NUMERICAL SIMULATION OF PIPING VIBRATIONS USING AN UPDATED FE MODEL

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ABSTRACT

Traditional design and condition monitoring of piping is mainly based on postulated events and on the application of allowable vibration levels. This approach gives only indirect information on the loading at the critical locations and generally leads to over conservative assessments. It is essential that developing piping failures can be anticipated and/or monitored and that any repair work is carefully planned ahead and carried out during regular outages. In an ongoing project a practical method is being developed to monitor the condition and remaining lifetime of process piping. This method combines both measurements - using a minimum number of fixed continuous measurements - and an adequate computational model.

Relatively simple piping in a NPP was chosen as the first pilot case. Measured modal shapes of the structure were excited using an impact hammer and a shaker. Results from experimental modal analysis were used in finite element (FE) model validation and updating process carried out using the FEMtools [1] code. This paper outlines the project and describes the main experiences and results of the model updating work.

NOMENCLATURE

EMA	Experimental modal analysis
$\{\psi_a\}$	Analytical mode shape vector
$\{\psi_e\}$	Measured mode shape vector
T	Superscript: Transpose of vector
H_{ij}	Response function, which expresses the response at DOF i if excitation is at DOF j
ω	Frequency
r	Subscript: mode number
N	Number of modes
ψ_{ir}	Eigenvector value at DOF i corresponding to the r^{th} mode
ψ_{jr}	Eigenvector value at DOF j corresponding to the r^{th} mode
ω_r	r^{th} natural frequency
ζ_r	Modal damping ratio for the r^{th} mode

1 INTRODUCTION

Condition monitoring and damage detection by means of monitoring modal parameters is based on the principle, that changes in modes are sensitive indicators of changes in the physical integrity of any mechanical structure [2]. Vibration testing offers an opportunity for different inspection techniques that may be able to detect structural failures and local structural damages, which can e.g. effect the stress fields of the structure.

Normally condition monitoring and damage detection, when done by monitoring modal parameters such as eigenfrequencies, mode shapes and damping ratio, has been based on comparison of results from experimental modal analysis of undamaged structure and damaged structure. These measurements can be made either with artificial excitation, e.g. with shakers or impact impulses or with ambient excitation in operational conditions [2 - 4].

Instead of using the modal properties of an undamaged structure as a reference baseline for comparison, modal properties of an updated FE model can also be used as a baseline. A verified, validated and usable mathematical model is the best knowledge base for the system under investigation [5]. In this work the aim was to create such an updated FE model which could later be used as a reference baseline and also to learn about appropriate modelling techniques and identify the difficulties concerning modelling of a pipeline and its components.

In the FE model updating phase a somewhat larger amount of measurements will probably be necessary than is possible in normal condition monitoring. Probably several iteration cycles are needed to come up to an adequately working FE model. This process is described in Fig. 1.

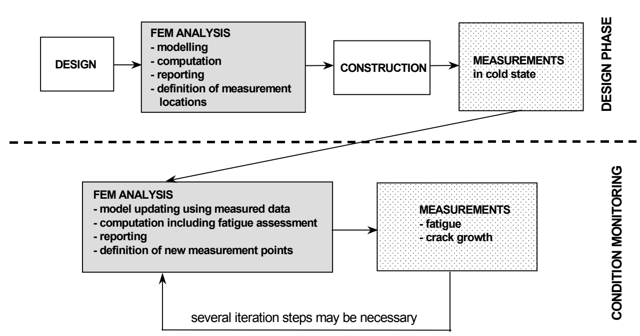

Figure 1. The approach to come to an adequate model to monitor the vibration behaviour of a piping system, starting from the piping design phase [6].

2 GENERAL DESCRIPTION OF THE STRUCTURE

The first pilot system, part of the auxiliary feed water system piping at the Olkiluoto NPP (OL1) was chosen based upon the following requirements:

- Reasonable in size,
- Cold in operation condition, no temperature effects nor insulation,
- Easy to access and measure in both operational and standstill condition (modal analysis),
- A clearly defined excitation (reciprocating pump).

