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ENGINEERING APPROACH FOR MEDIUM MODELING IN PIPING DYNAMIC ANALYSIS

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ABSTRACT

Two approaches to the problem of dynamic interaction between pipe and medium are compared in the given paper: 1) The first one treats medium as mass rigidly connected to the pipe finite-element model's nodes. 2) In the second one medium is modeled by the finite-element system of rod-elements. In this case the basic fluid-structure interaction (FSI) effects are taken into account.

The main techniques for FE modeling of some pipeline elements are presented in the paper. The second approach can be implemented by the use of general purpose FE programs. A model of a feed water pipeline of VVER-440 type NPP has been developed to study how the FSI affects on pipeline response. The results of the analysis which allow estimation of inaccuracy arising from medium dynamics neglecting are as follows:

- calculation of eigen frequency and mode shapes;
- seismic analysis using the response-spectrum method;
- accidental blast impact assessment with the use of time history analysis;
- operating vibration assessment on the basis of harmonic analysis.

It has become apparent that the way of medium modeling has an essential influence on the dynamic behavior of pipelines.

Keywords: fluid-structure interaction, pipe loads, vibration analysis, seismic analysis, pressure oscillation

1. INTRODUCTION

The medium mass is considered as non-structural mass (similar to thermal insulation mass) in most piping analyses which are carried out using the specific piping codes. The medium own dynamics therewith is ignored.

From the other hand there is a set of problems which can not be solved without taking into account the medium dynamics factor. Among them are the following:

- water-hammer problem;
- whip movements of a pipe after its rupture;
- flow-induced vibration of pipelines.

Two approaches may be applied when the medium dynamics problem can not be ignored:

- Uncoupled analysis that is performed in two steps. In the first step the hydrodynamic analysis is carried out with the assumption that a pipe is fixed and the interaction forces are calculated. Pipe oscillations caused by these forces are calculated in the second step.
- Coupled analysis in which hydrodynamic and mechanical tasks are supposed to be solved simultaneously. The solution of this task requires using the very specialized programs (Kratz, 2003). The latter approach known as Fluid-Structure Interaction is under active development nowadays.

In general FSI analysis is too complicated and expensive to be widely used.

A simplified method for FSI effects which allows carrying out such analyses using general purpose FE programs may be proposed for the limited set of tasks (Belytschko, 1986).

The main idea of the proposed approach is the following. "Steel" part of a pipeline can be modeled in the ordinary way – using the beam idealization. Medium inside pipe is modeled with rod-elements. These elements are able to simulate medium column axial oscillations (along the pipeline axis). Medium mass should be rigidly connected to the steel part of a pipeline in transverse direction. This approach shall limit, of course, a set of tasks to be solved. These limitations will be discussed below.

The possibility of changing a hydrodynamic model with a linear elastic one can be based on the fact that small harmonic oscillations obey the general differential equation:

$$\frac{\partial^2 \xi}{\partial x^2} - \frac{1}{c^2} \cdot \frac{\partial^2 \xi}{\partial t^2} = 0, \tag{1}$$

where x is a spatial coordinate, t – time, c – sonic velocity, ξ - harmonic wave function. The wave function can represent the behavior of velocity, pressure, displacement, force etc.

There is a set of formulas for evaluating the applicability of the FSI analysis to the pipeline under consideration. Following the terms of Standard Review Plan 3.7.2 (US NRC, 1989), a pipeline as a whole can be considered as "seismic system" and medium inside the pipeline – as "seismic subsystem". Two ratio - R_m and R_f – can be calculated therewith. R_m is a ratio of the "seismic subsystem" mass to the mass of "seismic system". R_f is a ratio of fundamental frequency of the supported system to dominant frequency of the support motion.

The following SPP criteria can be applied:

1) If $R_m < 0.01$, decoupling can be done for any R_f.

2) If $0.01 < R_m < 0.1$, decoupling can be done only if $R_f < 0.8$ or $R_f > 1.25$.

3) If $R_m > 0.1$, a subsystem model should be included in the primary system model.

4) If "seismic subsystem" is rigid compared to "seismic system" and also is rigidly connected to it, such subsystem may be accounted by including the subsystem mass.

It is obvious that the latter point is used in conventional analysis of pipelines, however in this case the assertion of rigid medium seems to be questionable. Beside this the connection between pipe and medium is rigid just in transverse direction; there is no connection along pipe axis.

