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DESIGNING STRUCTURAL DAMPING TO AVOID RESONANCE PROBLEMS IN STRUCTURES, PIPING AND SUBSEA EQUIPMENT: RISK REDUCTION AND FATIGUE LIFE IMPROVEMENT

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ABSTRACT

For weight optimised deep-water structures as well as subsea equipment and piping, designing for dynamic loading from e.g. ocean waves, rotating and pulsating equipment is a challenge. A special case is the acoustic vibrations experienced in the steel piping in either end of flexible risers. Excessive vibrations have been experienced both on topside and on subsea equipment. To the authors knowledge, fatigue failure in gas containing pipes has been the result in at least two known cases, one due to acoustic vibrations, and another caused by a traditional piston compressor.

During the design process it is generally a problem to predict the inherent level of damping in the structures or the equipment in order to estimate the response as accurately as possible. Much effort has been spent trying to predict the inherent damping. However, little has been done to deliberately increase structural damping in order to reduce the dynamic response significantly.

Controlled application of structural damping is an alternative to changing stiffness or inertia characteristics of the structure to avoid resonance, and is often the only solution for broadband loading where resonance cannot be avoided.

This paper describes solutions for two types of frequently occurring resonance problems on offshore installations and discusses general possibilities for the use of designed damping.

One solution, applicable for high frequency acoustically induced vibrations in piping, is successfully applied on a full-scale mock-up of a pipe segment with a blind flange and a flange with valve, representing two real world problem details. The applied damping solution is a tailored design.

Another example shows use of standard industrial dampers for vibration control of a piston compressor skid. The latter is implemented offshore and by visual control vibrations were significantly reduced. On site measurements will be conducted later.

The paper will cover design and construction of the actual vibration dampers including selection of damping material. Selection of damping material depends on the occurring frequency and temperature range. The dampers should be designed to obtain maximum damping effect given the stiffness, inertia, excitation and response amplitudes of the structure.

Avoiding resonance by designing natural frequencies away from excitation frequencies is sometimes close to impossible. Therefore, the deliberate addition of damping to substructures at which high stress is expected at high frequency and localized vibration is probable, can be a fatigue and risk reducing design measure that by far exceeds anything that can be achieved through other means.

INTRODUCTION

During design of structures subjected to dynamic loading at or near the structural natural frequencies, it is generally difficult to predict the structural damping in order to estimate the structural response as accurately as possible. Since material damping of steel, the most used structural material offshore, is extremely low, most offshore structures and components have very little inherent damping, and the damping is governed by uncontrollable factors other than material damping. "Guesstimates" of damping in the order of 1% of critical damping can easily be off by a factor of more than 2. Since resonant dynamic response for lightly damped structures scales almost linearly with damping, the response is off by the same factor.

To avoid resonance by controlling that the excitation frequencies are away from the structures natural frequency, one has to be able to predict both excitation frequencies and natural frequencies with good accuracy, and it has to be possible within reasonable means to design the structure or component so that its natural frequency stay clear from the excitation. Some times this is not the case.

In the design actions following a leak in a gas containing pipe, a lot of stiffening was applied, some helped, some had no effect, and some just created new resonances. Later, more detailed structural calculations were performed. The excitation was measured to have a possible frequency range from 50Hz to 600-700Hz, possibly even higher. The pipe system with all its details has natural frequencies closer than every 10 Hz in this range, making it in fact an impossible task to avoid resonance by stiffening or softening measures, or by adding inertia.

In the search for a better solution we designed and successfully applied structural damping to a full scale mockup of a pipe segment with a blind flange and a flange with valve, which was tested in the laboratory. This was achieved by the use of special visco-elastic damping materials.

In the exploration of damping and damping materials we also came across standard products for vibration damping that we have successfully applied to two other problems, one with a piston compressor, and one with vibrations from rotating machinery.

A search on the world wide web may indicate that fatigue failures in welded pipe details is a common and known problem, [1], [2].

In this paper we present the above mentioned examples and the advanced dynamic analyses and knowledge of materials that are necessary to successfully design structural damping.

We present why resonant vibration at high frequency so easily lead to fatigue failure in the short run due to the many number of cycles during short time, and in the long run due to the lack of an endurance limit for ultra high cycle fatigue.

We also discuss whether the application of structural damping should be a standard procedure in e.g. the design of support for a piping system.

CHALLENGES IN THE DESIGN AGAINST FATIGUE FROM HIGH FREQUENCY VIBRATIONS

When designing against fatigue failures in structures or components subjected to high frequency, 10Hz to 1000Hz or more, one faces additional challenges compared to e.g. design against ocean wave frequent response.

The number of cycles quickly rises to beyond the normal range of conventional SN-curves. As will be discussed, there are difficulties in predicting excitation amplitudes and frequencies, natural frequencies become harder to predict. The structures are often lightly damped in particular in local modes, and unless accurate measurements are made there are large uncertainties in the prediction of the inherent damping.

