Extension of Modal Analysis Results to the Operational Behavior of High-Vulnerability Piping at the Cernavodă Nuclear Power Plant

Andrei-Cristian COTOBAN*
Institute of Solid Mechanics of the Romanian Academy, 15 Constantin Mille Str., 010141 Bucharest, Romania, andrei.cotoban@imsar.ro

Dan ION
Institute of Solid Mechanics of the Romanian Academy, 15 Constantin Mille Str., 010141 Bucharest, Romania, dan.ion@icecon.ro

Polidor BRATU
Institute of Solid Mechanics of the Romanian Academy, 15 Constantin Mille Str., 010141 Bucharest, Romania, icecon@icecon.ro

* Author to whom correspondence should be addressed

Abstract: - The present work examines the influence of turbulent flow in pipeline systems at various fluid (water) flow rates, with an emphasis on the vibrations induced by this flow and the potential occurrence of resonance phenomena in specific sections of the pipe. To conclude, the study considers the Vortex-Induced Vibration (VIV) mechanism, associated with the periodic shedding of vortices in regions featuring geometric discontinuities such as elbows, tees, valves, reducers, and compares it with the natural vibration modes of the investigated system.

A structural modal analysis of the piping system is conducted using AutoPIPE to identify the natural vibration modes and corresponding participation factors. Flow-induced excitation frequencies associated with internal fittings are estimated using Strouhal-based correlations under representative operating conditions. The proximity between excitation frequencies and structural natural frequencies is quantitatively assessed to identify vibration modes potentially prone to resonance.

The results indicate that the first three and 11th vibration modes exhibit the greatest dynamic interaction influence under potential excitations. The proposed approach provides a practical, operational screening methodology for identifying vibration-resonance scenarios in existing piping systems.

Keywords: - modal analysis, turbulent flow, natural vibration modes, flow-induced vibration, excitation frequency, vortex-induced vibration.

1. INTRODUCTION

The issue of vibrations in pipeline systems must be carefully analyzed and monitored, as such vibrations lead to material fatigue in the pipes, with significant consequences for the processes in which these systems operate [1]. Failures, such as cracking and weld-region ruptures, are critical in nuclear power plants and must be prevented [2].

Flow-rate variations are common operational procedures in industrial installations and have a direct impact on the flow regime within the pipes, thereby influencing the occurrence of flow-induced vibration phenomena. In this context, analyzing the effects of such variations is essential for identifying conditions that may give rise to resonance and for assessing the safety and reliability of the pipeline system [3].

Previous studies have extensively addressed flow-induced vibration phenomena in industrial piping systems subjected to turbulent internal flow [4]. Wachel et al. [3] reported that geometric discontinuities such as elbows, tees, and control valves may act as sources of hydrodynamic excitation, leading to vibration levels that can coincide with the natural frequencies of the piping system and cause fatigue damage.

Naudascher [5] provided a comprehensive classification of internal flow-induced vibration mechanisms, highlighting the role of unsteady flow separation and vortex formation in generating characteristic excitation frequencies that may interact with structural modes [6,7]. Similar vibration problems in energy and power-plant piping systems [8] were documented in industrial studies, emphasizing the importance of modal analysis as a screening tool for identifying resonance-prone configurations.

In line with these studies, the present work focuses on turbulent flow as a function of the fluid piping's geometric configuration, along with an analysis of the associated dynamic interaction processes [9].

In this context, the objective of the present work is to evaluate the potential interaction by analyzing flow-induced excitation frequencies associated with internal fittings and the natural vibration modes [10] of an existing high-vulnerability pipeline system operating under steady-state conditions [11,12].

To this end, the study considers the Vortex-Induced Vibration (VIV) mechanism [13], which involves the periodic shedding of vortices in regions featuring geometric discontinuities, such as elbows, tees, valves, and reducers. The phenomenon was characterized using the Strouhal number, which describes the oscillatory nature of the flow and enables the determination of the vortex-shedding frequency. Subsequently, the results were compared with the natural vibration modes of the pipeline system obtained from its modal analysis [13].

2. MODAL ANALYSIS OF THE STUDIED PIPELINE SYSTEM

The pipeline system subjected to analysis, shown in Figure 1, is an existing system at the Cernavodă Nuclear Power Plant, known as the thermal cycle regeneration system. From the turbine high-pressure stage, the expanded steam enters the High-Pressure Preheater (HTR), where it condenses. The resulting condensate then enters the pipeline section under analysis (G001) and is transported to the degaser storage tanks and to the condenser by differential pressure. The branch to the condenser is not included in this analysis. The absolute pressure is between 1.17 and 1.3 MPa (a) at HTR and 0.57 MPa (a) at the degasser.

To determine the natural vibration modes of the pipeline system shown in Figure 1, the available input data were used.

