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PIPING AND MACHINERY INTEGRITY ON STRUCTURALLY RESONANT PLATFORMS AND FPSOS

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ABSTRACT

Operators face significant integrity risks on offshore production facilities due to vibration of machinery and piping systems. These applications are more challenging than land-based systems because compressors, pumps, and other rotating machines are mounted on steel modules that can be structurally resonant and cause excessive vibration. Vibration problems cause fatigue failures in the piping system, machinery component failures, and operator safety issues.

This paper identifies best design practices to find and resolve structural vibration problems. The recommendations are based on input and guidance from various offshore operators. The paper will highlight the results from recent field investigations into structural vibration and will evaluate engineering methods used to address structural dynamic issues during the design phase.

NOMENCLATURE

rpm - = *revolutions per minute*
Hz = *Hertz (unit of frequency)*

AIV = Acoustic Induced Vibration
AVM = Anti Vibration Mount
DA3 = Design Approach 3 (618)
EPC = Engineering, Procurement and Construction
FEED = Front End Engineering and Design
FEA = Finite Element Analysis
FPSO = Floating Production Storage and Offloading
MNF = Mechanical Natural Frequency
ODS = Operating Deflection Shape
SSME = Space shuttle main engine

INTRODUCTION

Installing and operating machinery offshore has been done for decades and is now commonplace. With advances in engineering capability and understanding of the offshore environment, this will continue to be an area where significant time and effort is spent. Over several years of involvement in numerous vibration related problems in the offshore environment, it became apparent that engineering performed for onshore applications is not suitable for offshore applications when it comes to controlling vibration and increasing the long-term reliability and integrity of machinery and piping systems.

The present paper identifies engineering that can help increase machine integrity by applying improved structural dynamics modeling and an engineering process that ensures vibration is considered before it is too late. Three case studies are presented to highlight the limitations of a typical project design process, identify solutions, and demonstrate successful implementation of an integrating vibration approach.

FUNDAMENTALS OF VIBRATION DESIGN

Whether it be compressor, engine, piping, or even platform the fundamentals behind vibration are the same. When considering vibration you need to refer back to what is called the vibration equation.

$$Vibration = Dynamic\ Force \times Dynamic\ Flexibility$$

In order to reduce *Vibration* you need to either reduce the *Dynamic Force* or reduce the *Dynamic Flexibility*.

The *Dynamic Forces*, which are sometimes referred to as loads, vary greatly based on the type of system under consideration. Some common examples of these loads are pressure pulsations, rotational imbalance forces, moments and couples, surge and/or water hammer, amongst many others. Lots of engineering can be performed to reduce these forces; however, at some point it is no longer feasible to reduce them further. In the case of resonance, even very low forces might cause excessive vibration due to the very high dynamic flexibility. Think of the Tacoma Narrows Bridge. When you have no control of the force or can no longer reduce the force the only option for a reduction of vibration is to lower the dynamic flexibility.

When conceptualizing *Dynamic Flexibility* one must consider that this term relates to the response of the system. As such, changes in *Dynamic Flexibility* can be achieved by an increase or decrease in static flexibility, in mass, or damping. When dealing with a resonance issue, where the force can no longer be modified, the fundamental goal is to reduce the vibration which is achieved by lowering the *Dynamic Flexibility*. A common practice to lower the *Dynamic Flexibility* in a resonant situation for offshore applications is the addition of braces, clamps, more structural beams, and larger diameter pipe. An increase in mass by adding steel plates, concrete, or epoxy grout can also be considered to lower the vibration response. However, care must be taken as significant increases in mass can cause other engineering issues. Although not common, the addition of structural damping can be considered in some applications. Throughout this paper you will notice discussions on some forces that should be considered as well as the engineering approach taken to decrease the dynamic forces or the dynamic flexibilities to resolve actual vibration problems.

DESIGN FOR STATIC LOADS VERSUS DYNAMIC LOADS

The design of platforms and FPSOs for reciprocating and rotating machinery includes the consideration of several issues not required for their static design. Some of these issues are either hard to understand, counterintuitive, or both. The static design of platform beams or topside modules requires consideration of loads that are much greater than the dynamic loading that the machinery can create. That being said, due to the resonance phenomenon, dynamic loads can be greatly amplified and cause significant issues for the machinery and piping systems. This is due to the dynamic nature of these loads. This section will describe the dynamic loads to be considered and highlight how they can be taken into account for a successful design supporting dynamic loads.

Dynamic Loads

Engines, motors, and reciprocating or centrifugal compressors and pumps generate dynamic loads that are composed of several harmonics or frequencies. The time variation of a typical force generated by such a machine running at 1,000 rpm is shown in Figure 1. The load is cyclic with a period of 0.06 s. Figure 1 illustrates the load in the time domain. Although vibration levels and stresses could be calculated in the time domain, it is usually more convenient to work in the frequency domain. This is because the loads of interest generated by these machines can be decomposed into their harmonics. Calculating the magnitude and the phase angle of each harmonic allows representing the force in the frequency domain. Such a representation is shown in Figure 2. It is seen that a force can be composed of several harmonics that occur at the runspeed (1X) and at each order of the runspeed: 2X, 3X, etc. This means that for a machine running at 1200 rpm, the first harmonic (1X) of the generated loads will have a frequency of 20 Hz, the second harmonic will be at 40 Hz, etc.

For most loads, the magnitude of the components goes down as higher order harmonics are considered. This is also illustrated in Figure 2 as it can be seen that the magnitude of the sixth harmonic (6X) is much smaller than the magnitude of the fundamental harmonic (1X), for example. Typically, the main harmonics to be considered for unbalanced loads are 1X and 2X. Beyond that, the magnitude of subsequent harmonics decreases rapidly; therefore they do not need to be considered. For pulsation loads, several additional harmonics, typically up to 10X, must be considered. Engines will typically also present 0.5X loads that might need to be included during the design phase.

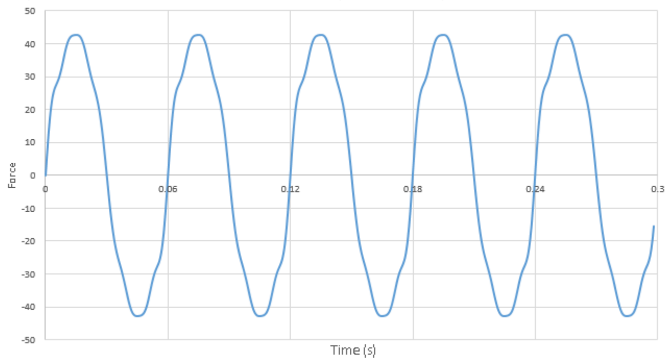

Figure 1: Typical Force in the Time Domain.

The previous discussion and Figure 1 and Figure 2 perfectly illustrate the case of a fixed speed machine. For a variable speed machine, the period of the time signal shown in Figure 1 will decrease and increase as the speed of the machine goes up and down. Similarly, for a variable speed machine, the spectrum shown in Figure 2 will shift to the right or to the left depending on whether the speed of the machine is increased or decreased. For such a machine, it is usual to see the spectrum as illustrated in Figure 3. The main difference between Figure 2 and Figure 3 is that in Figure 3, the frequency bands over which forces are produced get wider and wider as we consider higher harmonics. Depending on the speed range, harmonics might even overlap. For example, for a machine running from 600 to 900 rpm (10 to 15 Hz), the frequency content of the second harmonic (2X) will range from 20 to 30 Hz and the frequency content of the third harmonic will range from 30 to 45 Hz. This means that the loading for such a machine presents components whose frequencies vary from 20 to 45 Hz with no gap between the 2X harmonic when the machine is running at 900 rpm and the 3X harmonic when the machine is running at 600 rpm. Needless to say, the design of structural systems that can sustain such loads is more challenging than it is for a fixed speed machine.

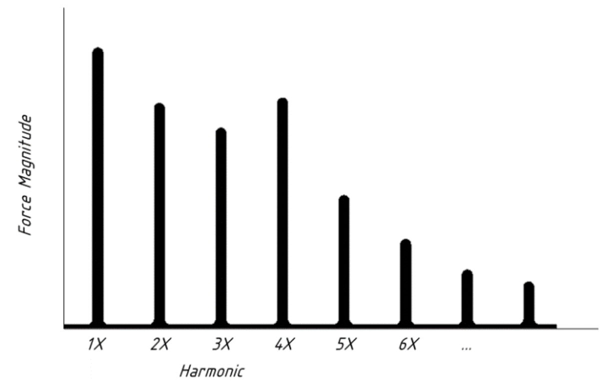

Figure 2: Typical force in the frequency domain.