Loads acting on the model must be accurately or conservatively described for the computed response to be meaningful. For the simulation model to actually reduce design risk, it must capture relevant aspects of the problem. The use of conservative modeling principles are cumbersome for dynamic problems as conservative assumptions on e.g. stiffness that are useful for static problems, only lead to a downward shift in frequency that may or may not lead to an under-prediction of response.

Therefore, it is useful to ponder the question on modeling accuracy when deciding how to address a potential problem. A few thoughts based on past experience will be discussed below. The discussion is provided to show that it sometimes can be constructive to show healthy skepticism to numerical prediction results. Naïve trust in unrealistic modeling can actually be more harmful than not making any analysis at all, as poor analyses may replace rules of thumb and experience values that are better.

We also wish to highlight that, situations exists that even with the use of state-of-the-art theory and simulations, there are still design inaccuracies that only can be overcome with a robust design. One thing that adds robustness to resonance problems is the addition of structural damping.

<u>Ultra high cycle fatigue, SN-curves and allowable stress ranges</u>

It is disputable; see ref. [3], whether or not an endurance limit exists, and whether or not linear extrapolation of SN-curves beyond 10^9 cycles is valid. In the lack of better information, we extrapolate the SN-curves linearly to infinity without any cut-off for endurance limit.

Table 1 Allowable stress ranges for high cycle vibrations

Allowable stress range calculation. Assume continuous resonance for period of operation		SN-curve B	SN-curve D	SN-curve E
Duration [hours]	336	m=4	m=3	m=3
Max damage	0.05	A=1.02E+15	A=1.52E+12	A=1.02E+12
Freq [Hz]	Number of cycles during operation	Allowable stress range [MPa]	Allowable stress range [MPa]	Allowable stress range [MPa]
5	6.05E+06	53.9	23.2	20.4
10	1.21E+07	45.3	18.5	16.2
20	2.42E+07	38.1	14.6	12.8
50	6.05E+07	30.3	10.8	9.4
75	9.07E+07	27.4	9.4	8.3
100	1.21E+08	25.5	8.6	7.5
150	1.81E+08	23.0	7.5	6.6
200	2.42E+08	21.4	6.8	6.0
250	3.02E+08	20.3	6.3	5.5
300	3.63E+08	19.4	5.9	5.2
350	4.23E+08	18.6	5.6	4.9
400	4.84E+08	18.0	5.4	4.7
450	5.44E+08	17.5	5.2	4.5
500	6.05E+08	17.1	5.0	4.4
550	6.65E+08	16.7	4.9	4.2
600	7.26E+08	16.3	4.7	4.1

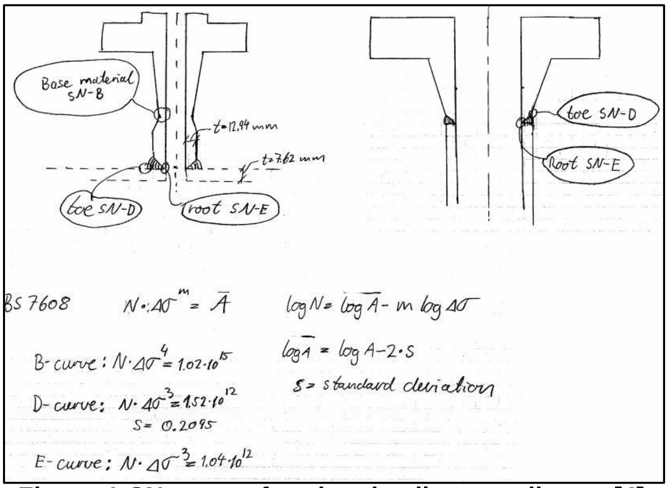

Figure 1 SN-curves for pipe details according to [4]. SN-curves used in table 1

Table 1 show the effect of resonance at high frequencies, the number of cycles grow enormous even for short periods of time. The table is set up as allowable stress ranges for continuous resonance for a 14 days period that would eat 5% of the fatigue life. At 100 Hz, there is 10^8 cycles during 14 days, the same as 20 years with typical ocean waves with zero crossing period of 6.3 seconds.

Fatigue tests of welded structures show a large scatter in the sense that for equal test pieces under equal loading, there is a large variation in number of cycles to failure. Therefore, operating experience one may have that a detail had sufficient fatigue life is no proof that the probability of failure is as low as desired.

If stresses exceed the allowable in the above tables, it does not mean that a fatigue failure is likely, but it means that the probability of failure is higher than the design codes require.