During normal operation the auxiliary feed water system is not in use except for the periodically performed tests lasting for five minutes each month. The expected - and measured - vibration amplitudes were so small that no integrity problems are anticipated due to this vibration.

The part of the piping system being under consideration is located on the outside of the containment between the containment penetration and the auxiliary feed water system pumps. The pumps are 3 piston plunger pumps running at a frequency of 4 Hz. The length of the modelled part of the pipeline is about 56 meters including two major branches attached to it. The length of the measured part of the pipeline is about 44 meters. There are also 17 supports and three different actuators (valves or restrictors) in the measured part of the pipeline. Support locations and general description of the pipeline can be seen in Fig. 2.

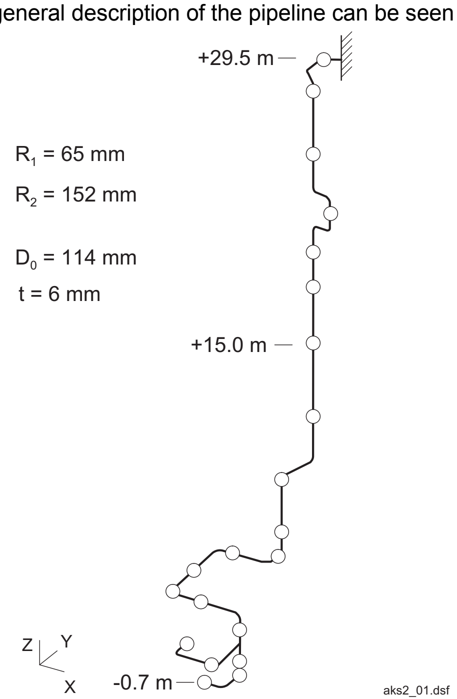

Figure 2. FE model and support locations.

This pipeline is made from DN 100 stainless steel pipe with nominal diameter of 114.2 mm and wall thickness of 6.02 mm. The design pressure, which is effective during the use of the pump, is 90 bar and the design temperature is 100 C. However, the piping is filled with water that does not exceed the room temperature during any anticipated transient. This means that the piping is not insulated and that temperature is not an issue.

There were basically four different types of piping supports, which were modified depending on their position and/or purpose. The purpose of supports was either to act as a support in all loading conditions or act as a support in case of a piping or neighbouring support failure. In latter case, the design drawings usually indicated a gap between pipe and support structure. During walk-down inspections it was found out, that visible gaps seldom existed between pipe and support structure.

3 MEASUREMENTS

Modal testing was done using both impact hammer [7] and shaker excitation [8], based upon 29 measurement locations and 76 measured DOFs. These measurement locations are shown in Fig. 3 along with the FE model. Modal testing was performed to experimentally characterise the dynamic behaviour of the piping. The mode shapes and associated frequencies were determined both during operation and in standstill condition. Thus, both operational and natural mode shapes were obtained.

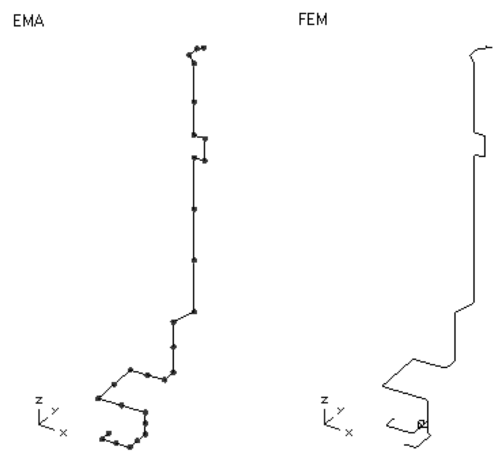

Figure 3. Measurement points (left) and the FE model (right).

3.1 Natural mode shapes

The natural mode shapes were excited, in case of impact test, with hammer impacts causing short time impulses with more or less uniform energy input over the significant frequency band [9]. The mode shapes themselves were then recorded but the data immediately after the impulses was neglected. The data, after the direct influence of the excitation has become negligible, were used to determine the modes and associated frequencies.

