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Offshore Requirements for Turbine Exhaust System Analysis and Design

Agron E. Gjinolli, Elden F. Ray, Cary D. Bremigan and Bruce J. Morris
Universal Acoustic & Emission Technologies
Stoughton, Wisconsin, USA

ABSTRACT

Offshore combustion turbine systems are typically installed on the topsides (deck) along with other auxiliary components. Combustion turbines are primarily used because they provide the most power in the smallest footprint. The exhaust system must perform its basic function of conducting exhaust gases that can be as high as 1200° F (650° C) safely away from the adjacent equipment, platform workers and other systems, and mitigate the exhaust noise under a wide range of conditions.
This paper presents a review of the design approaches and analysis process for development of an optimal silencer and duct system for a wide range of loading conditions including, internal pressure, environmental loads (wind, seismic, ice etc.), and aero-acoustic performance. The method to select and determine the appropriate duct system shell thicknesses, mounting points, expansion joint locations / requirements, and loads on the supporting structures are reviewed as well as the results of the final design.

KEY WORDS: exhaust system, expansion joint, base ring support, FEA (Finite Element Analysis), Von Mises stresses, buckling, and fatigue.

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INTRODUCTION

Usually, the topsides of offshore platforms require some type of power generation source that is commonly achieved by employing a combustion turbine or reciprocating engine driven generator. Offshore combustion turbine exhaust systems are complex and are required to withstand the extreme environment and operating loads including high temperatures and gas flow. It is essential that the turbine exhaust system components be designed with minimal weight for ease of installation and at the same time be adequately robust to resist the extreme environmental loads of the open seas. Figure 1 shows a typical platform with the exhaust systems installed on the topside of the platforms.
The offshore application typically has some combination of forces: snow and ice, seismic, wave action, wind forces, vessel movements, possible vessel impacts and potential fuel gas explosion. Seismic loads, vessel movements, wave action and possible impacts are commonly summarized as acceleration forces. In addition, the flow induced vibrations represent a critical load case that has been known to be a common cause for failures. These inputs must all be considered in the structural design to ensure that the system will have the necessary freedom for thermal expansion while minimizing stress levels at connection and support points. The inputs must also be considered when designing the expansion joint configurations to achieve the required design life. The exhaust system must continue to perform its functions under adverse conditions, requiring the engineer to design for worst case scenarios with multiple inputs and/or load combinations acting on the system.
In addition, the system restriction, acoustic performance and thermal impact on surrounding structures and equipment must be optimized, while maintaining a compact space envelope.

Fig. 1. Typical exhaust system locations (BP Bruce installation, 2011).

Turbine exhaust system modeling is challenging because of the complexity of designs and takes an experienced team to develop a robust design. This paper presents the design approaches and requirements for turbine exhaust systems installed on offshore platforms including modeling consideration based on the exhaust temperature, type of structural modeling and analysis, thermal modeling and allowance for thermal expansion, mass modeling and simulation of the controlling load conditions based on the current applicable codes. A discussion of the design consideration for the topside equipment specifically related to the exhaust silencer system and the structural steel support is presented.

STRUCTURAL ANALYSIS AND DESIGN CONSIDERATIONS

Finite element analysis (FEA) has become a popular method for evaluating the structural performance of turbine exhaust systems, and other topside auxiliary equipment. FEA is used to perform stress and stability/stiffness analysis and optimization of the structural design when subjected to various forces. In addition, it is used to perform structural failure investigations or studies for special loading conditions. Special loading conditions may include flow induced vibration failures, internal explosions, wind resonance (vortex shedding) failures, and thermal fatigue induced by differential thermal expansion during start up and shut down in combination with wind, seismic or other forces and loads.

Codes, standards and recommended practices typically used for analysis and design of the turbine and reciprocating engine exhaust components and systems include ANSI/AISC 360-05/10, "Specification for Structural Steel Buildings" by American Institute for Steel Construction, AWS (American Welding Standard), 2009/2012 IBC (International Building Codes), ASCE 7-05/10 (American Society of Civil Engineers) "Minimum Design Loads for Buildings and Other Structures", and the ASME (American Society of Mechanical Engineers), Pressure Vessel Code, Section VIII, Division 1 and 2, and BS 6235-1982 (British Standards), Code of Practice for Fixed Offshore Structures.