The allowable stresses are hotspot stresses including stress concentration factors. For e.g. ocean wave excitation, allowable stresses are more in the range of 30MPa to 75MPa, and 5-10MPa is very much lower than the static strength which is from 355 to 500 MPa for most piping and main load bearing structures. It shows how sensitive steel structures are to fatigue due to high frequency vibrations.

Accuracy in prediction of excitation amplitudes frequencies

Vibration from rotor unbalance excitation in rotating machinery has well determined excitation frequency and the unbalance force amplitude is clearly defined in standard design codes. However, the real unbalance force depends on many things, e.g. workmanship in the rotor balancing operation. Nevertheless, a good approach can be set up as design codes put conservative limits to unbalance loads and require the design to be robust with respect to limits for shaft- and bearing vibration.

However, a model of extreme detail is required to accurately analyze the dynamic response transmitted via the machinery and its supporting structure and further into the surround structure, e.g. for the purpose of determining whether vibration amplitudes may cause fatigue or be hazardous for workers.

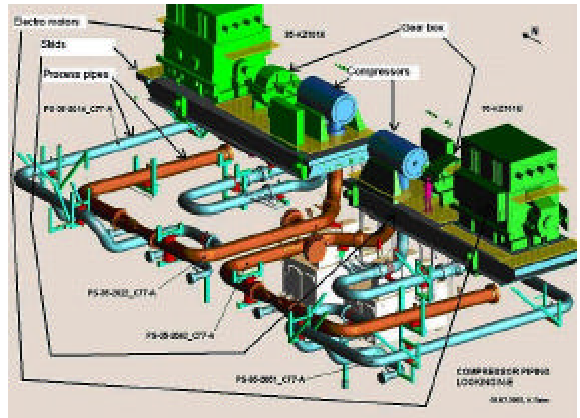

Figure 2 Vibrating generator skid with some of the surrounding structure

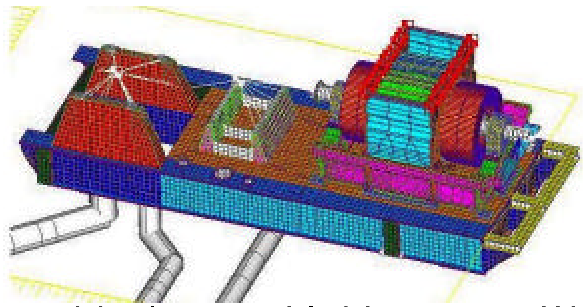

Figure 3 Finite element model of the generator skid in Fig. 2

Even with a detailed model as in Fig. 3., the vibration levels are uncertain, not the least because bearing properties for the rotor shaft can not be estimated with high accuracy.

Resultant forces from a piston compressor depends both on local pulses in the machine, and from reactions in the connected piping as the pressure pulse travels in the piping. Accurate predictions of excitation amplitudes are difficult even at the basic frequency. Reasons for this difficulty are that the medium being compressed may not be as perfectly homogeneous as assumed or because flow disturbances - or acoustic resonance- modulate the compression. Higher harmonic excitations resulting from pulses being not perfectly sinusoidal are even less accurate since the shape of the pulse and thereby the amplitudes of higher order Fourier components, is not known.

For both the above mentioned types of excitations, measurement of the response can be done, but for accuracy it has to be done on the completed system, i.e. when the machine is already built, and preferably also mounted in its operating environment. Because there may be a big difference in behavior when a machine is mounted on stiff fabrication floor, and when it is mounted on a typical offshore platform or module deck. The latter being rather soft dynamically in the frequency range 20 Hz –100Hz. Hence, there will still be uncertainties in the design phase.

Structural and acoustic vibration induced by flow is limited by the state of the art within theory and simulation tools. In fact, for many engineering cases neither source spectrum, nor source amplitudes can be predicted with great accuracy. Computational Fluid Dynamics analyses can be of help in estimating where and what type of acoustic sources there may be in a system. Source strength and spectrum must oftentimes resort to estimations based on experience and engineering judgment. Problem analysis can be narrowed once an acoustic source type and location has been identified. Response and natural frequencies can be calculated using weakly or fully coupled vibroacoustic analysis. However as long as the acoustic source strength and spectrum can not be estimated with accuracy, the response should be treated with healthy skepticism.

One limitation on source estimation is that excitation often involves mechanisms with a non-linear feedback from the actual response. Effects from flow, large displacements, pre-tensioning, environmental factors are more the rule than the exception for dynamic sources.