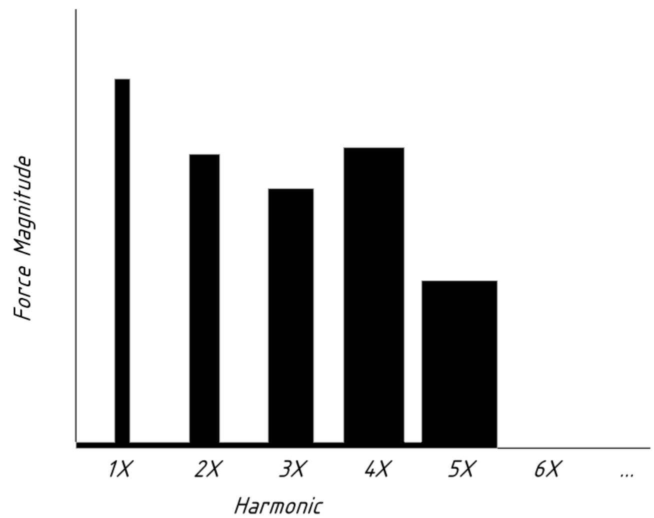

Figure 3: Typical Force in the Frequency Domain – Variable Speed Machine.

Effect of the Mechanical Natural Frequency

The previous discussion has illustrated the main characteristics of the applied loads, which is fundamental to an accurate prediction of the structural behavior of structures supporting a rotating or reciprocating machine. The next important component is the impact of such loads on a structure with multiple mechanical natural frequencies (MNFs). To simplify, we will first consider the case of a structure with a single MNF. The response of such a system is shown in Figure 4. On this plot, the vertical axis corresponds to the amplification of static effects, for example displacement or stress, that would be caused if the load was applied as a static load. The horizontal axis corresponds to the ratio of frequency of the applied load to the MNF. The amplification depends on the damping value, the curve shown in Figure 4 corresponds to a viscous damping value of 2% of critical damping. This is a typical amount of damping found in the structures considered in this paper.

Three zones can be identified in this plot. The first zone (left) corresponds to the case where the frequency of the force is small compared to the MNF. As can be seen, in this case, the frequency of the load is so small (compared to the natural frequency of the system) that its effects correspond to the static effects. In this case, it is common to evaluate these effects by running a static analysis, which is why these loads are often denoted quasi-static loads. It is generally accepted that this first zone covers values of excitation frequencies that go up to the natural frequency of the system divided by 2.4.

The zone to the right of the plot, Zone 3, corresponds to the case where the frequency of the force is 40% above the natural frequency of the system. In this case, the effects of such a load are smaller than the static effects of the same load as the curve is now below the value of 1 indicated on the vertical axis. As a result, such effects are generally not a concern. One will however notice that to reach Zone 3, the system will go through resonance, the condition for which the excitation frequency corresponds to the MNF, every time the machine is started or shut down. Since the frequency of excitation is usually ramped up or down fairly quickly, operating in Zone 3 is typically not a concern.

This paper is mainly concerned with the zone in the middle of the plot, Zone 2, for which the frequency of excitation varies between 40% and 140% of the natural frequency of the system. As shown in this plot, such a condition will result in significant amplification of the effects of such a load compared to its static application. For a damping ratio of 2%, the maximum amplification corresponds to 25 times the static effects. This means that at resonance, the system will experience displacement or stress that corresponds to 25 times the effects that this load would cause if applied in a static manner. This amplification and its avoidance are the main reasons why, although smaller than static loads, these dynamics loads must still be considered to ensure a safe and reliable design.

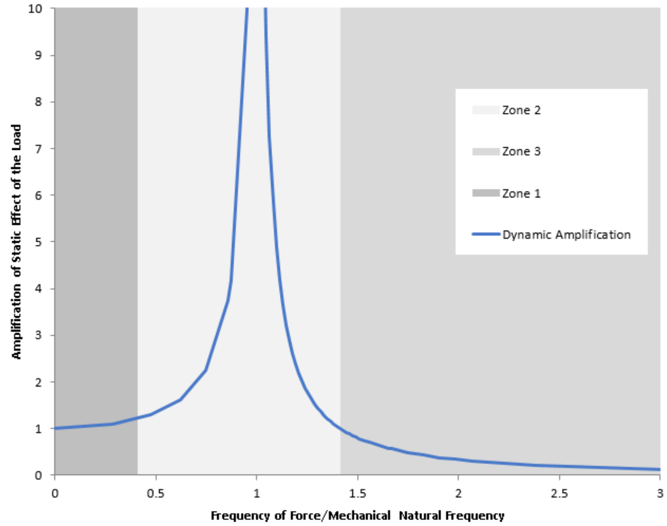

Figure 4: Response of a Single Degree of Freedom System to Harmonic Excitation.

A slightly more complex case is illustrated in Figure 5. The goal of this figure is to give the reader a better appreciation of the level of complexity involved in designing a structure that presents multiple MNFs subjected to a force composed of several harmonics. Figure 5 illustrates the case of a structure that presents two MNFs, one at 10.5 Hz and another at 28 Hz. The dynamic amplification of each MNF corresponds to the blue and red curve, respectively. This structure supports a machine that runs at 900 rpm. As explained before, this machine will generate load components at 15 Hz (1X), 30 Hz (2X), 45 Hz (3X), etc. These first three harmonic load components are illustrated as black vertical bars in Figure 5. It is seen that it is primarily the 2X load component that will generate dynamic effects, and will excite the second mode mostly. The first mode (blue curve in Figure 5) will not be excited as all three components of the loads have a frequency higher than the first MNF.

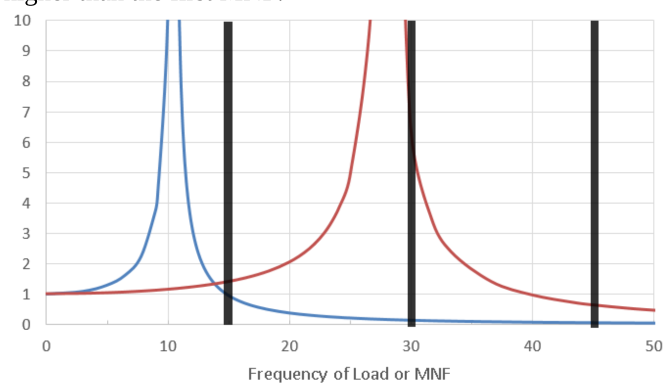

Figure 5: Response in the Case of Two MNFs and Forces up to 3X.

Another important difference between a design that takes dynamic *and* static loads into account is in the solutions that can be proposed. This is illustrated in Figure 5, as one way of reducing the impact of the 2X loads acting as 30 Hz is to change the value of the second MNF currently predicted at 28 Hz. Also shown in Figure 5, the impact of the loads can be reduced by either *increasing* or *decreasing* the MNF. Increasing the MNF can be accomplished by adding stiffness without increasing mass by the same proportion. What is less intuitive is that the same reduction in effect can be obtained by *reducing stiffness*, in other words, by decreasing the size or even completely eliminating some supporting beams. This will result in a lower MNF and reduce the impact of the 2X loads in the example shown in Figure 5. Such an approach is certainly different from the solutions that are sought to improve a design in the case of static loads. In the case of static loads, the main issues are typically related to excessive stresses which cannot be solved by decreasing stiffness. This is another example of the importance of looking at dynamic loads early in the process and have specialists deal with these loads as their effects are complex to predict.