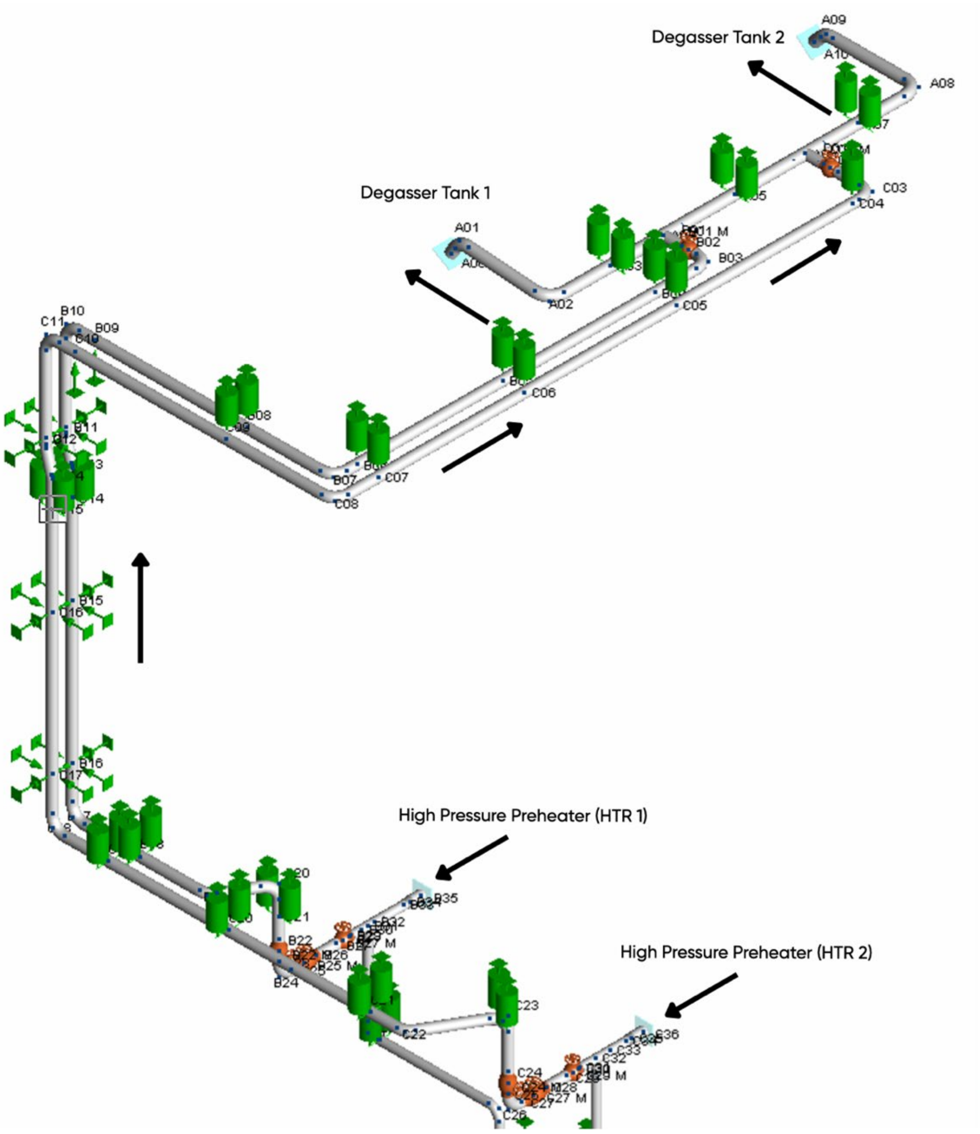

Figure 1. Pipeline isometric

The modeling of the pipeline system and the modal analysis [14] were performed using Bentley AutoPIPE CONNECT Advanced Edition, version 24.00.02.243, based on the input data specified below:

Operating parameters:
- Fluid temperature: 151ºC
- Fluid pressure:
  - Inlet: between 1.17 and 1.3 MPa (a)
  - Outlet: 0.57 MPa (a)
- Fluid flow: 160 T/h

Pipe Dimensions and Characteristics:
- Pipe diameters: 10” (273x8mm), 12” (324x8mm), 14” (356x12mm)
- Pipe material: ASTM A106 Gr. B
- The pipe is thermally insulated with mineral wool and plate protection

Supports characteristics:
- Spring hangers support (elastic)
- Rigid support
- Anchor supports

Design code: ASME B31.3

The method used in the modal analysis [14] within AutoPIPE software is the finite element method (FEM) [15], based on the discretization procedure.

AutoPIPE distributes the mass across multiple points of the pipeline system, employing a lumped-mass model, and solves it using finite elements with high precision.

The analysis focused on determining the first 14 vibration modes.