In case of shaker excitation a random noise signal was used to control the shaker output. Mode shapes were calculated with the Rational-Fraction-Polynomial-curve fitting method [8]. All measured FRFs were used together during this so-called global curve fitting.

Most significant (i.e. lowest) eigenfrequencies from both impact and shaker measurements are listed in Table1, excluding the lowest measured eigenfrequency (19.0 Hz) from impact tests due to measurement errors connected to that mode shape.

Table 1. The most significant eigenfrequencies from impact and shaker measurements.

Impact (Hz)	27.1	32.4	38.4	40.9	
Shaker (Hz)	27.2	32.7	39.2	40.8	42.3
Impact (Hz)	56.5		71.6	77.6	82.0
Shaker (Hz)	56.4	57.7	71.2		81.2

The mode shapes with hammer impact measurements and with the shaker excitations were quite close to each other and it seemed that in this case results were not dependent of the type of excitation (impact or shaker) used. It should also be noted that some of the deviation in the results could be explained by the fact that the water height in the piping was not necessarily the same during these tests.

4 FE MODELS

The FE models were originally generated with FPIPE [10] program and the models were translated into ABAQUS [11], which was then used as a solver. These analyses were conducted as a MSc. thesis [12].

The first FE model, referred here as Case 1, was modelled according to design drawings. This would also be the normal approach in the design phase and this model would also be used in first pretest analysis when the first measurements are planned. Of course, it was clear from the beginning that there are always differences between the actual structure and the ideal design drawing.

Properties of the FE model were varied in order to find out how different modifications affect the behavior of the model. Because it was known on beforehand that the critical aspect was to find suitable stiffness values for the piping supports, mainly their spring constants were modified. The stiffness values for the supports were estimated by using very simple FE models loaded by unit forces and moments. Also, more relevant information concerning the actual piping geometry was obtained by direct measurement and so-called walk down inspections made to the piping. This information was then applied in further analysis based on updated input data (Case 2 and Case 3).

The FE model used in the ABAQUS [11] analyses, main dimensions of the pipe line and support locations are shown in Fig. 2. The model consists of 180 elbow and pipe elements, 3 beam elements and 123 spring elements. Both pipe bends and adjacent straight segments are modeled with ELBOW31 elements and the bends were modeled with 2 elements. There are 5 integration points through the wall thickness and 20 integration points around the circumference of the section; six ovalization modes are used. The middle segments of long, straight pipe runs are modeled with PIPE31 type elements. To join the pipe segments modeled with different element types, warping of the ELBOW31 elements is prohibited at the nodes connecting the ELBOW31 elements to the PIPE31 elements. 1-dimensional spring elements are used in the appropriate directions to model the supports. Six SPRING1 elements with different stiffness values (one for each degree of freedom) are needed to describe one pipe support. One B31 element was needed to model an extension attached to the pipeline and two B31 elements were needed to model beam connecting two different pipe segments.

The locations and stiffness values of the supports as well as the pipe wall thickness values were modified according to inspection and measurements in Cases 2 and 3. In Cases 1 and 2 the pipe is assumed to be completely filled with water whereas in Case 3 the pipe is assumed to be filled only up to +15.00 m (see Fig. 2). This is done because, if the system has a height of more than 10 meters and the isolation valves at the top of the piping are perfectly tight, one may assume that there will be a vacuum in the upper part of the piping.

The material properties used in Cases 1 - 3 are listed in Table 7.2.2.

Table 2. Material properties.

Property	Case 1 - 3
Young's modulus	206 GPa
Steel density	7850 kg/m^3
Water density	1000 kg/m^3
Poisson's ratio	0.3
Temperature	20 °C

General descriptions of Cases 1- 3 are listed in Table 7.2.1.

Table 3. Analysed basic cases.