Frequency Avoidance or Forced Response

We have already discussed that rotating and reciprocating machinery will generate loads at a fundamental frequency called 1X as well as at multiple other harmonics (2X, 3X, etc.). We have also discussed that large dynamic effects such as displacement, force, or stress can occur when the frequency of the force is close to a natural frequency. Finding a design for which the natural frequencies and the forcing frequencies are well separated becomes almost an impossible challenge, especially for variable speed machines. This has already been discussed in the context of Figure 3 as we see that the frequency ranges between the forcing function ranges (rectangular black bands) keep shrinking until they become inexistent. Finding a satisfactory design in this case requires further refinement. It involves the calculation of what is known as modal participation factors. The modal participation factor for a mode measures the coupling between the forces applied to a structure and that vibration mode. If the participation factor for a specific mode is large, then it means that this mode is coupled to the force and it can easily be excited by the application of that force. If the frequency of that mode and the frequency of the forcing function are close, then large dynamic effects will occur. If, on the contrary, the modal participation factor is small, it means that the coupling between the force and the vibration mode is weak. In more mathematical terms, we can say that the forcing function does very little work as it goes through the vibration mode. This means that this mode cannot be easily excited. As a result, even if the frequency of that mode is close to the frequency of the force, no detrimental effect can be expected since even though the amplification of static effects will be important, these static effects will be so small that the resulting dynamic effects will still be small. The consideration of the modal participation factors requires the solution of a forced-response problem which is more challenging than the simple calculation of the MNFs. However, a forced response calculation reveals a lot more about the structure and its response to the dynamic loads. The previous discussion also demonstrates why design specifications that rely only on requirements for MNFs to be avoided within a certain frequency range of the machine operating speeds are not always relevant or practical for offshore machinery structures. Offshore structures have many MNFs that are not practical or necessary to shift away from the machinery operating speeds.

Another aspect of the design for dynamic loads that distinguishes it from static design is the importance of the mass of the structure. As previously discussed and illustrated in Figure 3, the MNFs enter into the calculation of the effect of a dynamic load. An accurate prediction of natural frequencies is consequently a requirement for the accurate prediction of the effect of dynamic loads. The natural frequencies of a structure correspond to the square root of the ratio of stiffness to mass. This means that not only must the stiffness of the structure be accurately modeled as it is the case for static design, but also the mass. This is a key requirement as a big mass located on a platform deck close to the skid-mounted machinery will play the role of a boundary condition that will reduce the vibration levels on the skid as well as their propagation away from the skid. The information on the mass surrounding the equipment being analyzed is certainly not trivial to obtain as this information is sometimes not known. It is however critical for the accurate prediction of vibration levels both on and off the skid.

Now that the basics of the dynamic loads generated have been described along with the response of the supporting structure, we turn our attention to the propagation of these loads from their point of application to where they will eventually be supported. The installation of rotating or reciprocating machinery on an FPSO or platform deck can be accomplished in two common ways: One way is to weld the skid to the platform. The other is to install Anti-Vibration Mounts (AVMs). Each alternative can result in reliable designs and the pros and cons of each are briefly discussed here.

Anti-Vibration Mounts or Direct Welding

AVM is a relatively generic term that is often associated with rotating equipment. In the scientific engineering world it is often considered as a mounting technique that will decouple the machinery or machinery skid from the supporting structure. AVMs come in many different shapes and forms, but generally use a combination of stiffness and damping to achieve the desired decoupling. Commercially available AVMs may be constructed of one or more elements such as steel springs, elastomeric elements, wire mesh pads or hydraulic components. Some devices are engineered and designed to achieve a specific balance of stiffness and damping that are unique for a particular project.

Regardless of the specific design of the AVMs, the overall goal of the AVMs installed between a machinery skid and a platform or FPSO deck is the decoupling of the vibrations occurring on the skid from those occurring on the deck. An AVM design is sometimes desirable as it allows the design of the deck and the skid to occur simultaneously. The challenge is, however, to find the right number and location of properly designed AVMs. The vibration modes and AVM design can be classified in two categories. The first category corresponds to the flexing of the AVMs and the rigid property of the skid. These low frequency modes are often called rigid body modes. The second category of modes corresponds to the flexing of the skid beams and the pedestals. These modes are sometimes called flexible skid modes; they correspond to higher MNFs than the modes in the first category. A second important consideration is that most of the energy coming from the forces generated by a rotating or reciprocating machine are at 1X and 2X runspeed. Keeping these two considerations in mind, we assert that a proper AVM design will locate the first category of modes below the 1X runspeed and the second category of modes above 2X runspeed. As will be shown later, these requirements lead to a stiffer skid presenting heavier beams than a design where the machinery skid is welded to the offshore structure. Because the skid is heavier, the choice of the AVMs and their number becomes critical as the AVMs must be strong enough to support the dead weight of the skid. However, adding more, and stronger AVMs will also make the connection between the skid and the platform deck stiffer, resulting in modes of the first category and higher MNFs, possibly getting close to the 1X runspeed. This is where the main challenge resides for a successful AVM design: placing the modes in the first and second category, and in the proper frequency band.

Another way to attach a skid to a deck is by welding. The design of the skid and the deck then becomes an integrated exercise which represents a challenge in itself. This is in part due to the platform already being under construction when this design exercise happens; beam sections have been selected and ordered. The requirement that the platform deck stiffness and mass be modeled also represents a challenge as this information is not readily available to the vibration consultant and the machinery skid packager. Changes to only the machinery skid structure design are not sufficient to eliminate vibration concerns. A comprehensive structural dynamic study of the skid and offshore structure is necessary to determine the required changes. The key to a successful project is to have the owner involved in making crucial decisions about design modifications, with input and direction from the vibration consultant.

Another challenge resides in the fact that anchoring points for the machinery skid are often required under the driver (engine or motor), driven equipment (pump or compressor) and the scrubbers. These locations cannot be accessed since a deck plate is usually welded on the top of the platform or FPSO deck. Even when the skid is placed directly on the platform beams, accessing these locations is only possible from the deck below which then requires above-the-head welding which presents access and safety issues. Another way to access these locations is by not installing a deck plate on the top of the skid during its fabrication. This is sometimes accepted by the packager but certainly represents an additional complexity and likely a tripping hazard during the installation of the equipment on the deck. One option is to install special access panels in the skid.

Finally, since the design becomes an integrated exercise, the model that must be set up to predict the vibration levels will be quite large and take more time to run. This is especially true when multiple skid-mounted machines are placed next to one another on the same platform or FPSO deck, as shown in Figure 6. The phase relationship of vibrations from different skids is not fixed or known. The vibration generated by each machine will interact, possibly adding to each other in some areas of the skids and the platform and possibly subtracting in others. A conservative design approach simulating the response of individual units and then adding the resulting vibrations is required.

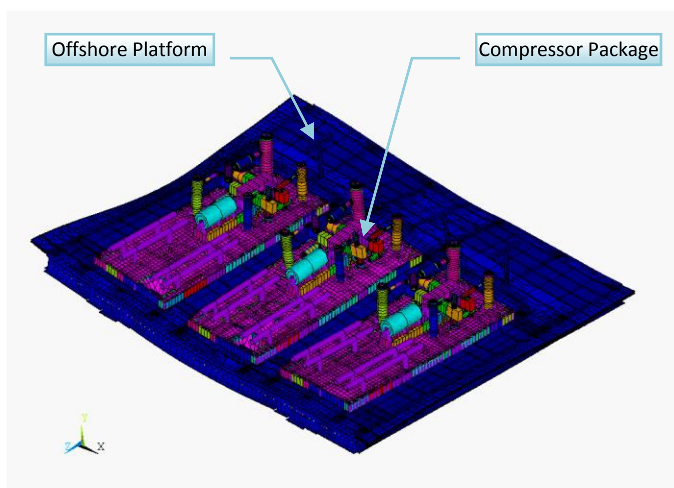

Figure 6: Consideration of More than One Unit and the Resulting Interaction of These Units.

It is seen that, although there are two possible methods of connecting a skid to a platform deck, both present design challenges that must be addressed as early as possible.

KEY FACTORS IN A SUCCESSFUL PROJECT

Two key success factors of a project are the proper timing for involving a vibration specialist in the project and the clear definition of roles and responsibilities during the project.

Timing

The timing or scheduling of activities in the project evolution has been shown to be one key factor in the success of a project. A typical project begins with recognizing a business opportunity. Preliminary planning and engineering is done early after recognition of the opportunity to determine technical requirements and operational limitations. This step is often referred to as the FEED (Front End Engineering and Design) stage. The project owner is typically involved or initiates the work with involvement from a general engineering consultant along with equipment suppliers and packagers. If the project does not encounter any technical or economic road blocks, the project progresses to a detailed design phase. The equipment packagers are awarded contracts at this stage and detailed engineering is done. Fabrication, construction, and installation takes place before the project is turned over to commissioning and operations. The timeline is shown in Figure 7.