R_m ratio can be calculated for a pipeline as:

$$R_m = \left[1 + \frac{\rho_s}{\rho_w} \cdot \left(\frac{D^2}{(D - 2 \cdot t)^2} - 1 \right) \right]^{-1}, \tag{2}$$

where D – pipe outer diameter,
t – pipe wall thickness,
ρ_s – pipe material density,
ρ_w – medium density.

R_m ratio calculation examples for some typical NPP pipes are given in Table 1.

Table 1. R_m ratio calculation examples

Medium	D, mm	t, mm	ρ_s, kg/m^3	ρ_w, kg/m^3	R_m
Water, p=9.0 Mpa	426	22	7800	1000	0.345
Water, p=0.29 Mpa	1220	10	7800	1000	0.792
Steam, p=6.8 Mpa	259	8	7800	33.5	0.0306
Steam, p=6.4 Mpa	426	24	7800	30.4	0.0142

As may be seen from the Table 1, the medium mass should be taken into account in case of water-filled pipelines, however it may be accounted just being added to the pipe mass provided that medium is rigidly connected to a pipe.

In case of steam-filled pipelines uncoupled analysis is permissible on condition that steam column eigen frequencies are spaced enough from dominant frequencies of the pipeline being excited.

These requirements are practically met if medium column fundamental frequency is above cut-off frequency for the dynamic excitation being considered.

For the simple pipes (without branches) the medium column eigen frequencies can be calculated by the following formulas:

$$f_n = \frac{n \cdot c'}{2 \cdot L} \qquad n = 1,2,3,... \tag{3}$$

$$f_n = \frac{n \cdot c'}{4 \cdot L} \qquad n = 1,3,5,... \quad , \tag{4}$$

where L – medium column height,
c' – medium sonic velocity with consideration for pipe wall elasticity.

Formula (3) is applicable in calculations of pipes with two closed ends as well as of pipes with two open ends, but in the second case fundamental frequency is equal to zero. Formula (4) can be applied for the open-closed systems.

c' value can be calculated by the Korteweg formula:

$$c' = \frac{c}{\sqrt{1 + \frac{K_w \cdot D}{E_s \cdot t}}} \quad , \tag{5}$$

where c – medium sonic velocity,
K_w – bulk modulus of medium elasticity,
E_s – Young's modulus of elasticity of the pipe material.

Medium sonic velocity is calculated by:

$$c = \sqrt{\frac{K_w}{\rho_w}} \tag{6}$$

Water sonic velocity weakly depends on pressure and temperature and in most cases it is taken as constant value. Steam sonic velocity should be defined using specific tables and formulas. For low-compressible media such as water it is advisable to correct sonic velocity by the formula (5), that is of most importance for pipes with relatively thin walls. Examples of sonic velocity calculations for some typical pipelines are presented in Table 2. Acoustic pipe length (L_{20}) calculated by formula (3) with f_1 = 20 Hz is also given in this Table.

Table 2. Sonic velocity calculation examples

Medium	p, MPa	T, °C	D, mm	t, mm	c, m/s	K_w, MPa	E_s, MPa	c', m/s	L_{20}, m
Water	9.0	180	426	22	1460	2010	$1.91*10^5$	1331	33.3
Water	0.29	20	1220	10	1464	2140	$2.00*10^5$	964	24.1
Steam	6.8	285	259	8	503	8.48	$1.82*10^5$	503	12.6
Steam	6.4	297	426	24	523	8.32	$1.81*10^5$	523	13.1

Abovementioned data show that FSI can affect the calculation results even in cases of relatively short pipelines.

2. GENERAL ASSUMPTIONS

The following assumptions have been made:

- steady-state flow in a pipeline is ignored. The steady-state velocity value is supposed to be much lower then the sonic velocity and has no influence on medium oscillations;

- the frequency range of interest is below the first natural shell mode frequency of a pipe. The latter can be obtained by FEM or calculated by simplified formulas. They can prove to be low enough for large diameter pipes with thin wall. Should this condition be met the beam model of a pipe remains correct and the medium mass can be connected to the beam node in transverse direction;

- Poisson coupling phenomenon is neglected;
- pipe-medium dynamic interaction occurs at such locations as bends, tees, reducers and closed ends where medium velocity or direction is changed;
- mutual friction coupling phenomenon is ignored;
- medium is considered to be homogenous. Multiphase media therewith can not be modeled;
- medium behavior is taken to be linear-elastic. Medium pressure oscillation is considered to be less than static pressure;
- thermodynamic effects due to pressure oscillation are ignored;
- dynamic displacements both of pipeline and medium are taken to be small in the FEM context, i.e. the task is still considered to be geometrically linear.

