber volume. For all

snubber volume models, the trends are quite similar based on varying buffer angle. For $H/D = 3.23$, start from 10 deg up to 30 deg, the pressure quite constant. And start to increase when $\theta = 30$ deg until 50 deg. For $H/D = 3.82$, start from 10 deg up to 35 deg, the pressure quite constant. Then start to increase when $\theta = 35$ deg until 50 deg. The unique occurred in $H/D = 4.41$, when pressure decline from 10 deg until 25 deg and incline smoothly from 25 deg until 35 deg then drastically from 35 deg to 50 deg. Pressure in the without buffer model is bigger than pressure with 45 deg buffer for all models. At point 2, outlet pipe, the mean pressure also indicates a unique pattern. All models show smooth increasing from 10 deg until 50 deg.

Applying equation (1) and (2), figure 17 and figure 18 can be made. These two graphs depict the objective functions. As known before P_{TL} and P_F are the pressure characteristics those are become optimization parameters. The goal of this optimization is to minimize both of those values. As can be seen in figure 17, P_{TL} is increasing as the increasing snubber volume. All models show curved-like trend with minimum value at 35 deg. For $H/D = 3.82$ and $H/D = 4.41$, the value decline smoothly start from 10 deg until 35 deg, after that incline with very small changes. And for $H/D = 3.23$, the value decline smoothly start from 10 deg until 40 deg, after that incline up to 50 deg.

The P_F is increasing from 10 deg until 50 deg for all models. According to the fact discussed above the optimization face a problem where the minimum value for both parameters is not occurred at the same buffer angle. Therefore the advance study to determine optimum value must be developed. This should consider not only minimizing mean pressure loss and pressure pulsation but also minimizing the snubber volume.

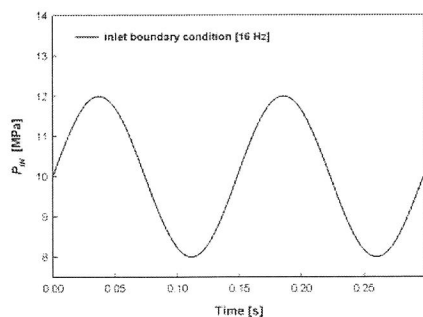


Figure 13: Inlet boundary condition

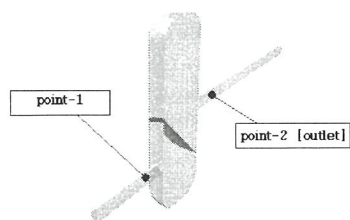


Figure 14: Measuring points at inlet and outlet pipe

Table I: Geometrical Data for Various Snubber Models

Parameters (mm)	H/D 3.23	H/D 3.82	H/D 4.41
Body diameter	170		
Body height	550	650	750
Inlet pipe diameter	30		
Outlet pipe diameter	30		
Inlet pipe length	100		
Outlet pipe length	500		
Inlet pipe position form bottom	175		
Outlet pipe position form bottom	310		
Buffer width	60		
Buffer thickness	10		
Buffer angle	10° ~ 50° (increase with 5°)		
Volume (m ³)	0.0124	0.0147	0.0170
Grid cells number	65,000	75,000	85,000

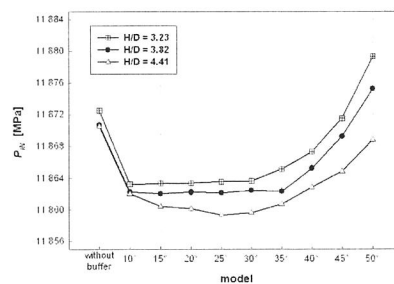


Figure 15: Mean pressure at point 1

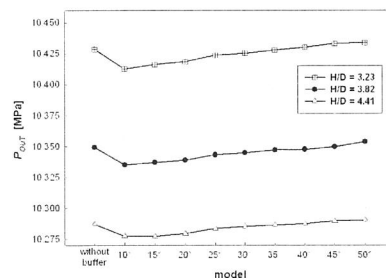


Figure 16: Mean Pressure at point 2

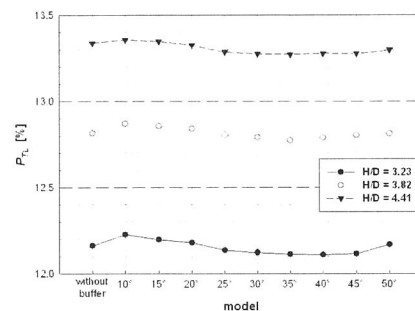


Figure 17: Pressure loss ratio of all snubber

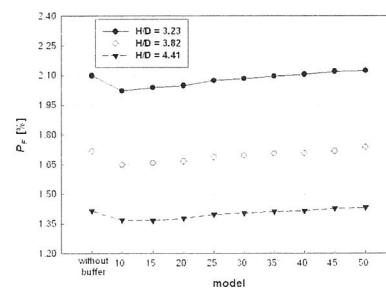


Figure 18: pressure pulsation ratio of all snubber

5. CONCLUSION

The present study has shown the computational fluid dynamics (CFD) can be applied to study pressure characteristics through the snubber. The numerical analysis on the snubber has been done and gave some information that summarized as follow: The objective in optimizing a snubber dimension is making the mean pressure loss and pressure pulsation as minimum as possible; The simulation using CFD commercial code with mentioned numerical setting can result a good representation of the real condition. This is proved by the very small difference ratio; The mean pressure both at inlet and outlet pipe show a unique characteristic shown by figure 15 and 16; The minimum value of mean pressure loss P_{TL} is occurred when buffer angle $\theta = 35$. Decreasing or increasing the buffer angle will increase the mean pressure loss P_{TL} ; The minimum value of pressure pulsation P_F is occurred when buffer angle $\theta = 10$. Increasing the buffer angle will increase pressure pulsation P_F ; The next study should solve the optimization problem not only regarding the minimum mean pressure loss and pressure pulsation but also the minimum volume of snubber.

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