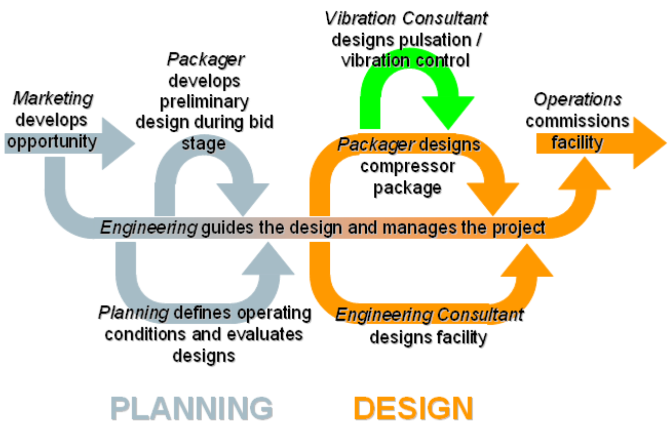

Figure 7: Typical Timeline for Vibration Assessment of Equipment Packages.

Projects following this timeline often have limitations or compromises in the final design to accommodate vibration control, as the vibration consultant is brought too late into the project. Many aspects of the facility and individual equipment packages which have a significant impact on the vibration control strategy have already been decided upon. Detailed design may have progressed to a point where changing the design to mitigate vibration concerns is not possible without causing significant schedule delays or major cost increases. One example is the selection of a horizontal opposed throw reciprocating compressor. A 4-throw compressor may be selected to minimize the size of a compressor package over a larger 6-throw compressor footprint. A 6-throw compressor typically has very low unbalanced mechanical loads as compared to a 4-throw compressor. A 6-throw compressor would have been a better selection to minimize vibration. The 4-throw compressor package may require a much stiffer skid (baseplate) and deck design and or more anchor points to minimize vibration. The extra time to redesign the structure at the detailed design stage, cost for extra material, weight of extra structural components, time for fabrication and possible compromises in maintenance access due to extra structure can result in the 4-throw package design being more costly than an early decision to use a 6-throw compressor.

The recommended timeline for involvement of the vibration consultant is illustrated in Figure 8. The vibration consultant should have input very early in the project planning stage. Key decisions such as mounting techniques for equipment packages, arrangement of equipment packages on module decks, preliminary sizing of process vessels, approach for pipe routing and design of small bore piping and instrumentation connections can be made with input from the vibration consultant very early in the project to minimize costly redesign late in the project.

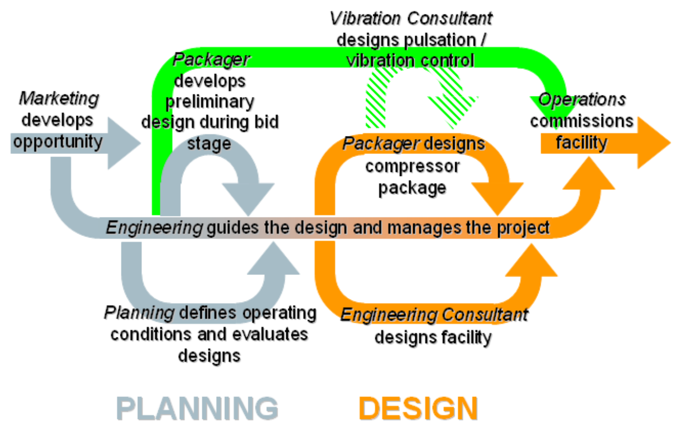

Figure 8: Recommended Timeline.

Roles and Responsibilities

A typical flow chart for a project from the FEED stage through to the operations is shown in Figure 9. Owners may not have the technical resources or workforce to carry out the engineering, purchasing, and construction of the project so an engineering consultant company (EPC) is hired. The EPC will then solicit bids for the different equipment packages for a project. The equipment package vendors will complete the design and construction of the package to the owner and EPC specifications. The vendors may require an engineering specialist company to meet particular requirements outlined in the specification. Each vendor may have different engineering specialists who are contracted. Additionally, the EPC or Owner may hire engineering specialists to perform commissioning and assist in implementation of the vibration integrity program.

This approach has several shortcomings mainly resulting from the work being done by multiple engineering specialists or consultants.

- There will be duplication of effort in the design process where there is overlap. For example, a consultant evaluating a pump package design for vibration will create a finite element model that will need to be duplicated by the consultant evaluating the structural design.
- Having many parties involved results in more complicated communication.
- Delays may result from coordinating schedules for many different parties.
- There may be a lack of consistency, overall vision, and goals for the project.

The process in Figure 9 has the disadvantage for owners that they often have less or no control on many vibration issues. The vibration consultant is directly responsible for the vendor package. The interests of the vendor packager may not be aligned with the owner's. One commonly seen limitation is that the owner's asset life cycle interests and risks are not addressed adequately.

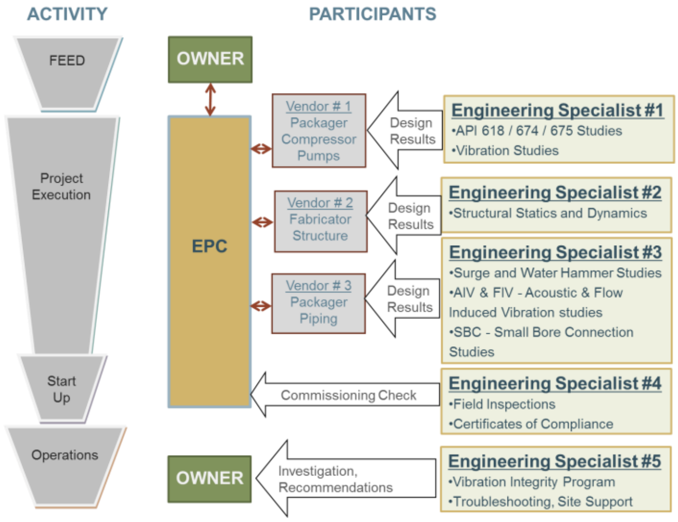

Figure 9: Typical Project Roles and Responsibilities.

It has been shown in many projects that one key to a successful project is more involvement by the owner. Ideally the owner will hire or specify the engineering specialist conducting vibration related studies. This step ensures the owner's goals and interests are a high priority. The owner can also be directly involved in making key decisions along the design process that better meet their objectives. Figure 10 illustrates the recommended process. A single engineering specialist or consultant is involved in providing the specialized design studies for minimizing vibration risks. This results in a short schedule, single point of contact and responsibility, and reduction in redundancy and costs. The engineering consultant must also be involved in the commissioning and site support during operations. This improves the effectiveness of the commissioning and operations support as the complete background factors and details of the design process are known by the consultant.

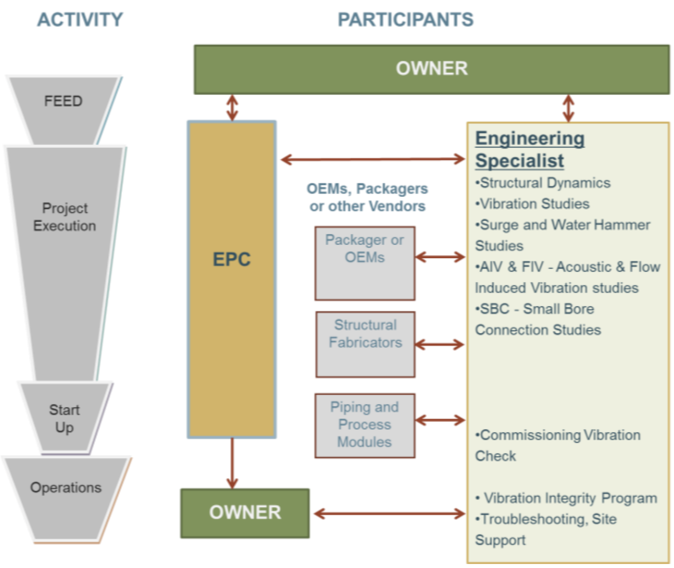

Figure 10: Recommended Project Roles and Responsibilities.

CASE STUDIES

The following three case studies are used to illustrate some deficiencies resulting from the typical design process, highlight the cost and struggles that operators experience by not using an optimal design process, and demonstrate the added value of hiring an engineering specialist early in the design process.

CASE STUDY 1 – SMALL PUMPS, BIG PROBLEMS

An operator offshore Malaysia was struggling with several piping and machinery failures on a small two 110 kw triplex pumps in glycol circulation service, as well as two 200 kw centrifugal pumps in a hot oil service. There was a gearbox failure every four weeks, while the lead time for a new gearbox was six weeks. The platform was also experiencing small-bore piping failures. Every time there was a failure on one of the units, the entire production platform had to be shut down. The operator originally took a trial and error approach of changing the gear oils, alignments and replacements of equipment, laser alignment of the skid, installing additional platform beams and charging and discharging dampeners. None of the attempts to resolve these issues made a positive impact, and the owner consulted BETA to help resolve the issues.