Damping at local hydraulic losses is not considered in this paper but can be easily implemented in the future.

Thus, only the basic FSI effects and linear approach are taken into account in the proposed method.

3. MODELING TECHNIQUES

As it was mentioned above, coupled system analysis assumes the pipeline model consisting of two parts. The beam part of the model represents the steel structure oscillations. The rod part of the model represents the acoustic medium oscillations inside pipe. The beam model can be created in ordinary way – using the beam idealization, the medium mass therewith is ignored. Consideration for the bend flexibility factor is an important point of modeling. Should general purpose FE code be used this problem can be solved by decreasing bending moments of inertia for relevant elements. Bends should be meshed into several straight pipes to provide the required accuracy of calculations.

The rod model creation is discussed more detailed below.

3.1 Straight Pipes

Medium is modeled by the rod using nominal density, reduced Young's modulus and cross section area calculated from the inner diameter. The reduced Young's modulus is obtained from the rod sonic velocity as:

$$c' = \sqrt{\frac{E}{\rho}} \quad => \quad E = (c')^2 \cdot \rho \tag{7}$$

The model detail in deformed state is shown in Fig.1.

Rod-element (green) has two own nodes, which are coincided in space with the correspondent nodes of beam-element (black). There are two additional nodes at each section, which spaced from central node toward local axes of the element.

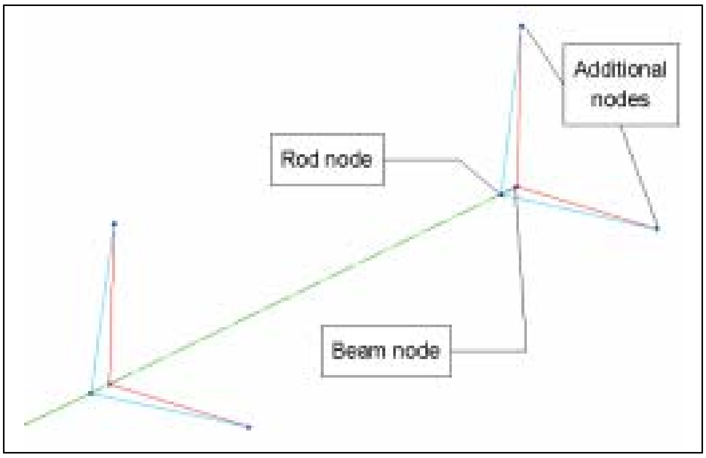

Fig. 1. Straight pipe model

The absolutely rigid element (red) has master node at a beam node and additional nodes as slave nodes. The node on a rod-element is connected to additional nodes by two spring-elements (blue), which have to be stiff enough. This combination of the elements provides free movement for rod-element along beam-element and no relative movements in transversal directions.

3.2 Bends

Bends are modeled similar to straight pipes. But we need to calculate free movement direction for each beam-element node. Two additional nodes have to be placed toward two orthogonal directions.

The model detail in deformed state is shown in Fig.2.