Property	Case 1	Case 2	Case 3
Supports	design documents	measured	measured
Supports	simple FE models	simple FE[1] models	simple FE[1] models
Gaps	low stiffness[1]	updated spring[2] stiffness	updated spring[2] stiffness
Wall thickness	nominal	measured	measured
Water level	full	full	level +15 m

1) Gaps in supports according to design documents are described using spring elements with low stiffness value.
2) Observed gaps in supports are described using spring elements with low stiffness value.

5 CORRELATION ANALYSIS BETWEEN ORIGINAL FE MODELS AND EXPERIMENTAL DATA

In order to validate the FE models, their correlation against experimental results needs to be evaluated and their quality must be reviewed numerically. Also, if results of correlation analysis are not satisfactory models must be modified and updated. Prior to any updating correct mode pairs must be identified, which can be a very problematic task.

The correlation was evaluated by comparing results from an impact hammer test [7] against the results from FE analysis. These analyses were conducted as a part of MSc. thesis [12].

As a first task the correlation between experimental and numerical results was evaluated in terms of modal assurance criterion (MAC) values and MAC matrices. In the beginning it was decided to filter terms with a value less than 15 % of maximum displacement from experimental eigenvectors and also to use 5 % double frequency tolerance. Filtering focuses the correlation analysis to areas where major modal displacements take place and double frequency tolerance enables combining frequencies within this tolerance limit. Correlation evaluations were mainly performed with the FEMtools [1] code. The following equation is used for evaluating MAC values:

$$MAC\left(\psi_a,\psi_e\right)=\frac{\left|\left(\{\psi_a\}^T\{\psi_e\}\right)\right|^2}{\left(\{\psi_a\}^T\{\psi_a\}\right)\left(\{\psi_e\}^T\{\psi_e\}\right)} \tag{1}$$

In general higher MAC value indicates better correlation between modes, although it is difficult to provide precise values that the MAC should take in order to guarantee good results. Ewins [13] has suggested following interpretation for the MAC values value less than 5 % indicates uncorrelated mode shapes and value higher than 90 % correlated mode shapes. Another estimate provided by Ingemansson Education [14] in their course material is that the MAC value below 50 % indicates poor correlation and values higher or equal as 70 % good correlation. This latter suggestion may also be reasonable in case of piping systems, where it may be difficult to define the actual measurement locations and directions accurately and where distances between measurement locations may be large.

Here the mode pair selection is based on visual inspection of mode shapes, MAC values and frequency errors. Usually the mode pairing is based on maximising MAC values and minimising frequency errors but in some cases this is not a feasible approach because it may cause mode pairing problems as described in reference [15]. Note, that experimental mode 1 is left out of all comparisons due to a measurement error in it's mode shape. Mode pairs for Cases 1 - 3 are presented in Tables 4 - 6.

Table 4. Mode pairs for Case 1.The average frequency error and MAC value are presented in the last line.

	Case 1		EMA			
Pair	Mode	[Hz]	Mode	[Hz]	Err (%)	MAC
1	18	25.15	2	27.13	-7.28	90.8
2	31	43.27	3	32.38	33.66	47.9
3	37	53.32	4	38.38	38.93	83.1
4	45	60.05	5	40.88	46.91	75.9
5	39	54.77	6	56.50	-3.07	41.1
6	43	57.76	7	71.63	-19.35	96.8
7	56	80.57	8	77.64	3.78	87.1
8	54	76.44	9	82.00	-6.79	53.9
Average					20.0	72.1

From Table 4 it can be seen that even if there are some acceptable mode pairs like pair 2 and 8 this model is not acceptable and it needs further refinement. This was not a surprise because the model used in Case 1 was based solely on design drawings, which were not always as accurate as hoped. During visual inspection of the pipeline and measurement of the support locations it was found out that the actual support locations differed sometimes significantly from locations suggested by design drawings. Also some of the supports have been altered from original design drawing. So it is extremely important, that design drawings used during the modelling phase are correct and up to date.

