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FSI Simulation of Two Phase Flows in Flexible M-shaped Jumper to Predict the Failure

Manivel R[1], Babin T[2], Sangeetha N[3] and Brathikan V M[4]

1 Professor, 2, 4 UG students, and 3 Associate Professor
Department of Mechanical Engineering , Kumaraguru College of Technology, Coimbatore, India

ABSTRACT

Flow induced vibration (FIV) is a major design concern for the subsea oil and gas lines. Various types of pipes were deployed in the transport of these oil and gasses. In this the jumper is used to transport oil and gas from production tree to manifold. The jumper is classified based on their material behavior into flexible and rigid jumpers. The Flexible jumpers can withstand high static and dynamic loads due to internal pressure, temperature and external sea current effects. Internal two-phase flows with greater turbulence flowing through the rigid jumper pipes can interact with internal surfaces of pipe with greater pressure which would generate high amplitude vibration known as flow-induced vibration. This paper deals with modeling, transient analysis simulation of rigid jumper pipes .In the FSI analysis, two phase flows of oil and gas is considered to be flowing at different volume of fractions through a Rigid M-shaped Jumper pipe, which is fixed at both ends. Two phase flows of fluid domain at different volume of fractions is analyzed at constant velocity inlet and static pressure outlet conditions using ANSYS Fluent 16.2 software. The internal gauge pressure is transferred to structural part of jumper using FSI coupling method in Ansys Workbench 16.2. From the simulation results, it is shown that, the increase in velocity of the two phase flow results in high amplitude generating vibrations which leads to fatigue failure of the Jumper pipe.

KEY WORDS

Fluid Structure Interaction(FSI), Jumper Pipe, Flow Induced Vibration(FIV), Volume of Fraction(VOF), Two phase flow, Oil & Gas.

1.0 INTRODUCTION

The main phenomena that the oil and gas industries need to observe is the Flow Induced Vibration(FIV), one of the main cause for fatigue. The jumpers, which is deployed to connect the rigid risers from the vessel, are particularly used to safeguard the riser from, fatigue caused due to the Floating Production Storage and Off-loading (FPSO) motions. The main source of vibration in jumper pipes are Vortex Induced Vibration(VIV) and the Flow Induced Vibration (FIV). Parameshwaran [4] had reviewed the relation between FIV and FSI and its importance. In FIV, the resonance is sometimes due to slug impact frequency. These slug impact frequency should be considered for design of jumper. In topside facilities, nearly 21% of the failures are caused due to this hidden vibration issue [9].

By Tronconi [8], the slug frequency of two phase flow in horizontal pipes depends mainly on the geometrical characteristics of the inlet. Similar studies revels that, bends are the most vulnerable part, that has high risk of failure caused due to vibration [2]. API RP 1111 [1] addresses the failures caused due to the unsteady slug forces in oil and gas production lines. He also recommends to consider the hydro-dynamic loads and the FIV while designing the pipelines.

FIV can only be avoided by using flexible pipelines with high factor of safety. To reduce the deformation, which results in failure of the pipe, structural deformation or flow field modification can be employed. Shoei Sheng Chen[6] discussed about the possibilities to reduce these vibration in his book. These techniques can be deployed to suppress the vibration caused in the pipelines thereby, reducing the deformations caused during operation.

2.0 Objective

This paper presents a numerical methodology for solving the FIV on M-Shaped jumper pipe along with the methods used to reduce the deformations. A numerical stimulation in FSI was conducted on the transient conditions due to slugging two phase flow at the bends which are subjected to high amplitude of vibration. A dynamically driven methodology is discussed for reducing the deformations caused due to resonance.

The principle objectives of this paper are i) to investigate the Flow Induced Vibration on M-shaped Jumper pipe carrying two phase flow of Oil & Gas at 60 % vof. gas concentration; ii) to find the bends subjected to high amplitude vibrations; iii) to reduce the deformations acquired due to the resonance by changing the natural frequency of Jumper pipe in dynamic way. Calculations are made based on the recommendations given under API RP 1111[3], and safety considerations for Jumper pipe is taken from ASME B31.4 and ASME B 31.8 standards.