Upon field inspection it was noted that there was high vibration on the pump motor piping, skid, platform beams, and deck plate in the vicinity of pumps. There was also high pressure pulsations measured on the discharge line of the triplex pumps.

On the triplex pumps, there was considerable flexibility on the pump skid and platform beams. Figure 11 is an operating deflection shape (ODS) of the unit which shows the vibration amplitudes. Due to the relative movement between the motor and gear box, undue stress was placed on the gear box which explained the frequent failures. Vibration levels were measured on the gear box at 0.30 ips pk at 5X plunger passing frequency and 0.34 ips pk at 7X plunger passing frequency and also on the platform at 0.57 ips pk at 7X plunger passing frequency. An acceptable guideline is 0.1-0.2 ips pk.

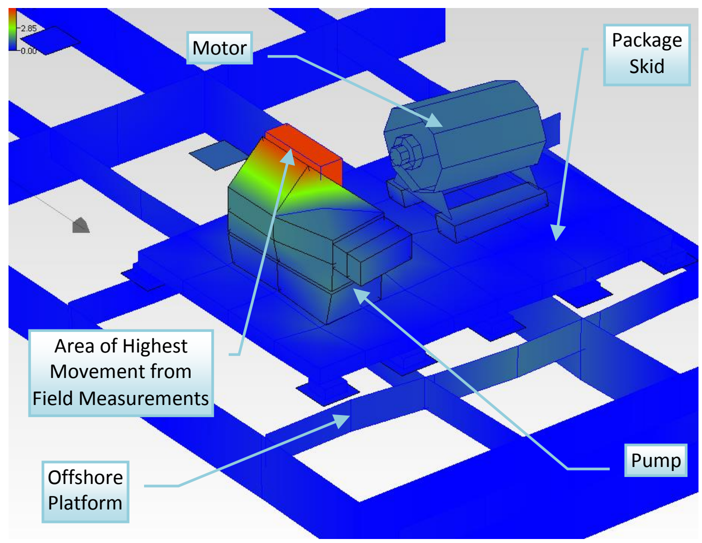

Figure 11: Field ODS Measurement Showing Relative Motion between Pump/Gearbox and Motor.

The hot oil pump was also experiencing relative movement between the pump and the motor which can be seen in Figure 12, showing the visualized ODS measurements. The four vertical posts used to mount the pump were moving in axial direction, the motor drive end was moving in vertical direction. Excessive force was being added to the magnetic coupling which was causing rubbing, as the clearance was not maintained. This resulted in a major overhaul requirement every six months and failures within the six-month period. Vibration was measured on the motor junction box 0.64 ips pk at 1X runspeed of the pump and on pump DE at 0.23 ips pk at 1X runspeed, and 0.20 ips pk at 1X runspeed on the deck beneath a skid pad (where the skid connects to the platform).

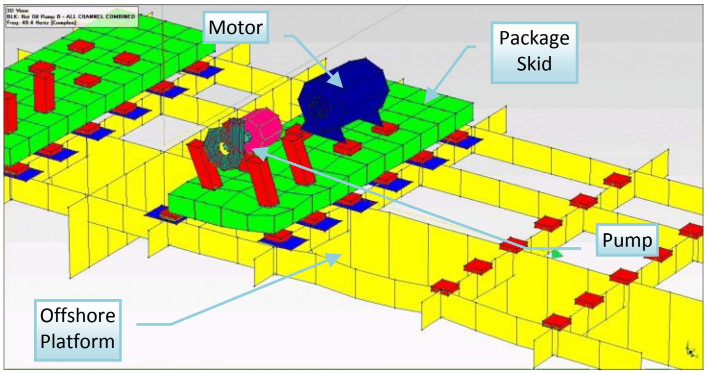

Figure 12: Field ODS Measurement Showing Relative Motion between Pump and Motor.

Both systems had structural problems which required FEA of the complete deck to understand the behavior of the systems and what type of recommendations were feasible to rectify the problems. Figure 13 to Figure 15 show the model used for the glycol pumps.

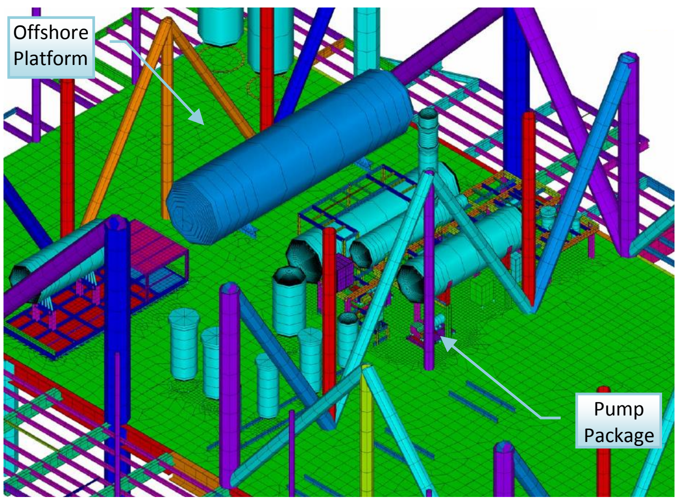

Figure 13: FEA Model of Cellar Deck Showing All Equipment.

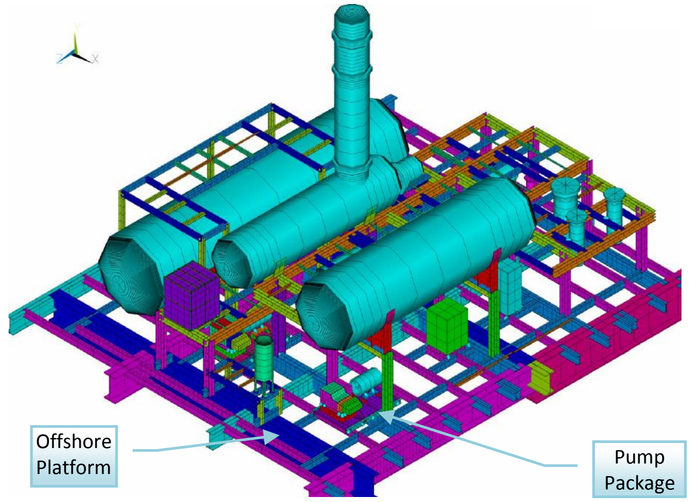

Figure 14: FEA Model of Glycol Module.

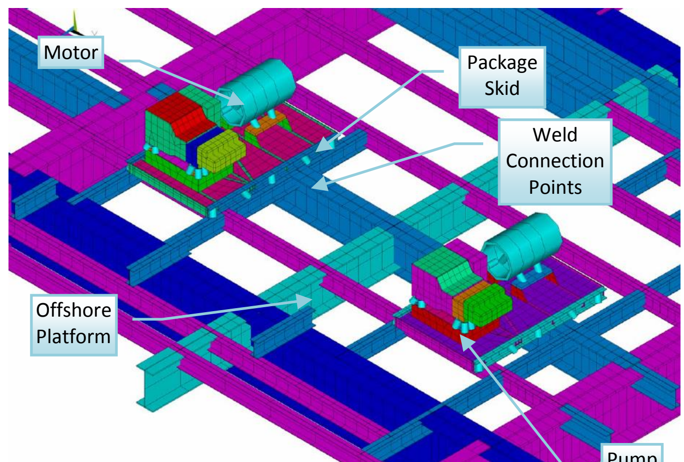

Figure 15: Close-Up of FEA Model of Glycol Pumps.

Several localized modifications were required to the skid, platform, and connectivity of between skid and platform. Modifications included adding T stiffeners to existing platform beams, boxing in skid beams, adding grout to areas of the skid, gusseting skid beams, gusseting pump pedestals, vertical support members, and mass to detune the deck plate, amongst others.

Significant time and cost was involved in order to make the appropriate modifications, but ultimately they were installed. Feedback from the operator is that vibration levels have decreased and the platform has not experienced any coupling or gearbox failures. It was the FEA models that provided the appropriate recommendations which could have been performed in the design phase. The owner/operator could have avoided the headaches and production losses if vibration was considered from the onset of the project.

CASE STUDY 2 – RECIPROCATING COMPRESSOR OFFSHORE MALAYSIA

This project followed the integrated vibration design approach, and to this date is a showcase of the approach and engineering to help ensure reliability and integrity of the machinery and connected system.