From the results presented in Tables 4 - 6 it is easy to conclude that both Case 2 and Case 3 have an improved situation over the original Case 1. Generally, results are better although in some mode pairs significant trade off has occurred between frequency errors and MAC values. Especially the largest frequency errors are reduced and the lowest MAC values improved while the highest MAC values have slightly deteriorated due to the trade off mentioned earlier.

In view of these results it is still somewhat unclear which one of the models, Case 2 or Case 3, would eventually provide the best possible base for further model updating. Also, it is impossible to determine with any certainty the actual water level in the piping from these results.

The selection to use Case 3 in model updating is based mainly on two factors:
1) knowledge that it is not likely that the piping is completely filled with water and
2) on engineering judgement based on marginally better correlation provided by Case 3.

Table 5. Mode pairs for Case 2.

	Case 2		EMA			
Pair	Mode	[Hz]	Mode	[Hz]	Err (%)	MAC
1	17	26.26	2	27.13	-3.17	90.4
2	25	37.95	3	32.38	17.21	32.1
3	29	43.05	4	38.38	12.17	56.8
4	31	44.48	5	40.88	8.81	83.5
5	38	54.92	6	56.5	-2.79	66.1
6	42	57.35	7	71.63	-19.93	94.5
7	44	61.66	8	77.64	-20.57	81.2
8	49	68.97	9	82	-15.89	72.1
Average					12.6	72.1

Table 6. Mode pairs for Case 3.

	Case 3		EMA			
Pair	Mode	[Hz]	Mode	[Hz]	Err (%)	MAC
1	15	25.21	2	27.13	-7.05	90.4
2	16	26.26	3	32.38	-18.88	57.5
3	28	43.68	4	38.38	13.82	56.8
4	30	44.48	5	40.88	8.81	83.5
5	36	54.65	6	56.5	-3.27	67.3
6	40	58.73	7	71.63	-18	94.5
7	60	94.25	8	77.64	21.4	84.3
8	56	86.14	9	82	5.05	71.9
Average					12.0	75.8

6 RESULTS FROM MODEL UPDATING

The Case 3 was selected as base model for the updating process and the main focus was concentrated to piping supports which were known as the most uncertain and ambiguous part of the pipeline. The updated version of Case 3 is referred as Case 4. All translational support spring constants were selected to be alterable parameters and experimental frequencies from 2 to 9 as well as corresponding modes were selected to act as responses. Later also the Young's modulus was also chosen as a parameter to be modified in order to improve the updating results.

So-called automated model updating was in this case impossible due to incorrect stiffness matrix, caused by use of the SPRING elements. This caused some serious difficulties to the updating procedure, which could be described as a loop, where certain steps were performed as follows:
(i) Importing of the ABAQUS [11] results into the FEMtools [1].
(ii) Performing correlation analysis and mode pairing in the FEMtools [1].
(iii) Sensitivity and updating analysis performed by FEMtools [1].
(iv) Re-editing of the original ABAQUS [11] input with the modifications suggested in previous step.
(v) Re-run of updated input in the ABAQUS [11].

Due to the incorrect stiffness matrix this loop was required to run several times and in order to avoid instability during the updating analysis the FEMtools [1] was allowed to make only small changes to the updating parameters.

Main changes caused by the updating analysis were in the stiffness of the translational spring supports, which were increased in some cases several hundred percent. Also the Young's modulus was increased in the lower (below level +15.00) part of the structure from 206 GPa to 210 GPa.

The mode pairing table of updated model Case 4 and experimental results based on 15 % filtering and 5 % double frequency tolerance can be seen Table 7. This table indicates improvement in the frequency correlation in general and also slight improvement in the lowest MAC values (mode pairs 2 and 3) over the situation with Case 3 (see Table 6). If mode pairs in Tables 6 and 7 are compared it can noticed that there has been some trade-off between frequency error and MAC values in pairs with high MAC values in Case 3 (Table 6).

Table 7. Mode pairs for Case 4.