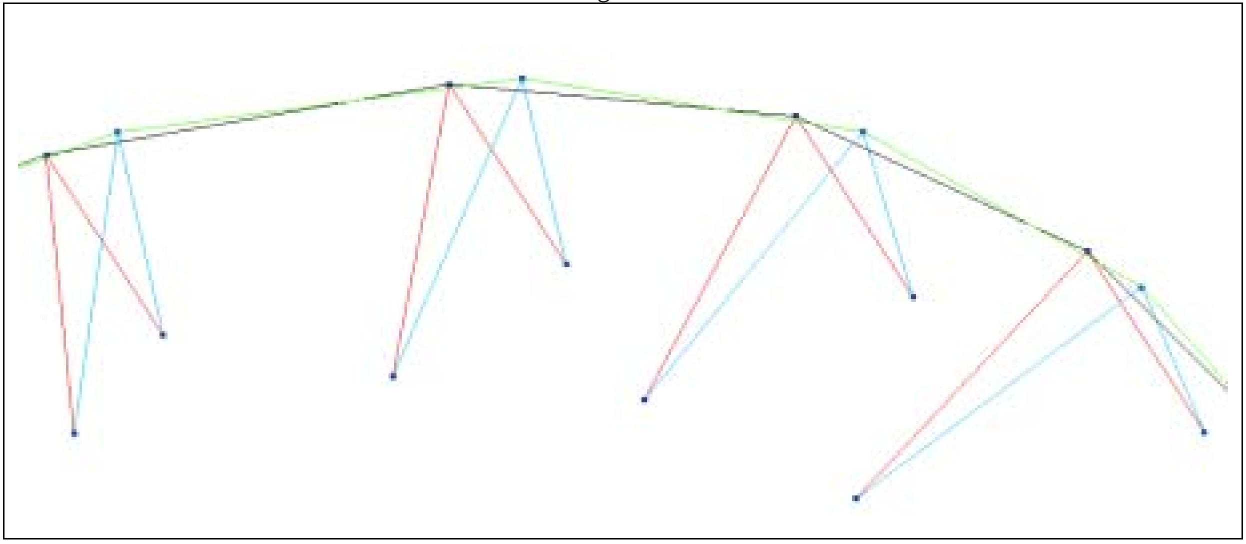

Fig. 2 Bend model

3.3 Reducers and Tees

The model for reducer is shown in fig. 3, and the model for tee is shown in fig. 4. Modeling is not so trivial these cases, because constant element volume condition has to be met. This condition can be expressed as:

$$\sum_i A_i \cdot x_i = 0 \quad , \tag{8}$$

where A_i – adjacent pipe cross section area,
x_i – nodal displacement toward the interior of reducer (tee).

In order to calculate geometry of the tee or reducer using formula (8) rod-elements have to be treated as rigid.

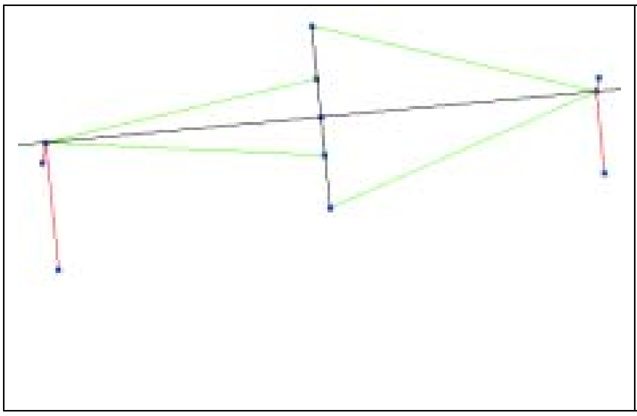

Fig. 3 Reducer model

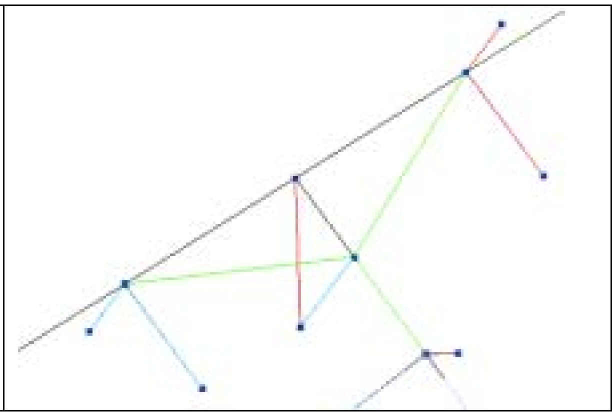

Fig. 4 Tee model

3.4 Free ends

Free ends of the pipeline may be modeled as open end or as closed end. Medium velocity is equal to zero for the closed end. It may be cap, closed valve or pump. In order to model closed end a correspondent beam and rod nodes have to be merged. Closed valve inside pipeline can be modeled in the same way.

Pressure is constant for open end. It may be nozzle of the vessel. There is nothing to do in order to model the open end.

If the number of the open ends in the model is more than one, there will be several zero-frequency eigen modes. These eigen modes will correspond to free medium movement from one open end to the other. The same effect occurs if model has the closed loops.

It is usual conventional practice to limit the pipeline model at rigid support, wall penetration etc. Proposed approach requires including full acoustical length of the pipeline to the model. Generally the coupled FSI model will be longer than conventional.

4 SAMPLE MODEL

The main goal of this investigation is to estimate the order of inaccuracy, which arises from neglecting of FSI. This estimation can be done by comparison of the computational results obtained from two different models of the same pipeline.

Feed water pipeline of the of VVER-440 type NPP was chosen to create these models. General model view is shown in Fig. 5. Some details are shown in Fig. 6, 7, 8 and 9.