The fixed leg platform was designed and built years ago; however, due to current and future field requirements there was a need to add additional compression on the platform. The greatest concern with this project was that the main production decks were full and the only available space was located on the cellar deck, requiring the addition of a cantilevered section for additional space. At this point, the owner began speaking with a specialized engineering consultant to better understand what could be done on this platform. As FEED engineering continued, it was confirmed that the equipment required was a gas engine-driven reciprocating compressor located on the cellar deck.

BETA continued to work through the detailed design with the owner and was appointed by the engineering consultant to perform the structural dynamic engineering, as well as with the equipment packager to perform the API 618 pulsation and vibration analysis. In the end, the entire scope of work included the following: API 618 DA3, small bore connection assessment, acoustic induced vibration (AIV), pipe stress analysis, torsional analysis, skid dynamic analysis and structural dynamic analysis (platform). It is also important to note that there was a line of communication established between the owner, EPC, Packager, and BETA to ensure ease of transferring information. Discussions were open so that the owner understood the different options and impacts of decisions.

The detailed engineering for the vibration consultant began with the torsional analysis. For this type of equipment it is extremely important that the torsional study has high priority and is performed early, as the coupling requires a significant lead time. In some instance (e.g., gas engine drive), this can often be done before the packager has a general arrangement (GA) completed.

The API 618 DA3, stress analysis, and AIV analysis follow close behind the torsional analysis, with the recommendations of the finalized bottle sizes to allow the center-line of the compressor to be established and the GA to be completed. Typical outcomes and recommendations were provided to the packager for their implementation. These include items such as finalized bottles sizes, restrictive orifice plate size and location, requirement for outboard cylinder supports, PSV supports, pipe work supports etc. The project structure allowed direct discussions with the owner to ensure the recommendations maintained their best interest.

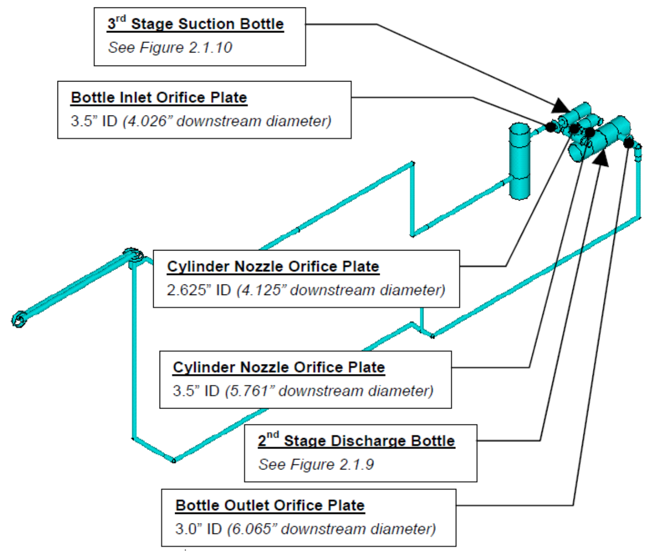

Figure 16: Interstage Orifice Plate Size and Location.

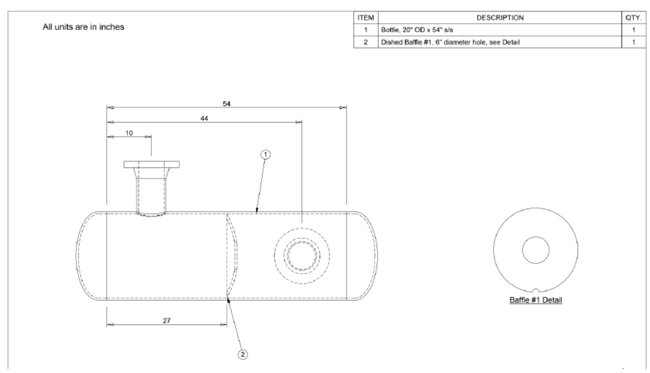

Figure 17: Example Bottle Drawing Showing Internals.

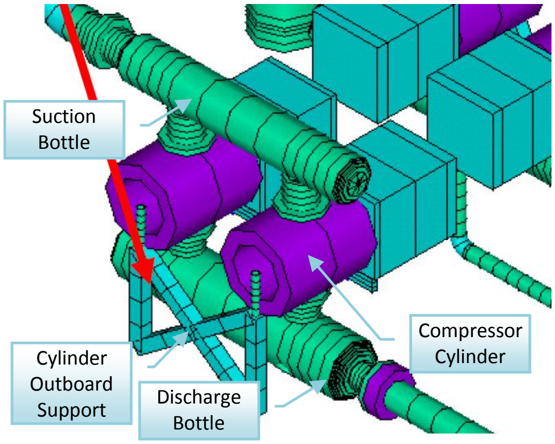

Figure 18: Outboard Support Requirement.

The skid and structural dynamic analysis was the last scope to be performed. This is also the most integrated as it links the designed package (above skid) with the skid design and connectivity to the platform into one complete system.

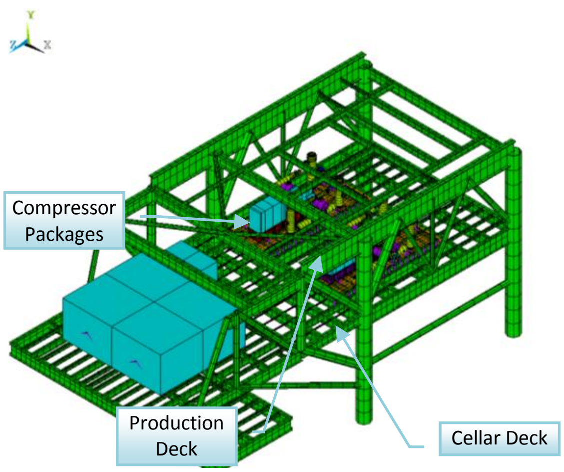

Figure 19: Complete Platform Model.

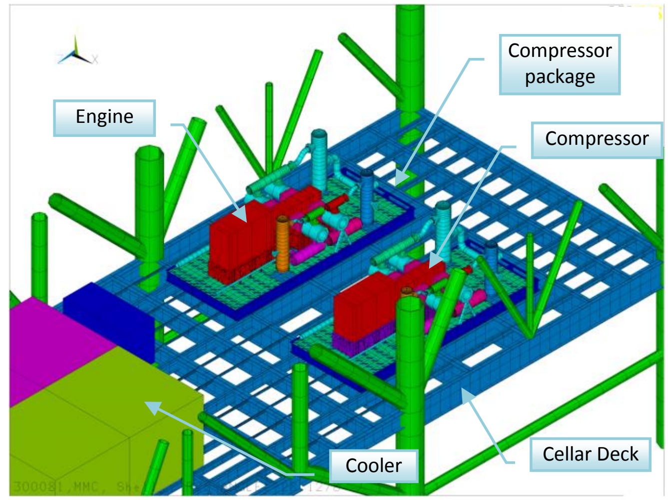

Figure 20: Platform Model Showing Compressor Detail.

After initial modeling, it was apparent that there was a significant integrity risk if the platform was not modified to reduce the vibration on the cellar deck. There were more than 30 MNFs that were coincident with the 1st and 2nd order of the compressor. That correlates to localized platform MNFs between 11.67 Hz and 20 Hz as well as 23.33 Hz and 40 Hz. Vibration was predicted to be 0.55 ips peak on the compressor deck and skid, which is more than 2x to 5x guideline levels typically used for this type of application (OEM, API, ISO, etc.). Design changes were required.

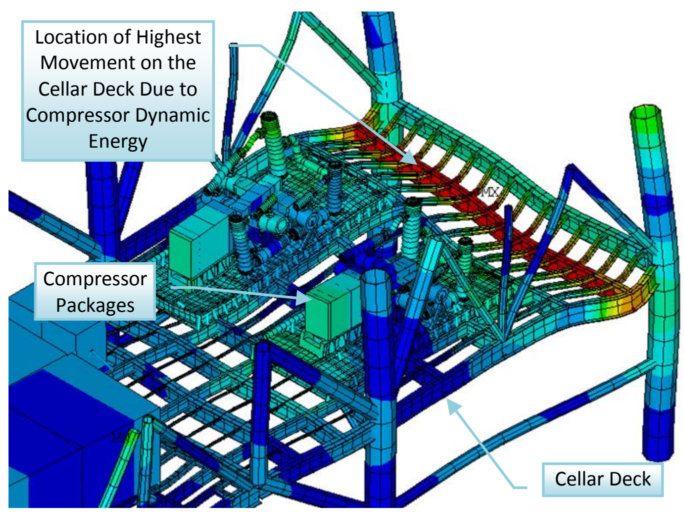

Figure 21: Vibration Response of Original Design.