	Case 4		EMA			
Pair	Mode	[Hz]	Mode	[Hz]	Err (%)	MAC
1	14	25.31	2	27.13	-6.7	88
2	19	30.11	3	32.38	-6.98	61.3
3	25	43.46	4	38.38	13.25	58.1
4	27	44.78	5	40.88	9.55	80.7
5	37	56.33	6	56.5	-0.31	54.8
6	48	74.9	7	71.63	4.57	88.8
7	54	83.72	8	77.64	7.83	86
8	52	80.59	9	82	-1.72	71.2
Average					6.4	73.6

6.1 Introducing damping

In order to evaluate possible effects caused by damping following procedure was performed:
(i) Analytical FRFs were synthesised from natural frequencies and corresponding mode shapes obtained from FE model used in Case 4 with the FEMtools [1] code. Here modal damping model with 1.5 % modal damping ratio was used and the FRF synthesis was performed according to following equation:

$$H_{ij}(\omega) = -\omega^2 \sum_{r=1}^{N} \frac{\psi_{ir}\psi_{jr}}{\omega_r^2 - \omega^2 + 2i\omega\,\omega_r\zeta_r}\,. \tag{2}$$

(ii) Resulting FRFs were imported into I-DEAS Test [16] software was used to perform modal analysis to the analytical FRFs obtained from previous step. This new model with damping is referred as Case 5.
(iii) Resulting natural frequencies and corresponding mode shapes were imported in to the FEMtools [1] for new correlation analysis.

The FRF synthesis was made by using three excitation co-ordinates (all three directions x, y and z were used) and by using all nodes of the FE modes as response co-ordinates. Modal analysis was performed with I-DEAS Test [16] by using so-called polyreference technique for extraction of the modal parameters (natural frequencies, damping and residue) and corresponding mode shapes were extracted with the frequency polyreference technique. Both techniques can be found summarised in I-DEAS Test [16]: Theory manual.

The resulting mode pairs for the new model, referred as Case 5, is presented in Table 8. Also here 15 % filtering was used and 1.85 % was used as a double frequency tolerance for mode pairs 1 - 5 and 4 % for mode pairs 5 - 8.

Table 8. Mode pairs for Case 5.

	Case 5		EMA			
Pair	Mode	[Hz]	Mode	[Hz]	Err (%)	MAC
1	11	24.71	2	27.13	-8.9	84.3
2	15	30.11	3	32.38	-6.98	74
3	19	44.6	4	38.38	16.23	58.4
4	21	46.16	5	40.88	12.92	90.1
5	26	56.33	6	56.5	-0.31	69.6
6	34	74.9	7	71.63	4.57	95.7
7	36	78.21	8	77.64	0.74	83.1
8	38	80.59	9	82	-1.72	74.2
Average					6.5	78.7

The frequency error is presented in Fig. 4 in terms of a 45° line where in the ideal situation all markers indicating mode pairs should lie on this line. These results indicate better frequency correlation for the updated Cases 4 and 5 than for the original Cases 1, 2 and 3. The Case 4 produces best results in mode pairs 2, 5, 6 and 8 along with Case 5. In mode pair 1 there is very little difference between different cases. In mode pairs 3 and 4 the best results is achieved with Cases 2, 4 and 5 . Cases 3 and 4 gives the best results for mode pair 4 and for mode pair 7 the best result is achieved with Case 5.

Both the actual frequency errors for all mode pairs in Cases 1 - 5 and the average frequency errors for individual cases are presented in Fig. 5. Also in Fig. 6 all the MAC values for mode pairs in Cases 1 - 5 as well as the average MAC values for individual cases are shown.

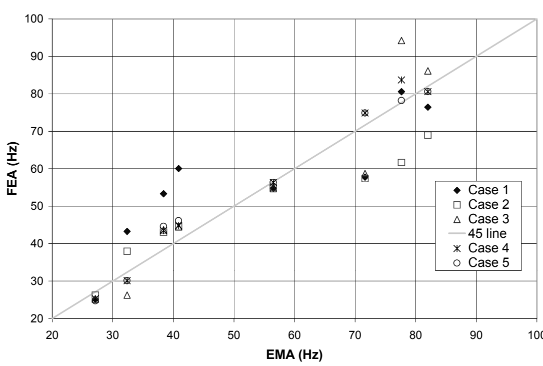

Figure 4. 45°-line comparison for eigenfrequencies in Cases 1 - 5.