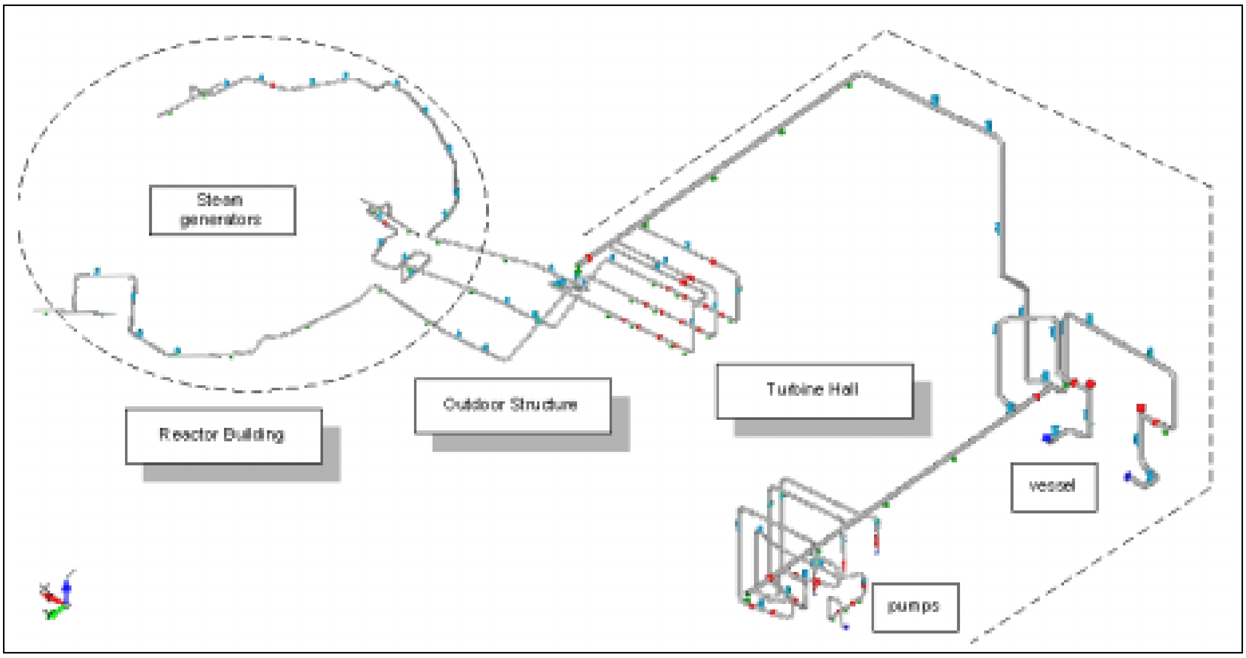

Fig. 5 Sample model overview

The pipeline starts from the four pumps in the turbine hall and ends inside steam generators in the reactor building containment. Some simplifications were applied. The model includes approximately a half of the real feed water pipeline. Some auxiliary pipes were excluded. Steam generators treated to be rigid. Some model parameters are below:

Main dimensions (X x Y x Z):	71.5 x 52.1 x 24.7 m
Medium:	Water
Pressure:	9.0 MPa
Temperature:	180 ‘C
Total length:	513 m
Pipe nomenclature (D x t):	426x22, 323x20, 273x15, 219x12 mm
Max. acoustic length:	214 m.

Estimated sonic velocity is 1331 m/s. The first estimated acoustic frequency is 3.1 Hz.

Source model was developed using special piping code dPIPE (Berkovski, 1997). This model was converted into two models for FEM analysis. FEMAP-Neutral file format is used to convert the model, so converted models are compatible with a number of the most popular FE codes. The two converted models will be referred below as Model A and Model B. Model A is made using proposed method for coupled FSI analysis. This model includes beam-element part as well as rod-element part to model the medium. Model B is made using conventional approach. This model includes only a beam-element part, the medium mass is rigidly attached to beam nodes.

Model B was successfully verified against source (dPIPE) model. Some parameters for both models are compared in Table 3.