Structural design analysis and verification typically includes a strength and stiffness check to ensure that adequate structural resistance is achieved related to the material yield strength and stiffness (deflections and rotations) for all structural components, a stability check to ensure the adequate resistance to buckling for all structural components as well as a local buckling check for the structural shell / plate elements under compression. In addition, analysis of thermal stresses and expansion of the connecting joints as well as supports is required.

The design loads for offshore topside equipment typically include dead loads (weight of the structural components and equipment permanently installed on the platform topsides), internal pressure, environmental loads (wind and seismic), thermal, construction loads (installation of the topside components, scaffolding etc.), and accidental loads (blast overpressure and accidental impact / collision).

Note that the modeling method described in this paper is presented as a general concept with specifics related to the turbine exhaust system; however, this modeling approach is applicable to other topside auxiliary components and structures. Generating the FEA model frequently requires modeling the entire structure including silencer, ducts, base ring support, shell support lugs, material properties at elevated temperature, material mass (density) and load modeling assumptions. The entire modeling and analysis tasks will be described within a typical example including controlling load cases per current ASCE 7-05, 10 load combinations.

Typically, the turbine exhaust system modeling uses two types of finite elements including beam (stick) and plate elements based on the Timoshenko's "Thin Plate Theory Analysis". However, there are situations where solid elements are used as well. Solid element is using standard 8-node isoparametric formulation. One of the most important FEA considerations includes modeling for buckling of beam (stick) and plate (shell) elements. Furthermore, the plate/shell elements need to satisfy the requirements for the adequate strength/stiffness and stability against localized plate buckling. Most of the commercial FEA software packages include Second Order Elastic Analysis (SOEA), usually called "P-Delta" analysis. When the structural model is loaded, it deflects and potentially rotates. The deflections in the members and plate elements of the model may induce secondary moments due to end joints of the members deflecting and/or rotating, and may no longer be lying on or passing through a single straight line in the deformed position (deflected joints are collinear). These secondary effects, for members and plate/shell elements can be accurately approximated through the use of P-Delta analysis that is usually called P-Large Delta. P-Large Delta method is based entirely on the nodal deflections and rotations. As an example, the introduction of additional nodes along the column where member displacement and/or rotation effects are at their maximum will adequately account for the effect of P-Little Delta, which may be of importance for member and plate elements reaching their elastic Euler buckling load capacity. Note that P-Delta analysis is required by both AISC 360-05 and 360-10.

Exhaust Stack System Structural Analysis – Numerical Example 1

Two exhaust systems, one straight and one slanted, both rising 150 feet above the deck have been selected for review. Figure 2 illustrates the exhaust ducting systems analyzed in this example. RISA 3D V12.0 and RISA Section V 2.0 FEA software packages in conjunction with RAM Connection V8.0 have been used to analyze these 2 exhaust systems. Finite element modeling, analysis and results for development of an optimal silencer and duct system while considering the wide range of loading conditions mentioned above, is discussed.