3.0 Computational Model & Boundary Conditions

Figure 1, 2 shows the jumper made up of API 5L X65 steel. The material properties and the geometric sizes are listed in table 1.

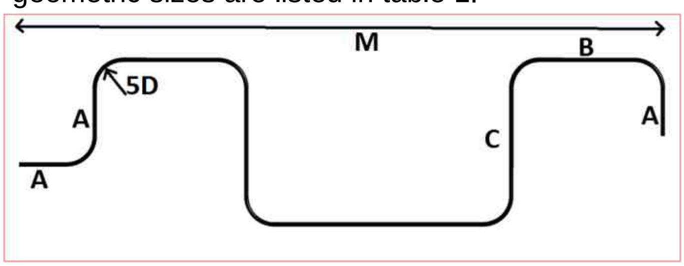

Figure 1. M-shaped Jumper Geometry

To approach this problem in a much realistic way, the boundary conditions at a depth of around 2000 meters where the operating hydrostatic pressure will be more than 200 bar is considered. According to Pontaza and Menon [6], the amplitude will be maximum when VOF of the gasses will be of 60% (0.6 upon 1). The secondary effects such as wave currents are not considered since the influence is much smaller than the FIV.

Table 1. Jumper dimensions and material properties

Jumper Parameter	Size
ID, m	0.27305
OD, m	0.20955
M, m	31.09
A, m	3.66
B, m	7.32
C, m	7.92

Table 2. Material Specifications

Material Properties	Quantity
Young's Modulus, N/mm^2	3×10^{11}
Poisson's ratio	0.303
Density, kg/m^3	7861.1
Yield Strength, N/mm^2	4.48159×10^8
Ultimate Tensile Strength, N/mm^2	5.30896×10^8
Elongation	18%

Table 3. Two Phase Fluid Properties

Fluids	Oil	Gas
Density (kg/m3)	835	272.264
Dynamic Viscosity (Pa-s)	0.0044	0.000015
Velocity (m/sec)	1	
Surface Tension (N/m)	0.031 (Oil - Gas)	

Structural boundary conditions include the support arrangements and the transferred force components. Jumper is clamped at both ends, i.e. fixed support. Hydrodynamic pressure is imported from CFD file to carry out structural analysis to find deformations due to high amplitude.

4.0 Design of jumper

To ensure the safety of jumper from internal fluid pressure of about 23 MPa, design and working pressure are evaluated based on API RP 1111[3] recommendations and safety considerations from ASME B31.4 and ASME B31.8 and shown in Figure.2. In this case, the internal fluid pressure is taken as 230 bar (23 MPa) for 2000 m depth. Hence, the design of jumper is safe with relevance to internal burst pressure.

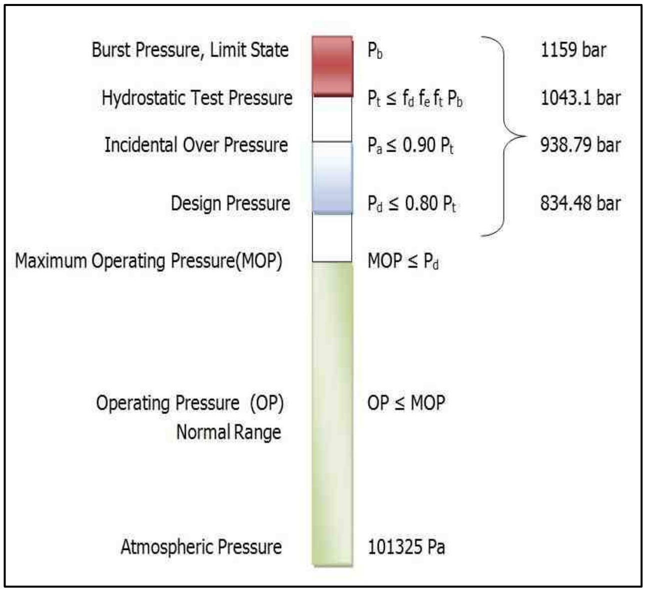

Figure 2. . Design and Working Pressure

The design calculations are made for checking the safety based on the load acting on the jumper and the corresponding stresses developed