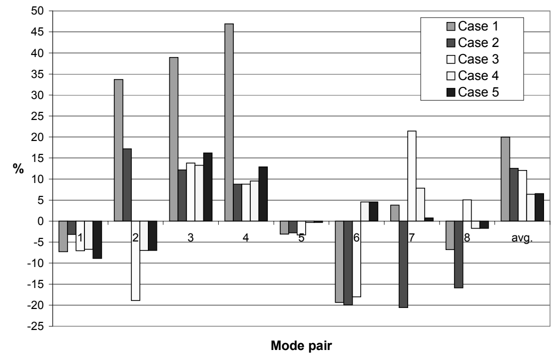

Figure 5. Frequency errors for mode pairs in Cases 1 - 5 and average errors in percentages.

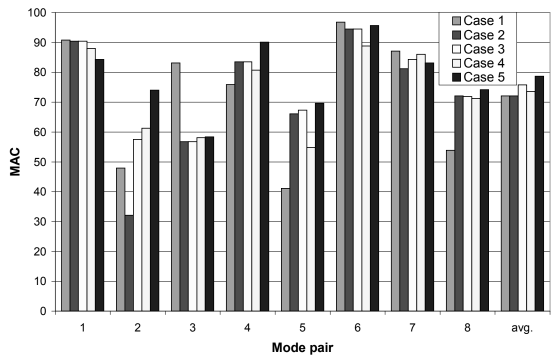

Figure 6. MAC values for mode pairs from Cases 1 - 5 and average MAC values.

Figures 5 and 6 as well as Table 7 confirm the improvement of the FEA results in Case 4 over the original Cases 1 - 3 especially in view of the frequency error. In case of MAC values the situation is not so clear due to the tradeoff mentioned earlier. In general updating of the FE model improved the MAC values in mode pairs with low original MAC values and in mode pairs with high original MAC values some loss of correlation did occur.

When the damping was introduced into the FE model in Case 5 also the MAC values improved as can be seen from Figures 5 and 6 and Table 8. On the other hand this caused some growth in the frequency errors, especially in mode pair 3. In general the average frequency error did not deteriorate significantly if compared with situation in Case 4.

7 CONCLUSIONS

It is fairly clear that the discrepancy, in case of piping with several supports, between FE models and real man made structures comes mainly from the uncertainties of the pipe supports. So from this point of view they are also the most suitable parts for modifications for updating a FE model. In order to enhance the possibilities of successful model updating some measurements should also be made from the supports and corresponding locations of the pipe. These measurements may reveal important information about the dynamic behaviour of the supports and about the interactions between the pipe and it's supports.

Special attention should be given to locating the actual measurement points accurately from the real structure.

Also it is important to ensure, as carefully as possible, that the actual measurement directions are correct. If the planned measurement direction is X then it is important that the measurement sensors are set in this direction. Both of these seemingly simple tasks can be extremely difficult in case of complex piping systems and some discrepancies and errors will always exist, which may cause serious and unexpected problems in correlation analysis and in later model updating.

During the FE modelling phase of a pipeline enough emphasis should be given to the boundary conditions, like supports or pipe-ends, and it should be remembered that a pipeline seldom ends with really rigid boundary conditions. When the FE model (geometry) ends, the effect of masses beyond this end point should also be taken into consideration if the pipeline is not rigidly anchored in this point.

Introduction of damping into the FE model had some improving effect to the modal correlation but these effect should be studied more carefully and with some other damping model like structural damping model before any conclusions can be drawn.

During the updating process it must be remembered that, although the updating is usually based on modifying some physically realisable properties such as Young's modulus, cross-section area, density, etc., there is no one-to-one correspondence between experimental and analytical models. In other words, the actual modelling errors are in fact compensated by adjusting design parameters selected for updating, rather than actually identifying and eliminating these modelling errors.

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