Resolving this engineering issue without substantial impact on the timeline and cost of the project required significant involvement and interaction.

Many different design iterations were performed to determine appropriate recommendations for this project. On a typical structural dynamic project it is very common to add additional stiffness to the platform where needed to change the resonant MNFs that result in unacceptable vibrations. The following images show some of the reinforcements required for this platform.

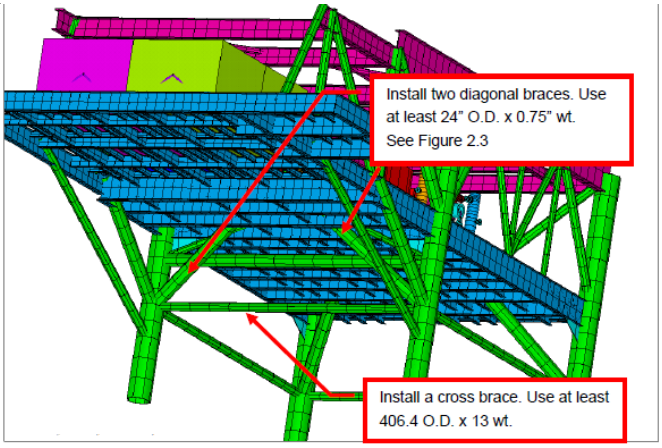

Figure 22: Recommended Platform Modifications under the Cellar Deck.

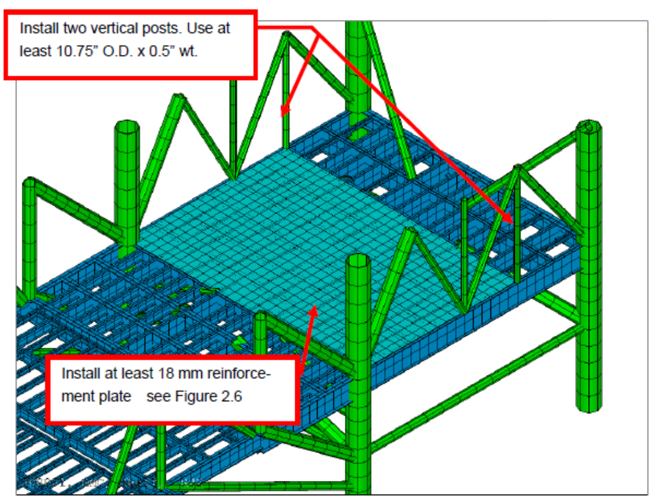

Figure 23: Recommended Platform Modifications above the Cellar Deck.

This was discussed with all parties and accepted, however, when it became clear that the platform was not going to be moved to a dry dock for installation, the cost of the cranes, scaffolding, and welding time offshore deemed this solution unacceptable.

The other common method to resolve resonance issues is adding mass. A typical material is concrete, which is often used in *on*shore applications; but not commonly used in *off*shore applications. Due to scheduling limitations the owner explored this option and worked very closely with BETA to find a solution. It is also important to note that this procedure of adding significant mass and pouring it *off*shore is not in line with the company's standard engineering practice, and significant exceptions to internal practices were required.

The final solution was to stiffen key areas, box in the area directly below the compressors, and add epoxy grout concrete to the entire depth platform. The maximum final vibration on the skid and platform was predicted to be 0.2 ips peak at any one frequency.

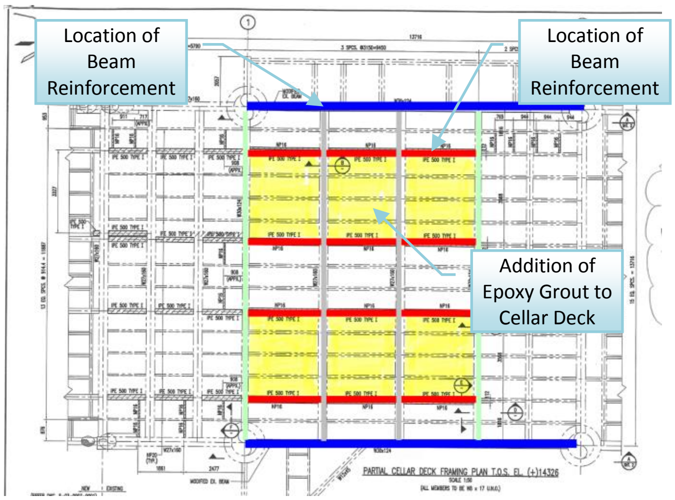

Figure 24: Final Implemented Platform Modifications under the Cellar Deck (Localized Beams Needing Reinforcement and Location of Grouting).

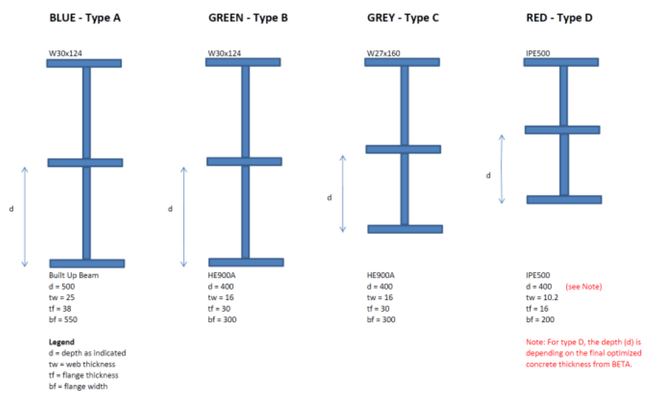

Figure 25: Final Implemented Modifications to the Structural Beams.

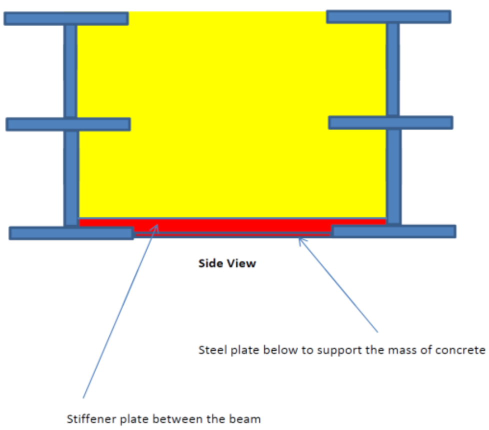

Figure 26: Final Implemented Platform Modifications to Allow Concrete Grout to Be Applied.

The steel reinforcement was performed offshore, with the pouring of the group being done in late November 2011. The compressors were commissioned shortly after. Vibration engineers were on call in case any issues arose with vibration. The owner did their own basic vibration measurements and recorded the greatest value at 3.88 mm/s rms, which equates to 0.22 ips peak overall. The greatest vibration measured at one particular frequency was 1.7 mm/s rms @ 90 Hz, which equates to 0.1 ips peak. The owner and operator were extremely happy with the unit and its operation and pleased with a reliability of above 97%.

CASE STUDY THREE – IMPACT OF AVM ON THE DESIGN OF PACKAGES

As mentioned earlier, AVMs are often used to isolate or decouple rotating equipment from the platform. Depending on the type of equipment and application they can be very effective. However, the final engineered design of a package can be very different. The following compressor package was being installed offshore on a fixed leg platform. The owner wanted to make use of AVMs as there were a total of seven packages being located on the same facility on the same compression deck.

The owner originally wanted all seven units on AVMs. BETA was involved in pre-engineering to determine the feasibility of AVMs with this application. During that process the low pressure compressors proceeded with a conventional welded design. Of the remaining three high pressure units, the owner decided to have one skid built for use with AVMs and the balance proceed with a conventional design.

The final compressor selected for these units was a 4-throw reciprocating compressor directly coupled to 1050 kW fixed-speed electric motor at 1000rpm.

The equipment, operation, and process requirements were identical for both the conventional welded design and isolated AVM designed skids.

The conventional skid relied on additional beams at specific locations, and additional gusseting and supports. The connectivity to the platform was a complete perimeter weld between the skid beam and to the platform beams (direct weld with no deck plate), and welding along with interior of the skid directly to the platform at specific locations (beneath the scrubber). Figure 28 shows the skid connection points to the platform.