Internal pressure Pi = 23 MPa;

For API 5L X65; σy = 450 MPa

Hoop stress design Factor F = 0.72 (ASME B31.8)

Stresses considered for safety checking:
σlong = σthermal + σAxial + σbending;

$\sigma_{thermal}$ = 26.325 MPa (α=6.5E-06 /°F; ΔT = 13.5°F)

σ_{Axial} = 37.9 MPa;

$\sigma_{bending}$ =2690.77 Pa; (x-axis bending)

σ_{long} = 64.23 MPa << 360 MPa (σy);

Therefore, the Jumper is in safe condition.

For Combined Stresses

Checking Criteria: $\sigma_{combined} \leq \sigma y \times 0.9$

From Von-Mises (Equivalent Stress theory)

$\sigma_{combined} = (\sigma_{hoop}^2 + \sigma_{long}^2 - (\sigma_{hoop} * \sigma_{long}) + 3\sigma_{torsion}^2)^{1/2}$

$\sigma_{torsion}$ = 0.11 Mpa;

$\sigma_{combined}$ = 88MPa < 405 MPa;

Based on the hydrodynamic loads and internal burst pressure, it is proven that the jumper chosen for the analysis is safe. But, 21% chance of failure occurs because of resonance due to flow induced vibration.VIV is attenuated by introducing splitter vanes around the jumper pipe.

5.0 Computational Approach

One-way Fluid Structure Interaction(FSI) coupling is performed for the entire geometry. The flow domain model is meshed with maximum skewness of 0.6066 and the average of 0.277 which includes 5,12,910 elements and 5,53,163 nodes as shown in Figure 3. A multi-phase volume of fluid model along with k-ε turbulence flow model is chosen because of its less computational time with optimum accuracy. The standard k-ε model is a two equation eddy viscosity turbulence model where, the eddy viscosity is computed based on the turbulence kinetic energy k, and the turbulence dissipation rate ε. A simulation with the time step of 0.01 seconds and a k-ε user relaxation factor of 0.2 is carried out to maintain the balance between accuracy and computational time.

The two-phase flow is initialized with bubble flow (60 % gas and 40 % oil) for better understanding of high amplitude zones due to bubble flow (high gas concentrations). As two-phase flow is assigned with initial velocity of 1 m/sec at inlet, the flow accelerates from 0.01 seconds, and the gas increases its velocity instantaneously at the bends due to difference in densities. Due to gravity, oil approaches towards the bottom portion of the cross section of pipe, and it tries to compress the bubble flow of gas against the wall, which in turn produces high circumferential stresses against the wall, especially near the bend. From the two-phase flow simulation, it reveals that the volume of fraction of gas is higher in bends, especially in 3rd and 4th bends as shown in Figure 4.

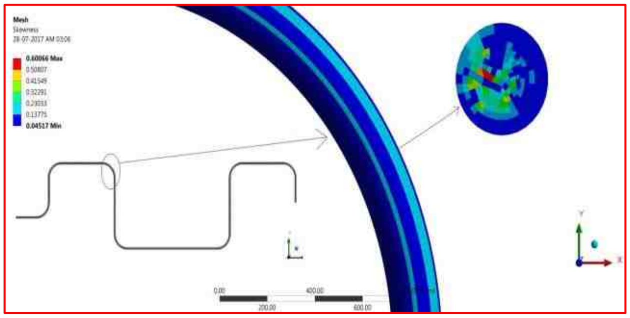

Figure 3. Meshed modal along with skewness

Irregular sized gas bubbles with large dimensions are occurring on the bend causing high slug frequency. This means that the existing duration of gas bubbles will be less due to the turbulence generated at the bend, and it continues till end of exit at the bend. . The slug frequency is found to be slug=1.219 Hz for the existing period of 0.82 seconds. These continuous and low frequency slug flow will generate high amplitude vibrations in jumper, especially at mid-span.