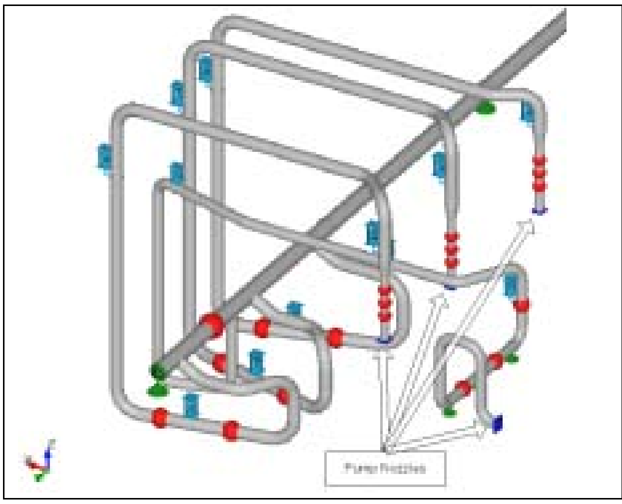

Fig. 6 Pipelines near feed water pumps

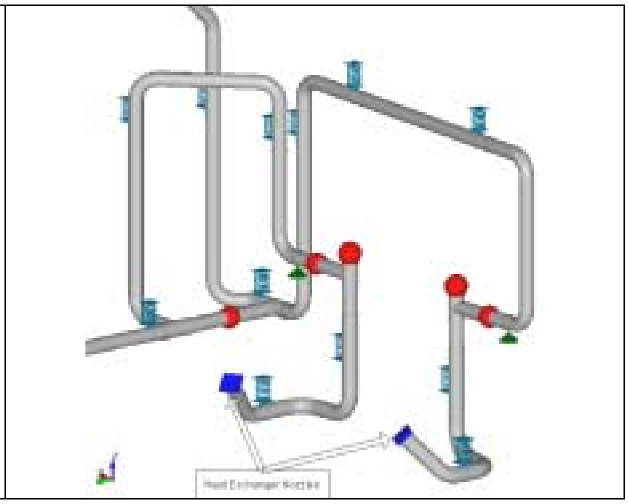

Fig. 7Pipelines near Heat Exchanger

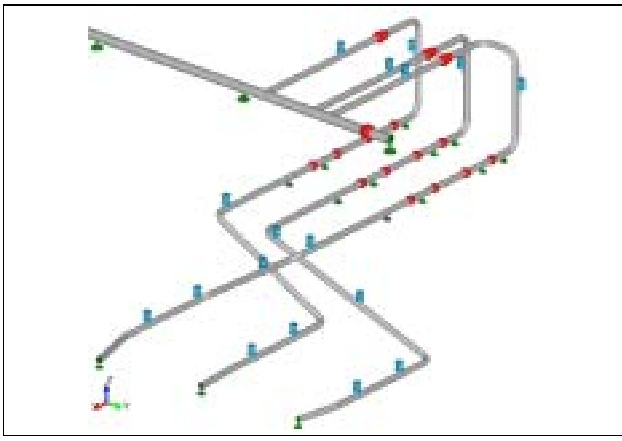

Fig. 8 Pipelines between Turbine Hall and Reactor Building

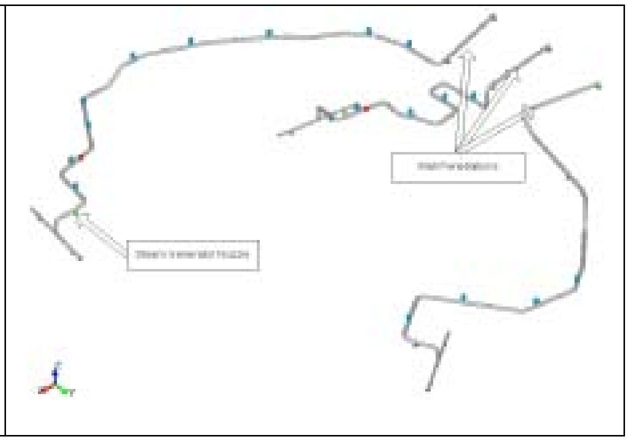

Fig. 9 Pipelines inside reactor building

Table 3. Parameters for the models

Parameter	Model A	Model B
Number of nodes	7557	2089
Number of elements	7574	2088
Number of DOFs	16461	10917
Overall mass, kg	154300	154350
Medium mass, kg (%)	35317 (22.9%)	-

There are no closed loops in the model. All free ends in the Model A are treated as closed ends to avoid zero frequencies.

Equal damping is used for pipes and for medium in the samples discussed below. However real damping for medium is usually less than for piping. This case FSI effects may become higher.

5 EIGEN FREQUENCIES AND MODES

Eigen mode analysis was made using SOLVIA code (SOLVIA, 2000) in the frequency range 0…30 Hz. First 20 eigen frequencies are compared in Table 4.