Accuracy in the prediction of natural frequencies and mode shapes

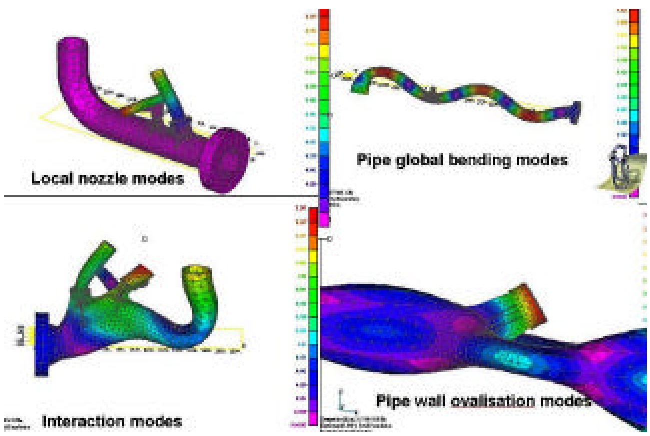

Figure 4, types of structural modes in a pipe system

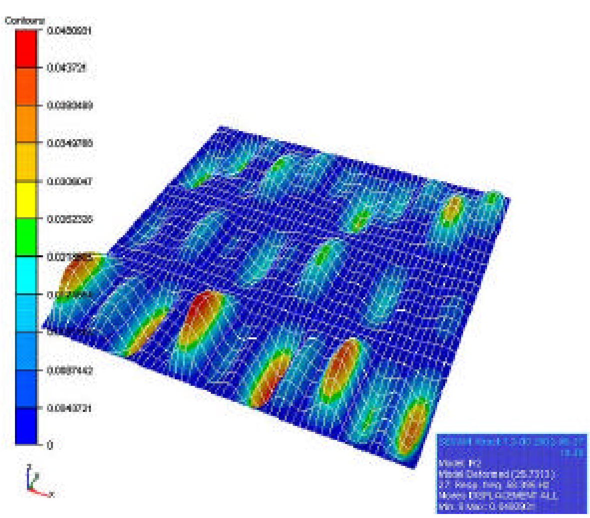

Figure 5, typical stiffened deck natural mode

Comparison with measurement indicates that for pipe modes like in Fig. 4, the accuracy of the frequencies can be 10-15%. For typical stiffened deck structures with nominally equally spaced stiffeners like in Fig. 5, there are many similar modes close in frequency. The real fabricated structure will have tolerances such that there will be a variation perhaps only 1 mm in stiffener spacing. But this is enough to cause real modes to be different from the modes predicted by numerical methods. And deck sections, fabricated to be equal, will in real behave differently for higher dynamic modes. There will be similar modes and frequencies in the analysis model and in the real structure, so one can with analyses estimate what type of response and if the excitation is known the response magnitudes, but exactly where in the deck and the exact shape of the resulting operating vibrations are hard to predict.

One rule of thumb often seen in the literature is to keep analyzed natural frequencies at least 15% away from driving frequencies. With the accuracy of predicting natural frequencies in mind, such rules have little value in avoiding resonances.

We have experienced that when finite element models, and hence the real structure, pass a certain complexity in terms of number of modes and how close modes are in frequency, mode shapes computed from different software differ more than they equal at high frequency. Thus, it is clearly motivated to question why a simulation model should be able to capture reality beyond a certain model complexity when state-of-the-art finite element software from two vendors shows significant difference in results.

Inherent damping level and accuracy in prediction of damping

Depending on the material quality steel tends to have material loss factor ranging between 0.02% and 0.005%. Built-up structures tend to have damping around 1%. Clearly, damping is governed by factors other than the material damping. Examples of damping mechanisms provided for free are transmission of vibration into connected structures, friction, air pumping in narrow gaps, and weak non-linear phenomena that add pseudo-damping. None of the governing damping mechanisms are actively considered or controlled in the structural design. Therefore, damping should be expected to vary with environmental conditions, i.e. the dynamic response should be expected to vary in amplitude from day to day.

"Guesstimates" of damping in the order of 1% of critical damping can easily be a factor of more than 2 too high or low. Dynamic response for structures with low modal overlap scales almost linearly with damping. The calculated response is therefore off by the factor the damping guesstimate is off.

Table 2 Measured damping in % of critical for four piping systems

Mode	System 1	System 2	System 3	System 4
1	2.93	2.88	1.90	0.53
2	1.44	1.35	1.96	0.96
3	2.91	1.31	1.15	1.05
4	1.95	1.96	1.82	0.61
5	1.90	1.00	2.22	0.89
6	0.76	1.18	0.96	1.55
7	1.89	0.50	0.98	
8	1.53	1.60	1.97	
9	1.92	2.56	0.87	
10	1.60		2.03	
11	1.19			
12	1.00			

Table 2 shows example loss factors obtained from experimental modal analysis of the fundamental modes of four different pipe systems.

Table 3 shows an example of the variation in loss factor for a pipe work mockup with changes in its boundary conditions, which clearly demonstrate that its dynamic response will be controlled by the amount of vibration energy that is transferred at its supports. Note that the loss factors provided in Table 2 and Table 3 apply for fundamental system modes and that values can be expected to drop down to the material damping (0.02%-0.005%) when modes with localized vibration shapes occur. A "guesstimated" damping of 1% can then lead to a dynamic response that is off by a factor 50 to 200 times.