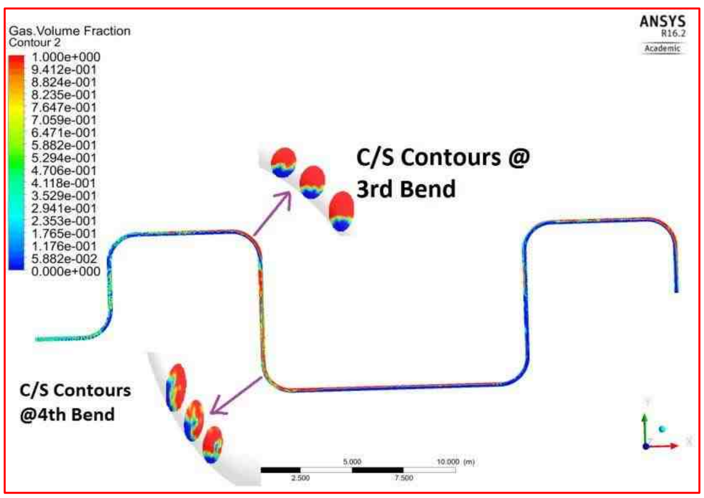

Figure 4. Gas Volume fraction contours at mid plane

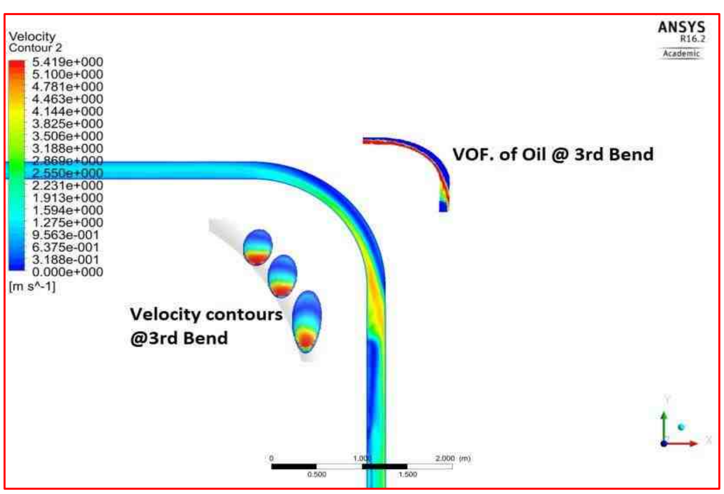

Figure 5. The oil velocity contours at mid-plane

For the assigned boundary conditions, the flow is stabilized at 60 seconds and the velocity contours confirms the increase in velocity of oil across the mid plane of jumper geometry is shown in Figure.5.