The participation factor was calculated using the following relationship [16]:

$$\gamma_i = \{\phi\}_i^T \left[M\right]\{D\} \quad (1)$$

where:

$\{\phi\}_i^T$ – vibration mode

$\left[M\right]$ – mass matrix

$\{D\}$ – displacement vector

The participation factors associated with each vibration mode are presented in Table 1. The corresponding captured modal mass for each mode is summarized in Table 2.

From the obtained results, it can be observed that the vibration modes [6, 7] significant for the pipeline system, in which a substantial percentage of the system’s total mass is involved, are:
- 1st vibration mode, 0.4892 Hz, captured modal mass on Z axis 9.337%
- 2nd vibration mode, 0.5848 Hz, captured modal mass on Z axis 3.188%
- 3rd vibration mode, 0.8317 Hz, captured modal mass on X axis 6.115%
- 11th vibration mode, 1,7564 Hz, captured modal mass on X axis 7.459%

Table 1. Participation factors for each mode of vibration

Mode	Freq.	Period	Participation Factors		
no.	Hz	(Sec)	X	Y	Z
1	0.4892	2.044	1.7	-0.1	-5.1
2	0.5848	1.71	-0.3	-0.4	-3
3	0.8317	1.202	-4.1	0	-1.8
4	0.9125	1.096	-2.7	0	1.5
5	0.9913	1.009	1.4	-0.2	0.6
6	1.3586	0.736	0.6	0.3	0.3
7	1.4822	0.675	-0.9	4.1	-0.1
8	1.5365	0.651	0.3	1.4	4.1
9	1.5868	0.63	-2.4	3.5	0.3
10	1.7174	0.582	-2	-0.2	0.3
11	1.7564	0.569	-4.5	-2.7	1
12	1.8953	0.528	-0.5	2.1	1.6
13	2.0911	0.478	1.3	-3.2	0.9
14	2.1561	0.464	0.1	4.4	0.7

Table 2. Capture modal mass for each mode of vibration

Mode	Freq.	Captured Modal Mass (%)			
no.	Hz	X	Y	Z	Avg.
1	0.4892	1.078	0.004	9.337	3.473
2	0.5848	0.03	0.049	3.188	1.089
3	0.8317	6.115	0	1.214	2.443
4	0.9125	2.758	0.001	0.812	1.19
5	0.9913	0.682	0.01	0.115	0.269
6	1.3586	0.144	0.04	0.031	0.072
7	1.4822	0.324	5.982	0.002	2.103
8	1.5365	0.033	0.739	6.05	2.274
9	1.5868	2.022	4.596	0.028	2.215
10	1.7174	1.492	0.011	0.025	0.509
11	1.7564	7.459	2.722	0.341	3.507
12	1.8953	0.094	1.653	0.946	0.898
13	2.0911	0.594	3.651	0.316	1.52
14	2.1561	0.005	6.904	0.193	2.367

3. DETERMINATION OF FLUID FLOW PARAMETERS

The pipeline system transports water at a temperature of 151 °C with a mass flow rate of 160 t/h. For both the transported mass flow rate and other scenarios, the flow parameters were determined for the following mass flow rates: 80 t/h, 100 t/h, 120 t/h, 140 t/h, 160 t/h, 180 t/h, and 200 t/h. The Reynolds number formula is [17,18,19]:

Table 3. Reynolds number for each flow and pipe inner diameters

Masic flow	Inner diameter (D)	Fluid velocity	Reynolds number
t/h	m	m/s	-
80	0.3556	0.24	439580
80	0.3048	0.33	512843
80	0.254	0.48	615412
100	0.3556	0.31	549475
100	0.3048	0.42	641054
100	0.254	0.60	769264
120	0.3556	0.37	659370
120	0.3048	0.50	769264
120	0.254	0.72	923117
140	0.3556	0.43	769264
140	0.3048	0.58	897475
140	0.254	0.84	1076970
160	0.3556	0.49	879159
160	0.3048	0.67	1025686
160	0.254	0.96	1230823
180	0.3556	0.55	989054
180	0.3048	0.75	1153897
180	0.254	1.08	1384676
200	0.3556	0.61	1098949
200	0.3048	0.83	1282107
200	0.254	1.20	1538529

$$Re = \frac{\rho \cdot u \cdot L}{\mu} \tag{2}$$

where:

- Re – Reynolds number
- $\boldsymbol{\rho}$ – density of the transported fluid, i.e., water at 151 °C: 915.7 kg/m^3
- u − fluid flow velocity determined from the relationship between the flow rate and the pipe diameter at various sections
- L – inner pipe diameter at various sections
- $\boldsymbol{\mu}$ – dynamic viscosity of the fluid at 151 °C (0.0001811 Pa·s)

The Reynolds numbers for each flow and pipe inner diameter are summarized in Table 3.

It is determined that the Reynolds number ranges between 439580 and 1538529, and considering the roughness of the SA106 grade B steel pipe (Table 4 [20]), places the flow in the turbulent regime (Figure 2).