Table 3 Variation in damping with support conditions

	Supported on steel beams		Supported on wood loading pallet	
Modes	**Freq. [Hz]**	**Damping % of Crit.**	**Freq. [Hz]**	**Damping % of Crit.**
Y-bending	*133*	*1.7*	-	-
X-bending	*171*	*0.6*	*178*	*0.16*
Y-bending	*219*	*1.0*	*225*	*1.0*
RZ-torsion	*358*	*0.4*	*360*	*0.02*
Z-pumping	*541*	*0.5*	*583*	*0.18*

STRUCTURAL IMPROVEMENTS TO FATIGUE LIFE FOR WELDED STRUCTURES

Earlier we have discussed the possibilities of stiffening, softening or adding inertia with its difficulties and limitations for some type of problems.

The fatigue resistance of welded structures are much less than for polished base material because of flaws, geometrical stress concentrations and residual stresses resulting from the weld process and the weld itself.

Fatigue life improvements to welds can be made. Weld sections can be increased [1], welds can be ground to remove flaws and improve geometric stress concentration, TIG dressing improves toe flaws of bad geometry of welds, and peening of welds help both geometrically and by changing from residual stress in tension to residual stress in compression.

Typical improvement in fatigue life for weld improvements is a factor 2 to 10. What do you do when such improvements are not sufficient? Given the above mentioned uncertainties in the response prediction, weld improvements should not be accounted for in conceptual design, it should be a measure to apply when problems are encountered.

Figure 6 Grinded and needled peened welds from improvements on vibrating gas pipe system

Also, extensive weld improvement for large portions of big structures is costly both in direct cost, and in time (schedule). Therefore structural damping can be an attractive alternative extensive structural re-design.

DESIGNED STRUCTURAL DAMPING

By designed structural damping it is meant a deliberate choice of a damping mechanism that is designed into the structure to increase damping beyond what is received by 'accident'. Damping is here defined as a process that removes energy from the system for each cycle, and not as response reduction in general.

The actual damping mechanism can be to include a standard product that is applicable to your problem, or to tailor make a damping application by the use of special damping materials with high loss factor.

In both cases dynamic structural analysis is required control the effect of the applied damping in the design process. In these analyses, one can not just assume a certain modal damping, or that damping generally is of a certain type or a level of damping. One has to physically model the damping one way or another. Then perform dynamic response analyses, in time domain or in frequency domain to see the effect on the response for the particular excitation.

Analyses and design of purposed made damping

Figure 7 shows loss factor and shear modules of visco elastic materials as function of temperature and frequency.

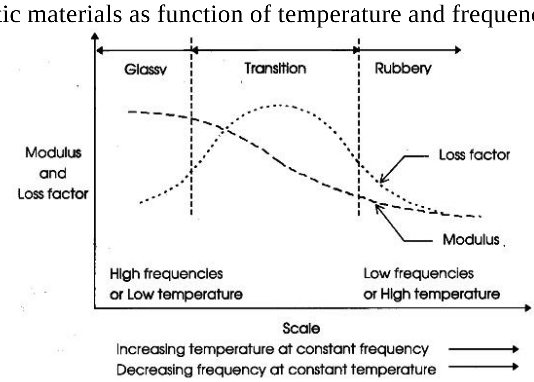

Figure 7 Properties of visco-elastic materials

One efficient way of using visco elastic material is to apply it as a structural shear element, either between plates as a damping over an area, or as a more concentrated link between locations of relative deformation.

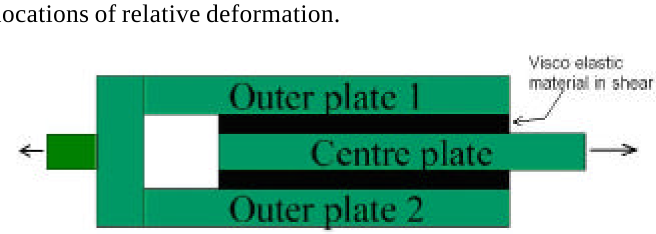

Figure 8 Visco-elastic material in shear deformation

To obtain high damping, the visco elastic material must be given the right level of shear strain, not too little, not too much, and the dynamic stiffness of the damper must be in correct proportion to the stiffness of the connecting structure. A requirement for effective damping of structures is that the damping material must be applied such that it takes up a significant portion of the dynamic strain energy. Hence the thickness and the size (area) must be adapted to the vibration magnitudes and dynamic stiffness of the structure in question.

With reference to Fig. 7, the material properties are also dependent on frequency and temperature. Careful material selection must be done for the problem at hand. To be able to fully control the damping in the analyses, the damping mechanism must be modeled physically.