Figure 6. Gas VOF over mid plane

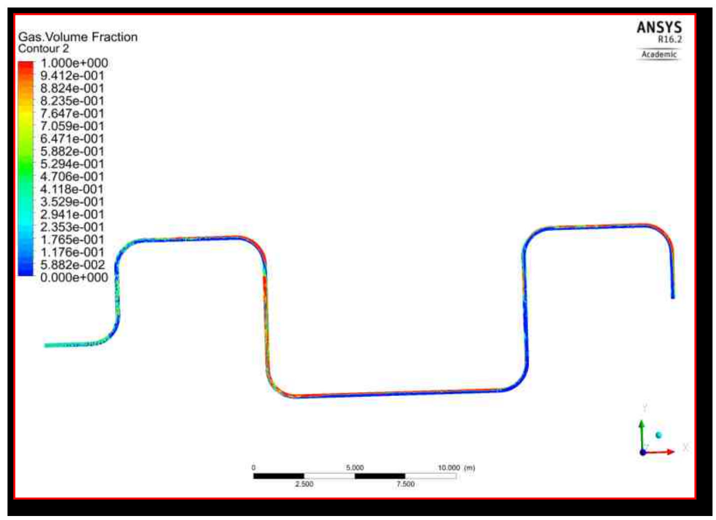

Figure 7. Oil VOF over mid plane

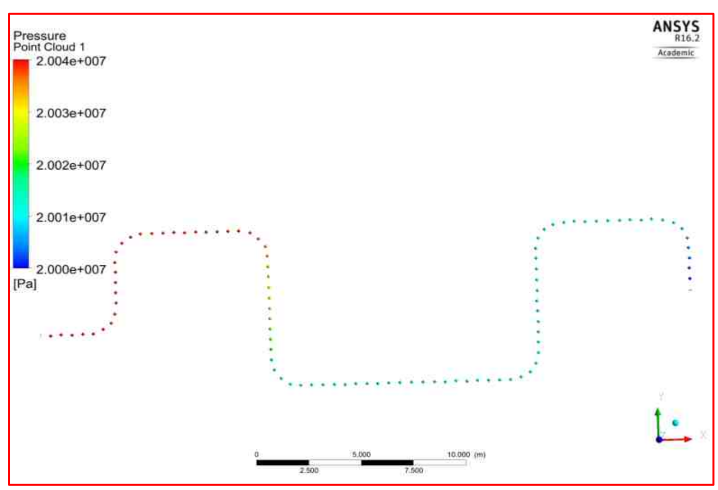

Figure 8. Pressure in 100 pts

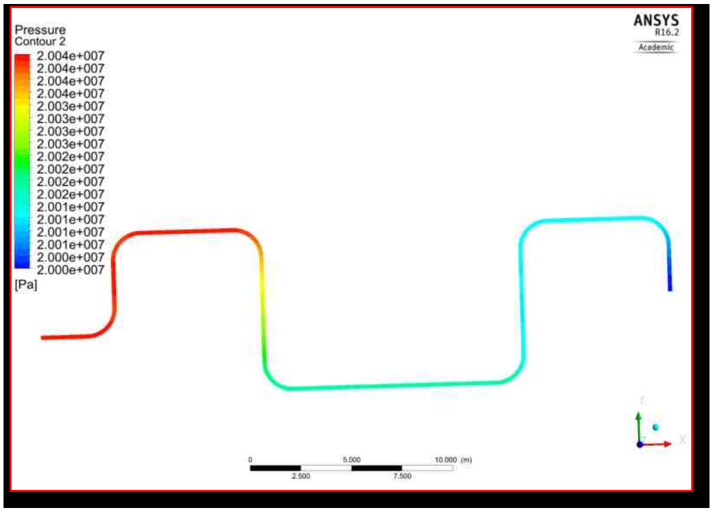

Figure 9. Pressure over mid plane

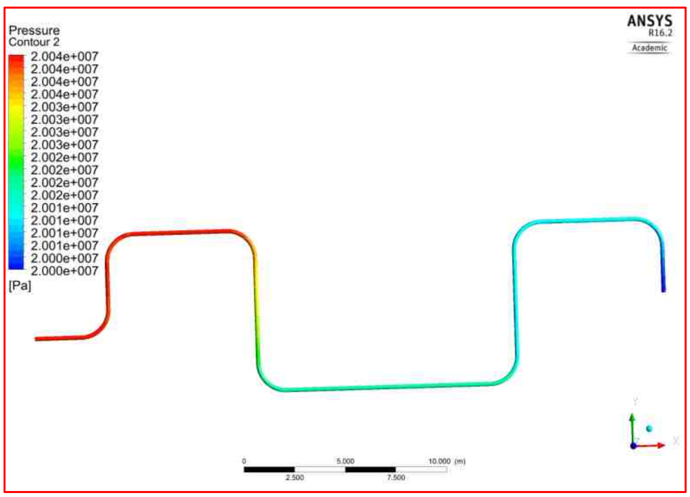

Figure 10. Pressure over the wall

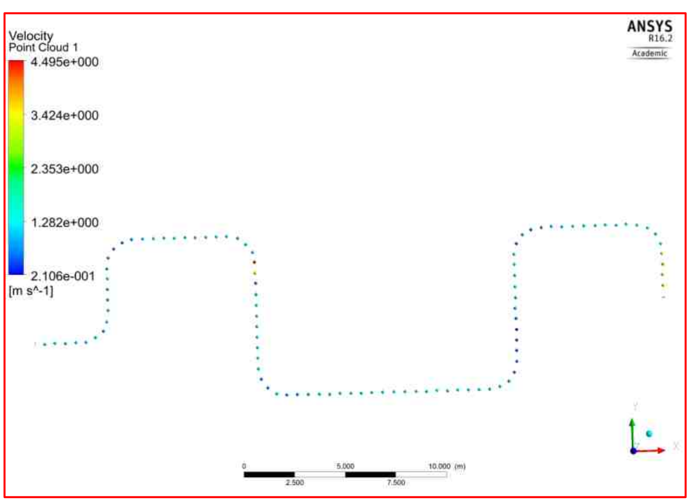

Figure 11. Velocity in 100 pts

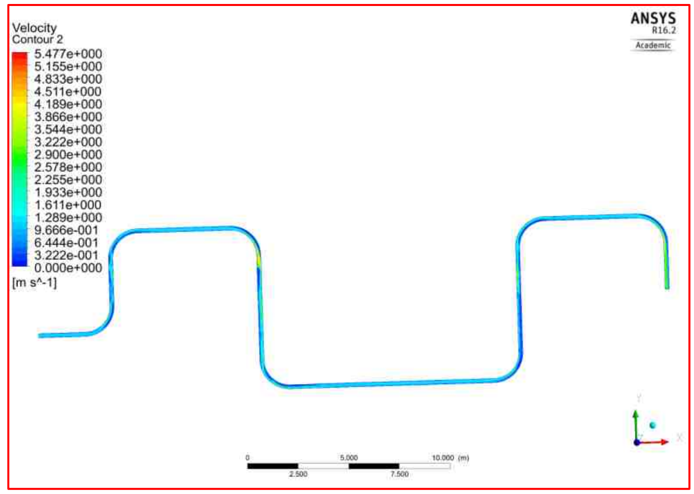

Figure 12. velocity over mid plane

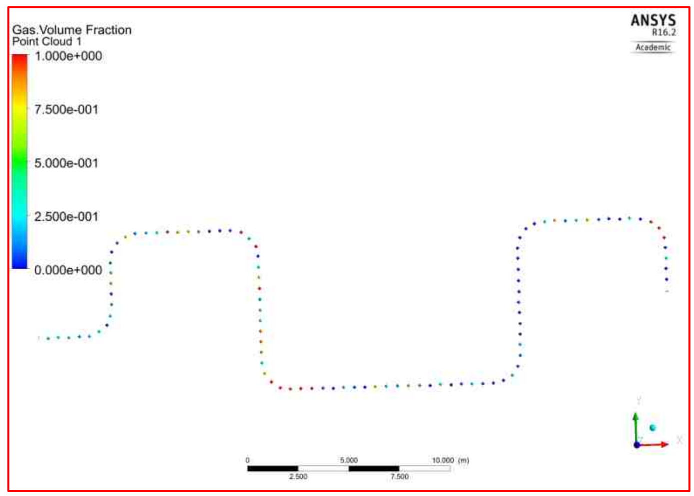

Figure 13. VOF of gas in 100 pts

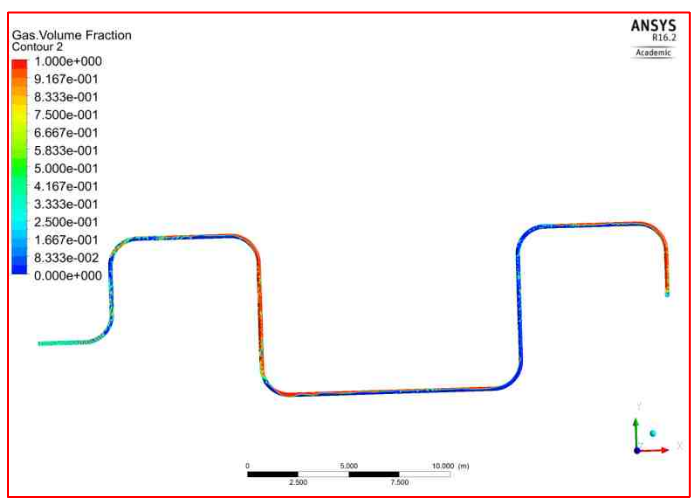

Figure 14. VOF of gas all over the wall

The above figures 6 to 14 showing the Volume of fraction, pressure and velocity over the mid-plane, over the wall and inside 100 points taken for further comparison.

6.0 Resonant Frequency

There is no mathematical approach to find the entire jumper natural frequency. However, study by J.C. Wachel [8] reveals that the frequency factors of first two modes for various configurations. The model is divided into five configurations and nine parts. To validate the simulation results of modal analysis, two parts of same configuration includes third and fourth bends are considered and the results are compared with calculated frequencies. The third and fourth bends were considered for as those bends are the locations where the maximum deflection will take place because of the pressure developed at those bends will be comparatively higher. First two modes of third and fourth bends from the FEA results are shown in Figures 15 to18.

In both parts, first one is L-bend in mode according to J.C. Wachel [8] study on natural frequency estimation for different piping configurations. Hence, it is verified that FEA analysis using ANSYS modal analysis shows better results. Hence, the natural frequency of entire M-shaped jumper is determined from ANSYS FEA analysis. The structural part of jumper is meshed using ANSYS containing 2,57,880 quad elements and 11,86,724 nodes and thickness of jumper is considered with five internal divisions. The fixed support is assigned at both ends as boundary conditions.