Table 4. SA106 grade b pipe roughness [20]

Type	Roughness ε [mm]	ε/D min	ε/D max
welded	0.05 to 0.1	0.0001407	0.0003937
ASTM A106 Grade B, drawn	0.02 to 0.1	0.0000563	0.0003937
A106 GR B, cleaned	0.15 to 0.2	0.0004218	0.0007874
galvanized, new	0.15	0.0004218	0.0005906
severely corroded	0.4 to 3	0.001125	0.011811
lightly corroded	0.1 to 0.4	0.0002812	0.001575
heavy scaling	1.5 to 4	0.004218	0.015748
light scaling	1 to 1.5	0.002812	0.005906
bitumen-coated	0.05	0.0001407	0.0001969

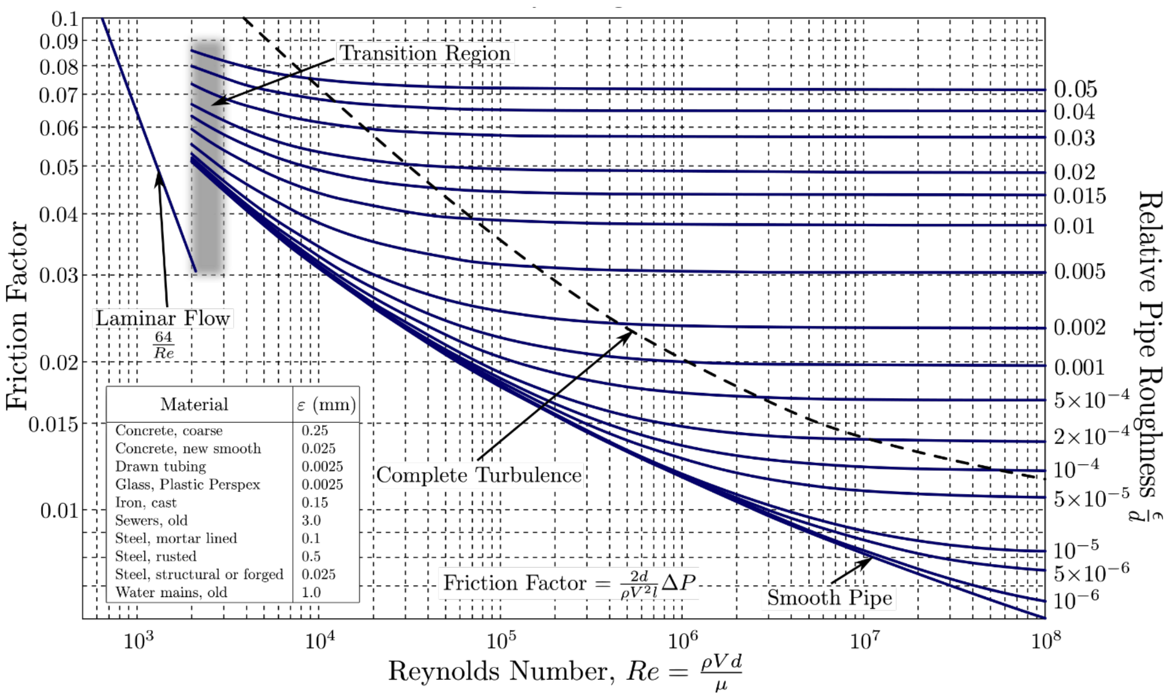

Figure 2. Moody diagram [21]

4. DETERMINATION OF THE STROUHAL NUMBER AND EXCITATION FREQUENCY

For the present analysis, the Strouhal number (St) was adopted as an approximate value based on the range reported by Chen (2024) [22] for internal flow in a tee with a closed branch, which lies between 0.3 and 0.6.

The values used in our simulation were: 0.3 for a 90° elbow, 0.25 for a 45° elbow, 0.35 for a tee, and 0.3 for a flow control valve. While the value for the tee is directly supported by Chen [22], the values for elbows and the valve are conservative estimates consistent with typical ranges for internal flow fittings, considering the Strouhal Reynolds diagram as shown in Figure 3.

These estimates provide reasonable approximations for the excitation frequencies of the fittings under internal flow conditions and ensure alignment with the interval reported by Chen [22].

For each 90° elbow, 45° elbow, tee, and control valve, the excitation frequency was determined using the following Strouhal equation [5,18]:

$$f = \frac{S_t \cdot U}{D} \tag{3}$$

where:
- f – excitation frequency
- S_t – Strouhal number
- U – fluid flow velocity determined from the relationship between the flow rate and the pipe diameter at various sections
- D – inner pipe diameter at various sections of elbows, tees, and valves (0.3556 m, 0.3048 m, 0.254 m) as in Tables 5-11

The Strouhal number and excitation frequency for 80, 100, 120, 140, 160, 180, and 200 t/h flows are presented in the following Tables 5 to 11.