Such tailor made damping can be designed for a wide range of amplitudes, frequencies and required damping. We have proved with a full scale mockup, se section below, that this type of damping can be made efficient up to 1000 Hz and with amplitudes less than 0.1mm.

Use of standard products

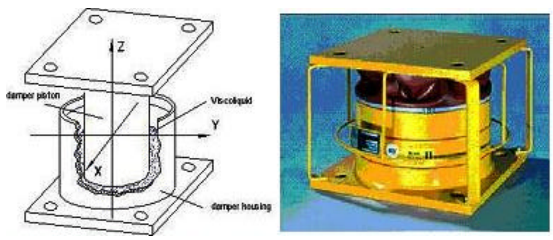

Figure 9 Standard pipe dampers

Figure 9 shows a standard product that works with relative deformation in six degrees of freedom between a piston and a housing filled with visco liquid. This product is efficient up to approximately 50 Hz with deformation amplitudes above approximately 0.1 mm. It comes in a number of sizes and with visco liquids that either is designed for a narrow temperature, or in less highly damped versions that are more temperature tolerant.

To use such a standard product efficiently, one must analyze the structure with the damping elements represented physically at the locations you design with. They may be included in a finite element model as frequency dependent dashpot elements. Again, complex eigen value analyses should be performed to find the effective modal damping, and dynamic response analyses must be performed to see the effect on the response to a certain excitation.

EXAMPLES OF APPLIED STRUCTURAL DAMPING

In this section we present three examples of application of designed structural damping to improve resonant vibrations.

Tailored damping design for pipe details subjected to broad band acoustic excitation

Two full scale mockups of a pipe with two details have been made. One of the details are identical to a real world detail that failed in fatigue and caused a gas leak on an offshore production platform. One of the mockups was equipped with tailored damping links made with visco elastic material. The other was kept un-damped for reference. The choice of damping material, the position of the links, the thickness of material, the size (area) of the shear layer, was all optimized using a finite element model.

Figure 10 Full scale mockup of piping detail with damping links (patent pending)

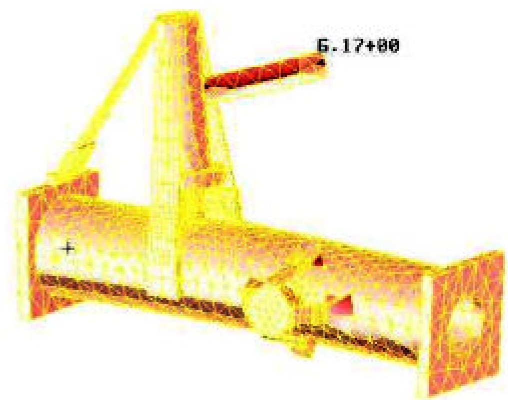

Figure 11 Finite element model of mockup

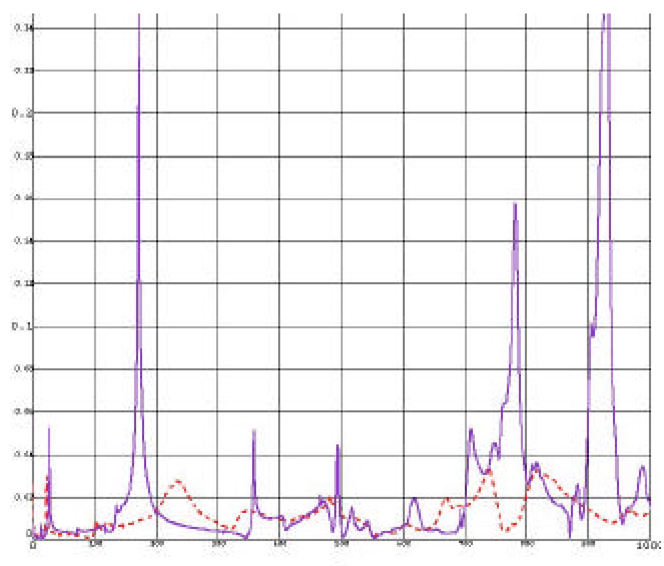

Figure 12 Damped and un-damped response

The mockups were instrumented and excited both with impact excitation and a shaker excitation. Figure 12 shows an example of response with and without the damping links. As can be seen, the resonance peaks are reduced with a factor of approximately 10. Modal damping ratios were measured. They increased from the values in Table 3, to from 10% to 40% of critical.

The mockup tests prove that it is feasible to make physical realizations of designed structural damping.