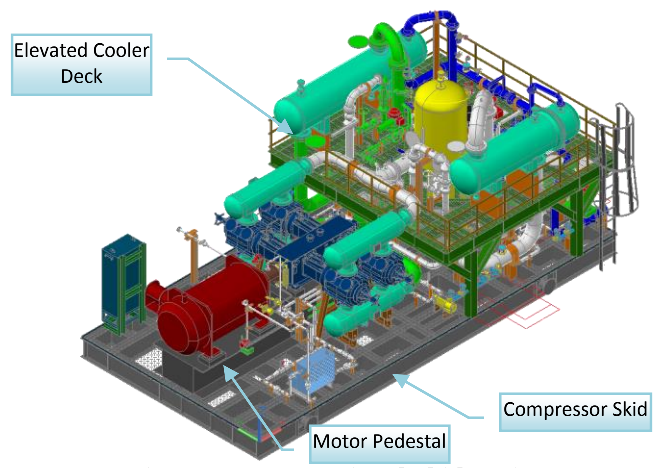

Figure 27: Conventional Skid Design.

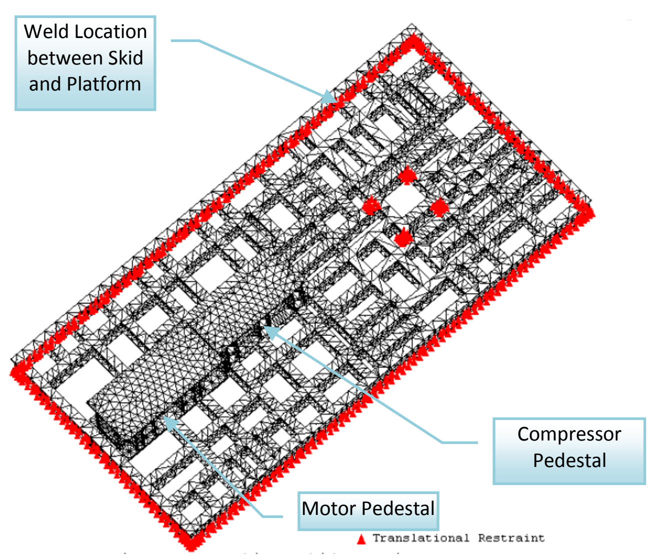

Figure 28: Conventional Skid - Platform Welding Locations.

This design weighed in around 81,000 kg with the vibration results shown in Figure 29.

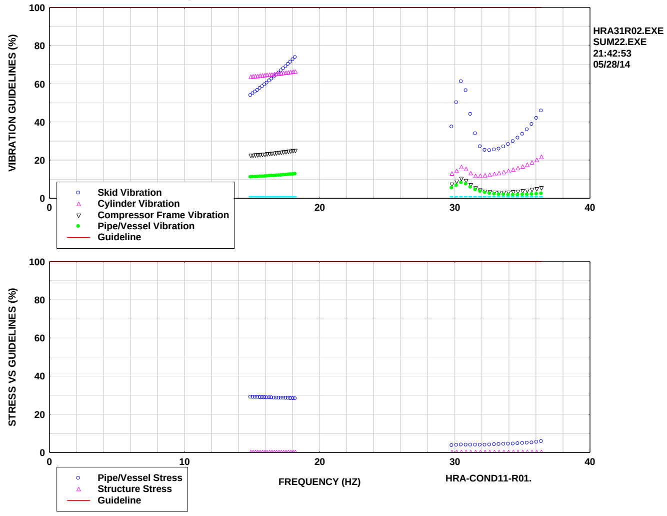

Figure 29: Conventional Skid - Designed Vibration Levels.

The AVM designed package was considerably different from the conventionally supported package. In order to achieve a reasonable dynamic response the following modifications were required for the skid:

- Increase the beam depth to 900mm
- Fill the entire skid with grout
- Modify and increase the robustness of the compressor pedestal
- Increased stiffness of the second level structure to adequately support the heat exchanges and pipe work
- Placement of 16 AVMs along the perimeter of the compressor skid

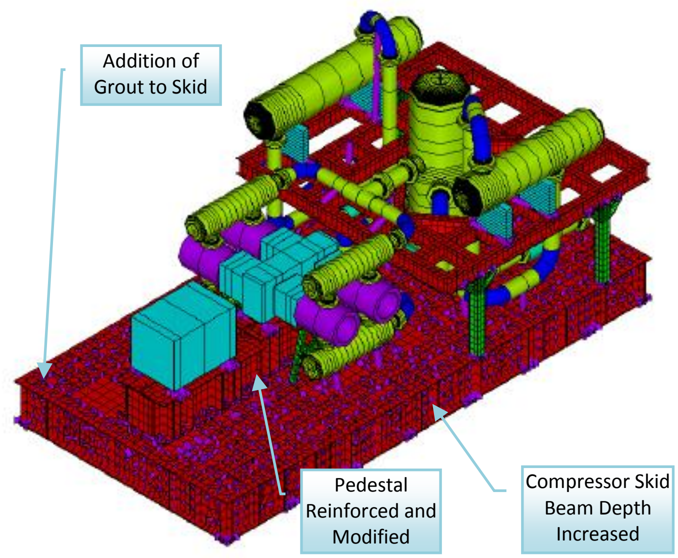

Figure 30: AVM Skid Design.

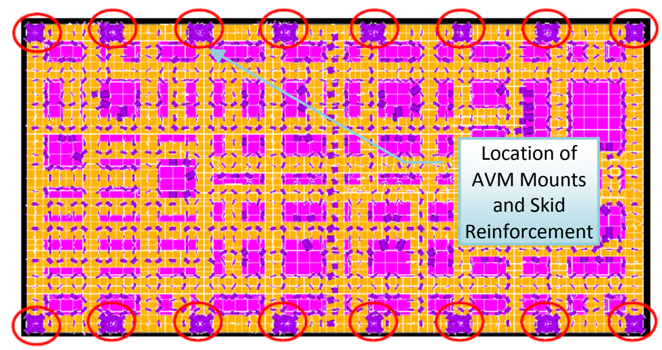

Figure 31: AVM Locations for Connectivity to the Platform.

This design came in weighing 225,000 kg with the calculated vibration levels shown in Figure 32.

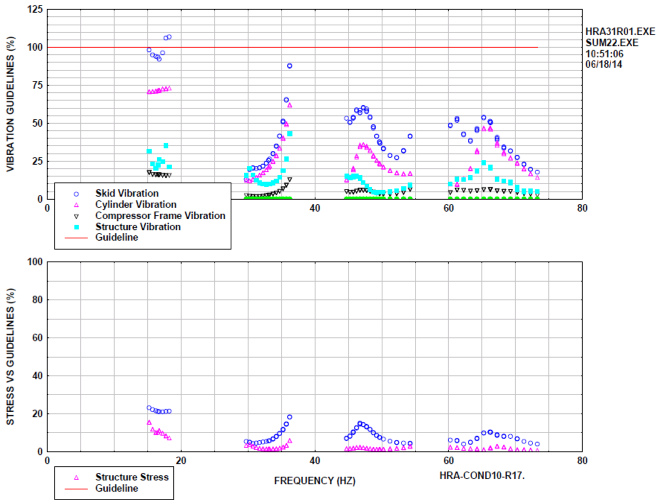

Figure 32: AVM Skid Designed Vibration Levels.

In the end both packages have achieved acceptable vibration levels, and increased the integrity and reliability of the machinery and attached pipe work, however, they achieved this in different ways. Early involvement with the owner allowed for feasibility studies to be performed, which significantly changed the path and final project. The owner's budget, risk tolerance, and project timeline only allowed for one compressor with AVMs, as a trial and reference for future projects. The overall cost of implementation, required design modifications, and maintenance costs will be monitored by the owner for their next project.

CONCLUSION

The currently accepted approach for structural design work goes a long way to optimize the steel used in a facility and ensures that the dead weight of all components is adequately supported; however, further engineering is required to ensure that machine dynamics are appropriately considered. This paper presents options and detailed engineering solutions to evaluate the dynamics of platforms, and uses several case studies to highlight the problems operators are facing to ensure the desired reliability and integrity of the machinery and piping.

Engineers involved in brownfield upgrades or greenfield projects must consider structural dynamics and vibration concerns early in a project life cycle. The project owner is recommended to have a vibration specialist consultant involved at the FEED stage. Involvement of the owner, engineering contractor, and vibration specialist throughout the project has been shown to be a successful approach for maximizing reliability and availability of machinery and piping systems on offshore facilities.

REFERENCES

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