Table 5. Strouhal number and excitation frequency for 80 t/h flow

Element type	**Flow**	Inner diam.	Strouhal number	**Excitation frequency**
-	**t/h**	m	-	**Hz**
90° elbow	**80**	0.3556	0.3	**0.2063**
45° elbow	**80**	0.3556	0.25	**0.1719**
Tee	**80**	0.3556	0.35	**0.2406**
FCV*	**80**	0.3556	0.3	**0.2063**
90° elbow	**80**	0.3048	0.3	**0.3275**
45° elbow	**80**	0.3048	0.25	**0.2729**
Tee	**80**	0.3048	0.35	**0.3821**
FCV	**80**	0.3048	0.3	**0.3275**
90° elbow	**80**	0.254	0.3	**0.5660**
45° elbow	**80**	0.254	0.25	**0.4716**
Tee	**80**	0.254	0.35	**0.6603**
FCV*	**80**	0.254	0.3	**0.5660**

*Flow Control Valve

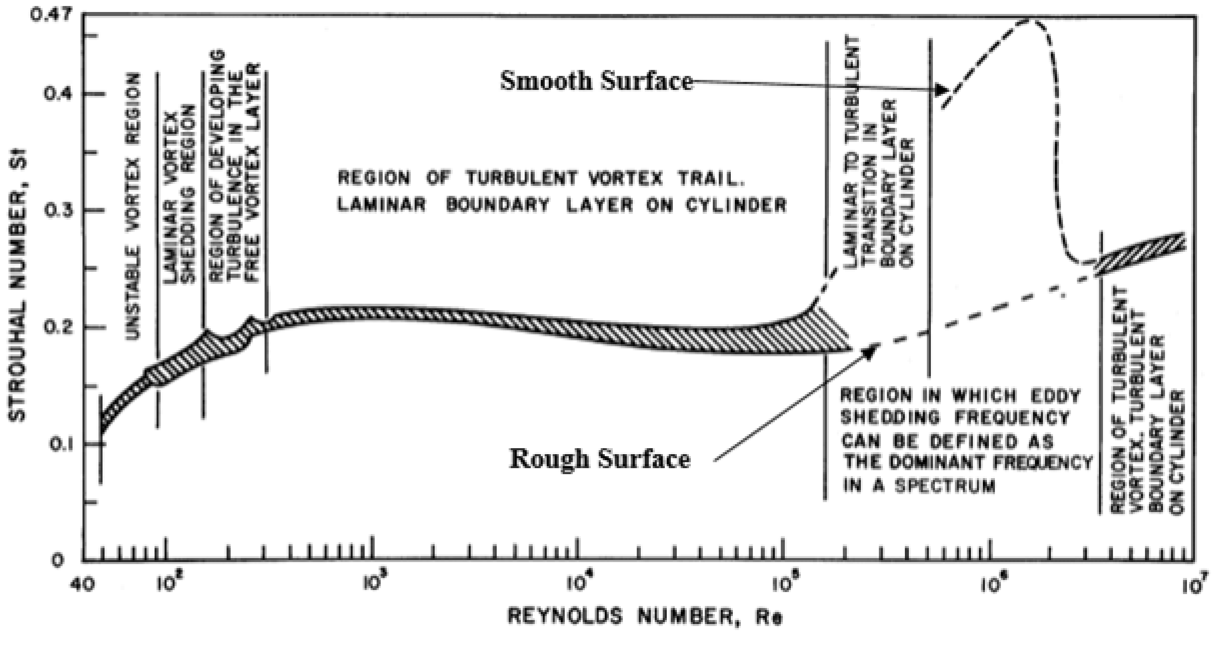

Figure 3. Strouhal-Reynolds diagram [23]

Table 6. Strouhal number and excitation frequency for 100 t/h flow

Element type	**Flow**	Inner diam.	Strouhal number	**Excitation frequency**
-	**t/h**	m	-	**Hz**
90° elbow	**100**	0.3556	0.3	**0.2578**
45° elbow	**100**	0.3556	0.25	**0.2148**
Tee	**100**	0.3556	0.35	**0.3008**
FCV*	**100**	0.3556	0.3	**0.2578**
90° elbow	**100**	0.3048	0.3	**0.4094**
45° elbow	**100**	0.3048	0.25	**0.3412**
Tee	**100**	0.3048	0.35	**0.4776**
FCV	**100**	0.3048	0.3	**0.4094**
90° elbow	**100**	0.254	0.3	**0.7074**
45° elbow	**100**	0.254	0.25	**0.5895**
Tee	**100**	0.254	0.35	**0.8254**
FCV*	**100**	0.254	0.3	**0.7074**

