Unsteady RANS Simulation of Acoustic Wave Propagation in Pulsation Damper

K A Romanov\(^1\) and G M Makaryants\(^1\)

\(^1\)Department of Automatic Systems of Power Plants, Institute of Engine and Power Plant Engineering, Samara National Research University, 34 Moskovskoe shosse st., Samara, 443086, Russia

Email: romanov.kirill.94@mail.ru

Abstract. A research on dynamic characteristics of a pipeline pulsation damper is presented in this paper. The study covers an investigation of acoustic wave propagation in a pipeline numerical model. For this purpose, the numerical model which was previously obtained for estimation of hydrodynamic noise after the damper diffuser has been verified. The verification is conducted using URANS model. The simulation results illustrate the adequacy of the present model in calculating acoustic wave propagation. Hence, the acoustic characteristics obtained in further calculations can be regarded as reliable. The numerical results demonstrate that an initially plane acoustic wave does not change its form and the overall amplitude level of acoustic oscillations varies in accordance with changes in the geometry flow cross-section.

1. Introduction
The study of oscillations in piping has become an important aspect of designing water supply and treatment systems. Acoustic waves induced by water pressure pulsations which are caused by periodic flow occurring downstream the pump, vortex shedding downstream the bends and changes in pipe cross-section are of particular interest and complexity. The pressure pulsations of working fluid may also result in unacceptable level of pipeline vibration [1, 2] as well as in high acoustic noise [3-7]. One way to solve this problem is to use pressure pulsation dampers [8-14] in the piping systems.

The acoustic noise induced by discontinuities of the damper structure is critical for damper effectiveness and efficiency in case of high flow rates. A diffuser connecting a damper with a pipeline can be considered as one of such discontinuities. In order to solve the problem of damper sound generation in a proper manner the process of vortex pulsation generation in the diffuser must be simulated. Therefore, the proposed model must have the necessary characteristics to calculate the propagation of acoustic waves.

The features of the flow passing through the diffuser and the hydrodynamic processes inside the diffuser have been widely discussed in the literature. McDonald & Fox [15] and then Kwong & Dowling [16] have demonstrated that a conical diffuser is affected by transitory stall. Dequand et al. [2] concluded that separation flow in the diffuser induces duct acoustic pulsations caused by vortex shedding.

In the observed papers great attention is paid to common phenomenon of the fluid passing through the diffuser. However, the methods and techniques of calculating the sound generation in the pipeline system components are also of particular interest. Several studies have been conducted on this issue. One part of the research papers is devoted to predict the vortex-born noise by means of semi-empirical
techniques. For example, Karekull et al. [5] searched for a scaling law for the sound power of the orifices, bends and dampers. The research in the other part of the papers is focused on applying Unsteady Computational Fluid Dynamics (CFD) methods in order to directly simulate the vortex sound generation, see, for example Lam et al. [17], Gloerfelt & Lafon [18], Singh & Rubini [13]. This approach allows us to obtain more detailed information on vortex noise.

Although considerable research has been devoted to direct computation of the noise radiated by vortex flows, rather less attention has been paid to CFD calculation of turbulent flows in a diffuser. The third-octave spectrum obtained from the numerical simulation of the damper agrees well in the low-frequency region, however, at higher frequencies there is a strong discrepancy with the experimental data in the newly released research paper [19]. Thus, it was necessary to check the impact of mesh resolution on direct hydrodynamic noise computation.

The aim of this paper is to verify the proposed numerical model of the pulsation damper in order to simulate pressure wave passed through the damper duct obstructed by the diffuser.

2. Operation principle of a piping noise damper

The piping noise damper consists of a non-flowing vessel placed parallel to a constricted duct (figure 1). In high frequency range the vessel has significant acoustic conductivity while the constricted duct poses high acoustic resistance. That is why in high frequency range the flow oscillation energy is isolated inside the non-flowing vessel and it does not pass through the damper. Isolated oscillation energy is attenuated in hydraulic throttles connected to the constricted duct and the non-flowing vessel. The vessel and the duct act as reactive wave resistances and the hydraulic throttle is the active one. While the wave impedance of the pipeline is frequency independent and active the hydraulic throttle must provide the similar character of wave resistant in the inlet and outlet of the damper. Therefore, this type of dampener is called the dampener with constant active resistance [12].

![Figure 1. Schematic representation of the damper: (1) – dampener housing assembly, (2) – constricted duct, (3) – non-flowing vessel, (4) – hydraulic throttles, (5, 6) – connecting flanges.](image)

3. Methods

The process of the modeling of the acoustic wave propagation was devoted to the flow analysis using the unsteady Reynolds-averaged Navier-Stokes (URANS) model. Calculating the acoustic wave propagation with the flow rate assumed to be zero involves the user-defined functions for specifying the pressure pulsations at the inlet. During the calculation the low-Reynolds $k - \omega$ SST turbulence model was used to closure governing equations. The choice of such turbulence model was made due to its best ability to predict the flow separation from the wall [20]. Figure 2 depicts the geometry of flow domain region.
3.1 Governing equations