Figure 13 shows a design for cold mounting, i.e. no need for shut down in production of hydro carbons, that although not tested yet, we are confident that will work as remedy for vibration problems on existing plants

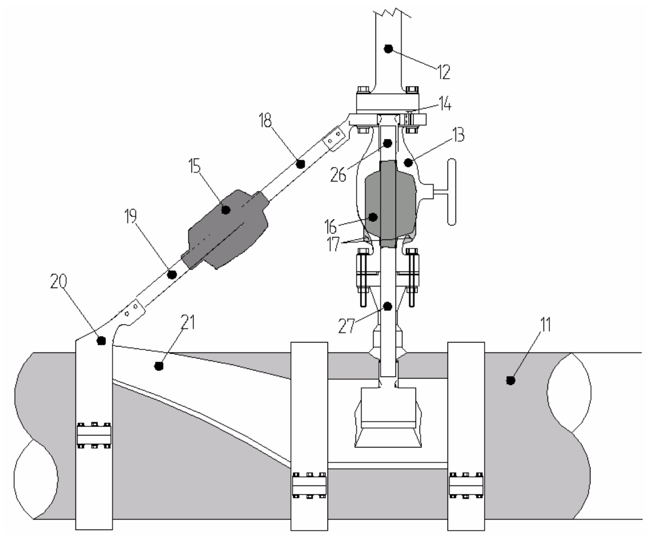

Figure 13 Design for cold mounting on existing piping

Use of standard products for damping of piston compressor skid

Figure 14 shows two of the local eigen-modes in a water gas compressor skid.

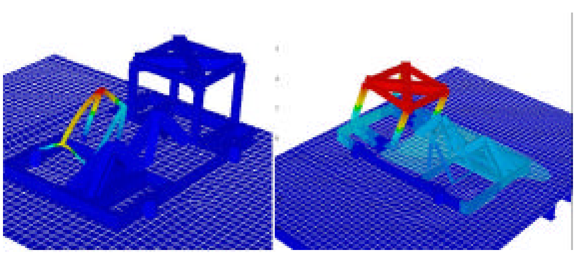

Figure 14 Local eigen-modes in a water gas compress skid

Excessive vibration due to resonance with the piston compressor led to a fatigue failure in a gas containing pipe. The skid was originally bolted directly to the platform deck. Measurements of the operating deflection shape were made. The problem was multi resonant and was judged hard or impractical to solve entirely by stiffening the skid and all of it components. One would also run the risk of just creating other resonances.

The problem was solved with a mixture of stiffening and mainly by using standard pipe work dampers mounted on the skid and supported on stiff locations on the platform deck. The locations of the dampers were chosen by trial and error using a finite element model of the skid and the platform deck in the vicinity of the skid

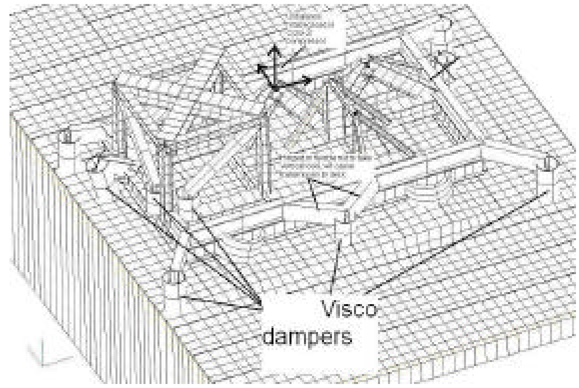

Figure 15 Arrangement of stiffening and visco dampers

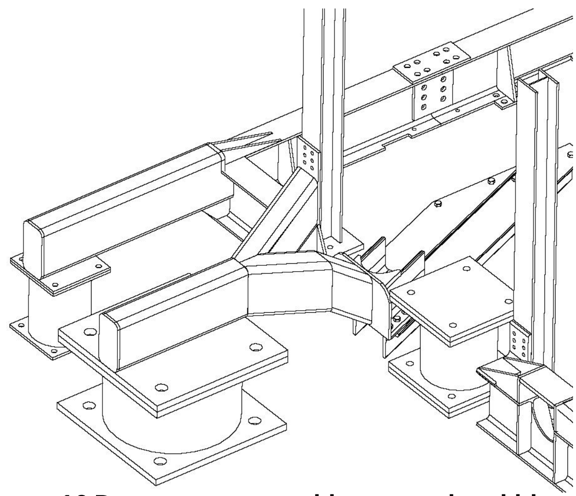

Figure 16 Dampers mounted between the skid and stiff locations in the deck, and internally on the skid

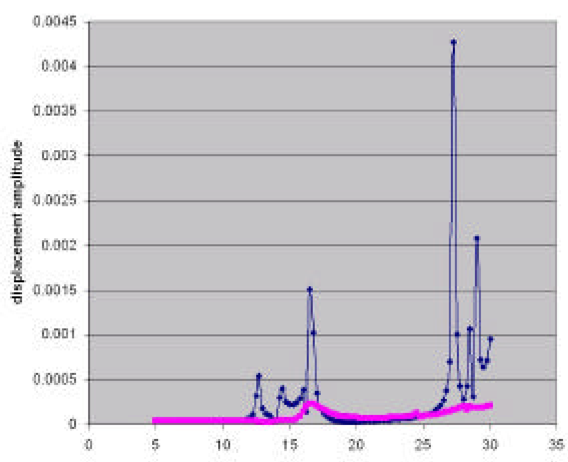

Figure 17, damped and un-damped frequency response of the skid.