*Flow Control Valve

Table 7. Strouhal number and excitation frequency for 120 t/h flow

Element type	**Flow**	Inner diam.	Strouhal number	**Excitation frequency**
-	**t/h**	m	-	**Hz**
90° elbow	**120**	0.3556	0.3	**0.3094**
45° elbow	**120**	0.3556	0.25	**0.2578**
Tee	**120**	0.3556	0.35	**0.3609**
FCV*	**120**	0.3556	0.3	**0.3094**
90° elbow	**120**	0.3048	0.3	**0.4913**
45° elbow	**120**	0.3048	0.25	**0.4094**
Tee	**120**	0.3048	0.35	**0.5732**
FCV	**120**	0.3048	0.3	**0.4913**
90° elbow	**120**	0.254	0.3	**0.8489**
45° elbow	**120**	0.254	0.25	**0.7074**
Tee	**120**	0.254	0.35	**0.9904**
FCV*	**120**	0.254	0.3	**0.8489**

*Flow Control Valve

Table 8. Strouhal number and excitation frequency for 140 t/h flow

Element type	**Flow**	Inner diam.	Strouhal number	**Excitation frequency**
-	**t/h**	m	-	**Hz**
90° elbow	**140**	0.3556	0.3	**0.3609**
45° elbow	**140**	0.3556	0.25	**0.3008**
Tee	**140**	0.3556	0.35	**0.4211**
FCV*	**140**	0.3556	0.3	**0.3609**
90° elbow	**140**	0.3048	0.3	**0.5732**
45° elbow	**140**	0.3048	0.25	**0.4776**
Tee	**140**	0.3048	0.35	**0.6687**
FCV	**140**	0.3048	0.3	**0.5732**
90° elbow	**140**	0.254	0.3	**0.9904**
45° elbow	**140**	0.254	0.25	**0.8254**
Tee	**140**	0.254	0.35	**1.1555**
FCV*	**140**	0.254	0.3	**0.9904**

*Flow Control Valve

Table 9. Strouhal number and excitation frequency for 160 t/h flow

Element type	**Flow**	Inner diam.	Strouhal number	**Excitation frequency**
-	**t/h**	m	-	**Hz**
90° elbow	**160**	0.3556	0.3	**0.4125**
45° elbow	**160**	0.3556	0.25	**0.3438**
Tee	**160**	0.3556	0.35	**0.4813**
FCV*	**160**	0.3556	0.3	**0.4125**
90° elbow	**160**	0.3048	0.3	**0.6550**
45° elbow	**160**	0.3048	0.25	**0.5459**
Tee	**160**	0.3048	0.35	**0.7642**
FCV	**160**	0.3048	0.3	**0.6550**
90° elbow	**160**	0.254	0.3	**1.1319**
45° elbow	**160**	0.254	0.25	**0.9433**
Tee	**160**	0.254	0.35	**1.3206**
FCV*	**160**	0.254	0.3	**1.1319**

*Flow Control Valve

Table 10. Strouhal number and excitation frequency for 180 t/h flow

Element type	**Flow**	Inner diam.	Strouhal number	**Excitation frequency**
-	**t/h**	m	-	**Hz**
90° elbow	**180**	0.3556	0.3	**0.4641**
45° elbow	**180**	0.3556	0.25	**0.3867**
Tee	**180**	0.3556	0.35	**0.5414**
FCV*	**180**	0.3556	0.3	**0.4641**
90° elbow	**180**	0.3048	0.3	**0.7369**
45° elbow	**180**	0.3048	0.25	**0.6141**
Tee	**180**	0.3048	0.35	**0.8597**
FCV	**180**	0.3048	0.3	**0.7369**
90° elbow	**180**	0.254	0.3	**1.2734**
45° elbow	**180**	0.254	0.25	**1.0612**
Tee	**180**	0.254	0.35	**1.4856**
FCV*	**180**	0.254	0.3	**1.2734**

*Flow Control Valve

Table 11. Strouhal number and excitation frequency for 200 t/h flow

Element type	**Flow**	Inner diam.	Strouhal number	**Excitation frequency**
-	**t/h**	m	-	**Hz**
90° elbow	**200**	0.3556	0.3	**0.5156**
45° elbow	**200**	0.3556	0.25	**0.4297**
Tee	**200**	0.3556	0.35	**0.6016**
FCV*	**200**	0.3556	0.3	**0.5156**
90° elbow	**200**	0.3048	0.3	**0.8188**
45° elbow	**200**	0.3048	0.25	**0.6823**
Tee	**200**	0.3048	0.35	**0.9553**
FCV	**200**	0.3048	0.3	**0.8188**
90° elbow	**200**	0.254	0.3	**1.4149**
45° elbow	**200**	0.254	0.25	**1.1791**
Tee	**200**	0.254	0.35	**1.6507**
FCV*	**200**	0.254	0.3	**1.4149**

*Flow Control Valve

5. RESULTS AND DISCUSSIONS

A ±20% frequency proximity criterion was adopted as a conservative vibration screening measure. Similar frequency separation margins are recommended in established industrial vibration design practice, for example, in API 610 [24] for rotating equipment and in API 618 [25] for compressor installations and associated piping systems, to avoid coincidence between excitation frequencies and mechanical natural frequencies. All excitation cases reported in Tables 5–11 were considered, as each case represents a distinct excitation occurrence associated with a specific fitting, diameter, and operating condition.