The prediction of pressure pulsations in the diffuser as well as their propagation requires the calculation of the flow parameters time dependency. For this purpose the three-dimensional complete system of hydrodynamics equations which consists of the continuity equation:

\[
\frac{\partial \rho}{\partial t} + \sum_{j=1}^{3} \frac{\partial (\rho u_j)}{\partial x_j} = 0,
\]

and the momentum equation

\[
\frac{\partial \rho u_j}{\partial t} + \sum_{i=1}^{3} \frac{\partial (\rho u_j u_i)}{\partial x_i} = -\frac{\partial p}{\partial x_j} + \frac{\partial}{\partial x_j} \left( \tau_{ij} \right),
\]

where \( \rho \) is a density, \( u_j \) and \( x_j (j = 1,2,3) \) are components of velocity and displacement on the axes of a Cartesian system, \( p \) is a pressure, \( \tau_{ij} \) is a component of stress tensor, \( t \) is a time, is solved. The complete system of hydrodynamics equations includes the state equation. The assumption of the adiabatic wave processes allows one to eliminate the temperature from the state equation. Density and pressure are correlated using Tait equation of state:

\[
\rho = \rho_0 \sqrt{\frac{K_0 + n \Delta p}{K_0}},
\]

where \( \rho_0 \) is a reference fluid density, \( K_0 \) is a reference bulk modulus, \( n \) is a density exponent, \( \Delta p = p - p_0 \), \( p \) is an absolute fluid pressure, \( p_0 \) is an absolute reference fluid pressure.

3.2 Boundary conditions and time discretization

The evaluation of acoustic oscillations has been conducted using URANS analysis. Harmonic pressure of the fluid was set as a boundary condition at the inlet. Flow rate was set to zero at the outlet, thus ensuring the propagation of acoustic waves.

The time step value which was selected in the preceding research based on the required maximum frequency in the spectrum of vortex pulsations has been considered as insufficient. In the present paper the Courant number has been regarded as a key factor when choosing the time step value:

\[
\Delta t = CFL \cdot \frac{\Delta x_{\text{min}}}{c_{\infty}},
\]

where \( \Delta t \) is the time step value, CFL is the Courant number which describes how fluid is moving through computational cells, \( \Delta x_{\text{min}} \) is the minimal cell size, \( c_{\infty} \) is the speed of sound in a medium.

4. Results and discussions

4.1. Comparison of different time step values

In what follows, the results of the numerical calculation of the acoustic wave propagation are described. The acoustic wave propagation was calculated in order to determine whether the generated
A mesh model can adequately calculate acoustics, i.e. whether the wave changes during its downstream propagation as the wave should remain plane at the outlet and how the mesh size affects the results of the further calculation. Figure 3 depicts the results of pressure pulsation calculation for different time step values selected based on the Courant number for $CFL = 10, \Delta x_{\min} = 10^{-3} \text{ mm}$ and $c_{\infty} = 1200 \text{ m/s}$. The data has been obtained from the center points of various cross-sections (figure 2, points 3.0 and 6.0).

![Figure 3. Pressure pulsations for different time step values](image)

(a) Point 3.0  
(b) Point 6.0

As can be seen from the graphs on figure 3, the time step $\Delta t = 10^{-5} \text{ s}$ provides the necessary accuracy, but it does not require much calculation time as for $\Delta t = 5 \cdot 10^{-6} \text{ s}$. On the other hand, the time step $\Delta t = 2 \cdot 10^{-5} \text{ s}$ does not provide adequate results of the processes which occur in the pipeline.

Figure 4 shows the static pressure distribution field for the case of $\Delta t = 10^{-5} \text{ s}$ and $t = 0.0075 \text{ s}$ which illustrates the wave propagation.

![Figure 4. Static pressure distribution](image)

### 4.2 Check of the wave planeness

To check the planeness of the propagating acoustic wave, measurements have been taken at 5 points in 8 different cross-section planes for the time step $\Delta t = 10^{-5} \text{ s}$. In the case of a plane wave, all values of the pressure oscillations should not differ by more than 5% in each cross-section, which can be observed in figure 5. Figure 5 depicts the results of pressure pulsation calculation for different cross-sections. The data has been obtained from cross-sections 2 and 7 (figure 2, points 2.0, 2.1, 2.3, 2.5, 2.7 and 7.0, 7.1, 7.3, 7.5, 7.7).
Figure 5. Pressure pulsations for different cross-sections

The graph from figure 6 which shows the overall level of pressure oscillations for the time step $\Delta t = 10^{-5}$ s also demonstrates that the proposed numerical model allows one to adequately calculate the acoustics and that the discrepancy of the calculation data with the experimental one from [20] is not connected with the characteristics of the mesh model. The data has been obtained from the center points of every cross-section.

Figure 6. Overall level of pressure pulsations

From all of the above, we can conclude that the calculated model allows us to adequately calculate the propagation of acoustic waves, while one should pay attention to the calculation parameters, for example, the time step value.

5. Conclusion
A numerical simulation of acoustic wave propagation was performed. It was noted that the accepted time step value calculated from the Courant number is sufficient to obtain adequate data. An acoustic wave passing through the damper model is a plane wave. The proposed design model makes it possible to further calculate vortex pulsations, for more accurate calculation of which further investigations are required.

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