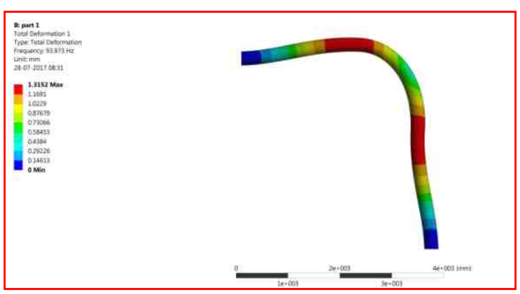

Figure 15. First mode of third bend

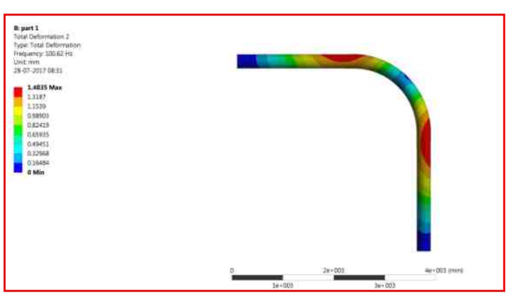

Figure 16. second mode of third bend

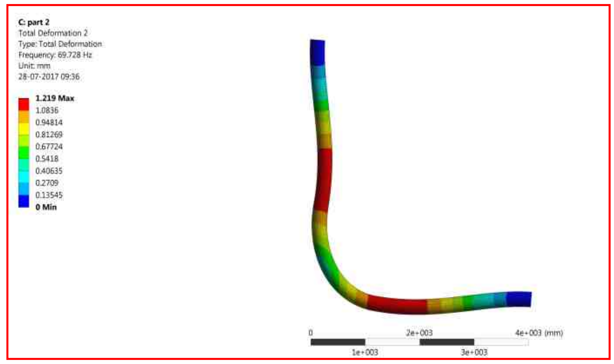

Figure 17. First mode of fourth bend

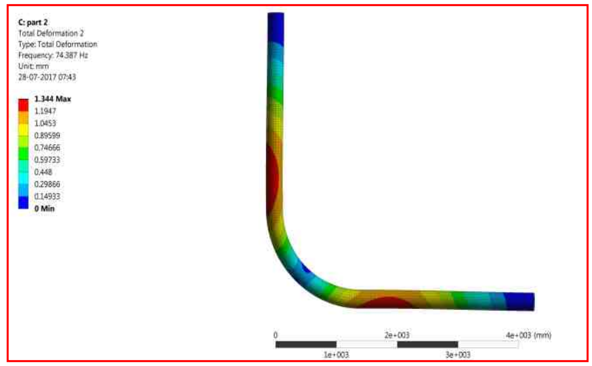

Figure 18. Second mode of fourth bend

7.0 Results and Discussions

FSI simulation shows that the excitation frequency due to two phase slug flow in M-shaped jumper is 1.219 Hz, which is in very close to second mode of natural frequency. The VOF of gas plays a major role in this part. After 60 seconds of flow stabilization, the VOF of gas across each bend is studied with respect to time. It reveals that the importance of slug frequency for each bends, especially in third and fourth bends. Similarly, velocity effect due to slug flow across each bend is studied and the variations are shown in plot (Figure 19).

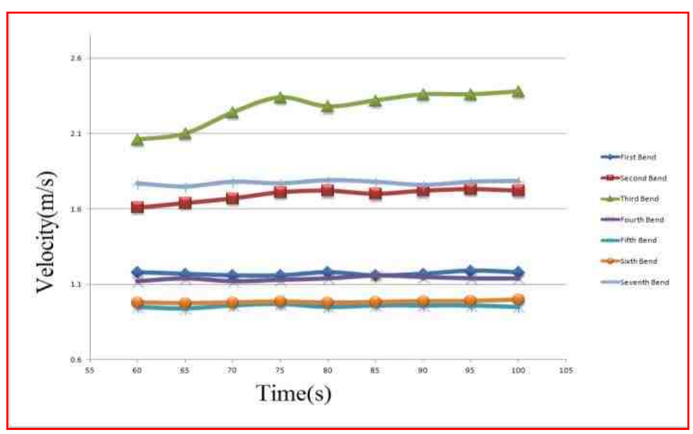

Figure 19. Velocity across each bend for the mixed flow

The variation shows that the VOF of gas across each bends remains more or less same with respect to time. This study reveals that the VOF of gas concentration is the major factor for failures due to FIV. Pressure drop occurs at each bend and net pressure drop of 40 kPa occurs over the entire M-shaped jumper. The pressure drop occurs due to the turbulent dissipation of fluid energy in terms of pressure. Since the third and fourth bend experiences high amplitude vibrations, the attenuation mechanism is needed in the bends.

6.0 Conclusions

In this paper, FSI simulation for FIV is analyzed for a jumper carrying two phase flows and the excitation frequency due to slug is carried out. After verifying the modal analysis results for known piping configurations, the natural frequency of entire M-shaped jumper pipe is evaluated and compared with excitation frequency. The locations of high amplitude zones due to slug flow are identified from FSI simulations and attenuation mechanism is suggested to avoid FIV resonant failures. Finally the method is verified from the modal analysis and jumper is ensured safety from FIV failures.

8.0 Reference

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