Based on this criterion, 52 out of 84 excitation cases fall within the ±20% frequency proximity bands of the considered vibration modes (Modes 1, 2, 3, and 11). Due to partial overlap between the proximity bands of adjacent modes, some excitation frequencies are relevant to more than one vibration mode and are therefore counted accordingly. Specifically, 23 out of 84 excitation cases fall within the ±20% frequency band of Mode 1, 22 cases within that of Mode 2, 20 cases within that of Mode 3, and 4 cases within that of Mode 11. This behavior reflects the physical possibility that a given flow-induced excitation may interact with multiple structural modes.

The obtained distribution demonstrates that the lower-order vibration modes dominate the potential resonance scenarios, while Mode 11 represents a secondary contribution that should be analyzed at higher excitation frequencies.

The Strouhal numbers adopted in Tables 5–11 are used to estimate the characteristic flow-induced excitation frequencies for the investigated fittings under different flow rates and pipe diameters. These excitation frequencies define the frequency range in which hydrodynamic excitations may act on the piping system. When compared with the structural natural frequencies obtained from the modal analysis, the results allow the identification of operating conditions for which flow-induced excitations may interact with the vibration modes of the pipeline. Therefore, Tables 5–11 provide the basis for the resonance screening assessment performed in this study.

6. CONCLUSIONS

Many of the excitation frequencies determined for the analyzed flow rates, corresponding to the fluid flow in the pipeline system, often exhibit values close to the first three and the 11th natural vibration frequencies obtained from the modal analysis performed in Bentley AutoPIPE.

This overlap indicates the potential occurrence of resonance phenomena, particularly in regions with geometric elements that disrupt the flow, such as elbows, tees, and control valves. In these areas, hydrodynamic excitations can generate structurally amplified vibrations, leading over time to cyclic loading and material fatigue processes. The consequences may include the initiation and propagation of cracks in the pipe wall or weld regions, affecting the integrity and service life of the installation.

The proposed methodology can be applied as a preliminary vibration screening tool for existing industrial piping systems operating under steady-state conditions.

By combining structural modal analysis with Strouhal-based estimates of flow-induced excitation frequencies, the approach enables the identification of operating flow rates and pipeline configurations that may be prone to flow-induced resonance. This makes the method particularly suitable for high-vulnerability installations, such as power-generation and nuclear facilities, where conservative screening assessments are required prior to detailed dynamic investigations.

The results of the present study can support operational decision-making related to flow-rate management, system modifications, or the prioritization of pipeline sections for enhanced monitoring and inspection. In this context, the methodology provides a practical basis for identifying critical scenarios that may warrant further detailed analysis.

In addition, future work could focus on quantifying the amplitude of the oscillatory hydrodynamic forces acting on the pipe walls in the context of flow-induced vibrations. Such forces could subsequently be introduced in a structural analysis software (e.g., as harmonic loads in AutoPIPE), enabling a more detailed assessment of vibration response and stress levels beyond the frequency-based screening approach adopted in the present study. Further extensions may include the investigation of transient operating conditions and experimental or numerical validation of the excitation mechanisms.

The presented results refer exclusively to the steady-state, single-phase flow regime under nominal operating conditions. Transient or nonlinear effects—such as water hammer, flow pulsations, or two-phase flow regimes—that could further amplify vibration levels were not considered in this analysis.

REFERENCES

[1] Posea, N., Anghel, A., Grigore, N., Mincu, V., *Statics and Dynamics of Pipeline Systems,* Romanian Academy Publishing House, 1996.

[2] Collet, Å., Källman, M., Pipe Vibrations. Measurements, vol. Report 2017:351. Energiforsk, 2017. Available online: https://energiforskmedia.blob.core.windows.net/media/22212/pipe-vibrations-energiforskrapport-2017-351.pdf (accessed on 15 November 2025).

[3] Wachel, J. C., Morton, S. J., Atkins, K. E., Piping Vibration Analysis, 1990, doi: https://doi.org/10.21423/R16M23 (accessed on 15 November 2025).

[4] Zachwieja, J., Stress analysis of vibrating pipelines, *AIP Conference Proceedings*, 2017, doi: https://doi.org/10.1063/1.4977691 (accessed on 15 November 2025).

[5] Naudascher, E., *Flow-induced Vibrations: an Engineering Guide*. Routledge, 2017.

[6] Bratu, P., *Theoretical Mechanics.* Bucharest: Impuls Publishing House, 2006.

[7] Wachel, J. C., *Piping Vibration and Stress, Engineering Dynamics Inc.*, Available online: https://engdyn.com/wp-content/uploads/2020/04/27-piping_vibration_and_stress_-_jcw.pdf (accessed on 15 November 2025).