This example shows that it is indeed feasible and not too difficult to apply standard damping products to typical offshore components that experience resonant vibrations. Damping provides a robust solution that is efficient for a range of frequencies and that a damped system is not so sensible to inaccuracies in inherent damping and structural finite element models. This approach adds reliability and decreases risks.

Use of standard products for damping of skid modes in resonance with rotor dynamics

The generator skid from figure 3 showed to have skid bending and torsional modes in the operating range of the compressor. Structural analyses showed that resonance with skid modes increased the rotor shaft vibrations of the engine. The shaft vibration has contributions from several modes, rotor modes, motor housing modes and skid modes. As a part solution, the skid modes were damped using a row of standard pipe work dampers along the main beams of skid, between the skid and platform deck. The dampers worked then in parallel with the traditional three point supports. The platform deck was stiffened locally at the damper support points.

USE OF DESIGNED STRUCTURAL DAMPING AS A STANDARD PROCEDURE

Standard pipe work dampers have been around for years. In fact the authors are surprised that they are not more widely used. Given the uncertainties in prediction of inherent damping, the low material damping of steel, the uncertainties in the prediction of dynamic excitation in pipe systems, adding structural damping seems like a natural thing to do. The dampers do not take static loads, they do not cause deformation loads due to e.g. thermal expansion, in fact they are ineffective unless there are vibrations or shock loads in the system, and hence there is no harm even if they were not strictly needed.

If only planned for early in design, like one design static supporting, the adding of structural damp ing is not costly.

The standard pipe work dampers will primarily damp only fundamental system modes, but will to some extent reduce local vibration. Because one of the excitation mechanisms for local vibration is that motion in the main pipe act as "base excitation" for the attached details.

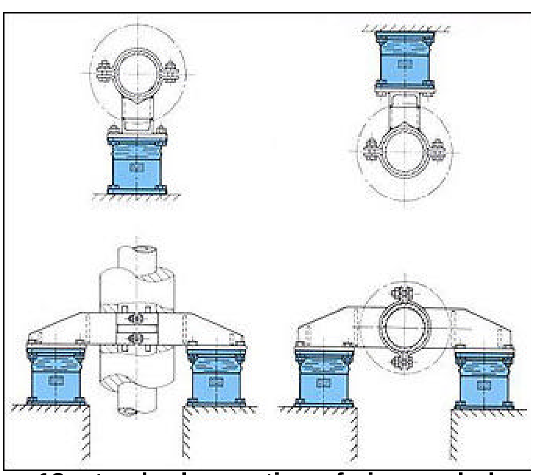

Figure 18, standard mounting of pipe work dampers.

Existing products known to the authors are efficient up approximately 50 Hz, this will cover most standard pulsating and rotation machinery, and we are confident that with product development it is possible to make standard products that will work also above 50 Hz.

CONCLUSIONS

Dynamic response at resonance is governed by the level of structural damping. Still structural damping seems to be the forgotten design factor.

Other improvements can help with a factor of 2 to 10, but if you need improvement more than one order of magnitude, structural damping is what you can do with resonant response to broad band excitation. Or sometimes adding a little damping may be cheaper and simpler than a completed redesign of the structure or component.

When your lightly damped dynamic problem passes a certain complexity, the uncertainties in prediction of excitation, natural frequencies and inherent damping, become so large that a robust design against resonance is required. The best way to obtain robustness against resonances is in many cases to apply more structural damping.

Avoiding resonance by designing natural frequencies away from excitation frequencies is sometimes close to impossible. Therefore, the deliberate addition of damping to substructures at which high stress is expected at high frequency and localized vibration is probable, can be a fatigue and risk reducing design measure that by far exceeds anything that can be achieved through other means.

We have shown both in theory and by the full scale experiment that is possible to increase structural damping of typical steel structures by a factor of more than 10, hence a stress reduction of the same factor. A stress reduction by a factor of 10 is an increase in fatigue life of 10^3, as fatigue damage scales with the stress range to the power of the slope of the SN-curve.

Subsea equipment may be tested upfront for vibrations from rotating and pulsating machinery, but can not be tested with realistic flow conditions for flow induced acoustic vibrations. It is a rather expensive form of prototype testing to install sub-sea equipment for gas production that is not robust to acoustic excitation.

To conclude, to add structural damping is indeed feasible for many type of applications, it adds robustness, it increases fatigue life, reduces uncertainties in response prediction and reduces risk for resonant vibrations.

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