[8] Ayazuddin, S. K., Qureshi, A. A., Hayat, T., Vibration analysis of primary inlet pipe line during steady state and transient conditions of Pakistan research reactor-1, *OSTI OAI (U.S. Department of Energy Office of Scientific and Technical Information)*, Nov. 1997.

[9] Chae, J., Wood Group, Considerations for Flexibility Analyses of Piping Systems in Vibration Applications, Gas Machinery Conference 2017, Pittsburgh, PA, USA, 2017, Available online: https://vdn.woodplc.com/assets/pdfs/Technical_Articles/GMRC/Considerations_for_Flexibility_Analyses_of_Piping_Systems_in_Vibration_Applications_R1.pdf (accessed on 15 November 2025).

[10] Radeș, M., *Mechanical Vibrations*, Printech Publishing House, December 2008.

[11] Nicolae, G., Nitu, M. C., Bratu, P., Iliescu, M., Furdui-Lunguț, E.*, & Dogaru, M. (2025). Vibrations Transmitted to The Human Body Under the Dynamic Action of Surfaces and Mechanical Contact Points with Impact on Human Health. *Romanian Journal of Acoustics and Vibration*, 22(1), 66-72.

[12] Nițu, M.C., Iliescu, M., Furdu-Lunguț, E. Specific Protection Requirements and Psychosensory Effects to the Noise Generated by Construction Equipment. *ACTA Technica Napocensis* - Series: Applied Mathematics, Mechanics, and Engineering, [S.l.], v. 68, n. 1 & 2, jun. 2025. ISSN 2393–2988.

[13] Nakamura, T., Kaneko, S., *Flow Induced Vibrations.* Elsevier, 2008.

[14] Bratu, P., *Dynamic isolation.* Bucharest: Impuls Publishing House, 2021.

[15] Olaru, D., *Seismic dynamics and engineering for coupled systems*. Timpul Publishing House, 2001.

[16] Bratu, P., *Analysis of elastic structures. Behavior to static and dynamic actions*. Bucharest: Impuls Publishing House, 2011.

[17] Ziada, S., Shine, S., Strouhal numbers of flow-excited acoustic resonance of closed side branches, *Journal of Fluids and Structures*, vol. 13, no. 1, pp. 127–142, Jan. 1999, doi: https://doi.org/10.1006/jfls.1998.0189 (accessed on 15 November 2025).

[18] Iwamoto, Y., Kondo, M., Minamiura, H., Tanaka, M., Yamano, H., Unsteady Flow Characteristics in a 90 Degree Elbow Affected by Developed, Undeveloped and Swirling Inflow Conditions, *Journal of Fluid Science and Technology*, vol. 7, no. 3, pp. 315–328, 2012, doi: https://doi.org/10.1299/jfst.7.315 (accessed on 15 November 2025).

[19] Chavan, V. U., Sewatkar, C. M., Effect of Reynolds Number, Strouhal Number and Amplitude of Oscillation on Pulsating Flow Characteristics Through a Circular Pipe, *Lecture notes in mechanical engineering*, pp. 353–362, Sep. 2016, doi: https://doi.org/10.1007/978-81-322-2743-4_34 (accessed on 15 November 2025).

[20] ASTM A106 grade b pipe | SA 106 gr b seamless pipe supplier, Dec. 15, 2023. Available online: https://www.marutimetals.com/astm-a106-grade-b-pipe/ (accessed on 15 November 2025).

[21] Moody, L. F., Friction Factors for Pipe Flow, *Journal of Fluids Engineering*, vol. 66, no. 8, pp. 671–678, Nov. 1944, doi: https://doi.org/10.1115/1.4018140 (adapted from the original drawing in: https://en.wikipedia.org/wiki/Moody_chart#/media/File:Moody_EN.svg).

[22] Chen, Z., Vortex Shedding in a Tee with a Closed Branch, *Becht*, Mar. 04, 2024. Available online: https://becht.com/becht-blog/entry/vortex-shedding-in-a-tee-with-a-closed-branch/ (accessed on 15 November 2025).

[23] Chen, S. S., *Flow-induced Vibration of Circular Cylindrical Structures*. Argonne National Laboratory, Argonne, Illinois, 1985. (adapted from the original drawing in: Mohammed A., Flow-sound Interaction Mechanism of a Single Spirally Finned Cylinder in Cross-Flow, DOI:10.13140/RG.2.2.12457.65123)

[24] American Petroleum Institute. *API 610 - Centrifugal Pumps for Petroleum, Petrochemical, and Natural Gas Industries*, Twelfth Edition, January 2021.

[25] American Petroleum Institute. *API 618 - Reciprocating Compressors for Petroleum, Chemical, and Gas Industry Services*, Sixth Edition, May 2024.