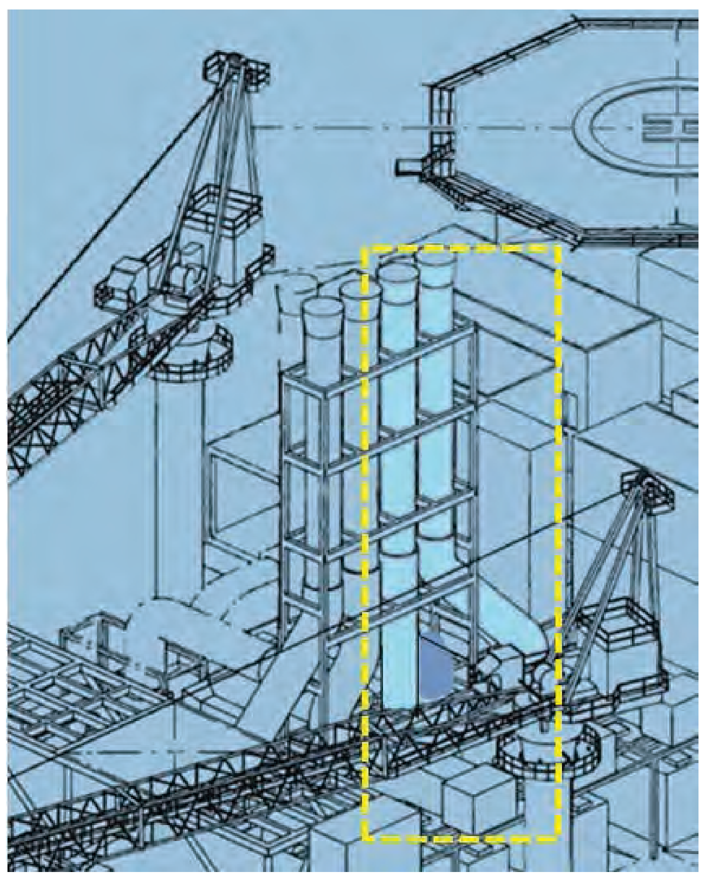

Fig. 2. Isometric view of the analyzed vertical and partially slanted vertical exhaust systems.

For these stacks, wind loads between 100 and 120 Mph, and exposure "D" in accordance with ASCE 7-05, 10 have been used. Total topside seismic spectral acceleration requirements in horizontal and vertical direction are in excess of 0.8g and 1.2g respectively.

The example exhaust stack system represents a combination of both internally lined (insulated duct) and unlined components. Lined construction/components include the silencer and ducting up to the second expansion joint. Lined duct components (cold shell) are typically manufactured using carbon steel for the outer structural shell and stainless steel liner, which is supported via scallop bars that in turn are attached to the outside shell. This stack system includes 316L stainless steel design for both lined and unlined structural (outside) shells. Stainless steel is commonly used for marine applications. Insulation pack (basalt, fiberglass, ceramic fibers, etc.) is placed between the outside shell and the internal liner to keep the outside shell considerably cooler than the high internal gas flow temperature and for acoustic purposes. The lined exhaust silencer is at the base of the system, and it is connected to the deck through double base ring construction and anchor bolts. Silencer is lined similarly as the lined ductwork section/components with the acoustic element attached to the liner. Gas flow temperature depends on the turbine type, size and operating conditions, it typically varies between 950 and 1200º F. To account for the thermal growth of the stack system, cold-to-cold expansion joints between the lined duct sections are installed. Similarly, thermal expansion between the lined and unlined duct components (hot shell) are connected via cold-to-hot expansion joints.

Three types of finite elements are used to complete the analysis of this system, including 2-node beam, 3-node triangular, and 4-node quadrilateral plate elements. Approximately 85,000 plate and 4,500 beam elements are used in modeling of these two stack systems. The first step in FEA modeling includes determining the strength and stiffness (elastic) properties of the components at the design temperature, Table 1. This task includes computing yield and ultimate strengths at temperature, modulus of elasticity, shear modulus, Poisson's ratio, as well as the coefficient of thermal expansion. In order to complete this task, a CFD (Computational Fluid Dynamic) analysis needs to be performed to determine the potential hot spots as well as temperature of the components in critical locations, specifically in the hot shell areas. The solution to a fluid dynamics problem typically involves calculation of various properties of the fluid, such as flow velocity, pressure, density, and temperature, as functions of the component(s) cross sectional area and time. ANSYS Fluent software has been used to perform the CFD analysis of these 2 stack systems.

Table 1. Assumed material design properties used in FE modeling.

Design Material Properties for Unlined SS316L Shell Components				
System Component	Yield Strength Fy (psi)	Modul of Elasticity E (psi)	Shear Modul G (psi)	Coeff. Therm. Expansion, $\alpha_{T,}$ x(10^{-6}) º F
Shell	14,975	22,800	8,840	10.30 (950 º F)
Supp. Lugs	19,500	25,400	9,770	9.90 (590 º F)
Flanges	16,333	24,300	9,340	10.05 (780 º F)
Stiffeners	17,000	24,800	9,540	10.00 (690 º F)
Base Rings	26,800	27,900	10,730	8.90 (190 º F)
Design Material Properties for Lined SS316L Shell Components				
Shell, Head Collar	23,000	27,000	10,380	9.20 (300 º F)
Supp. Lugs	26,800	27,900	10,730	8.90 (190 º F)
Flanges, Stiffeners	23,000	27,500	10,580	9.20 (300 º F)

Density is used in the calculation of the member and plate (shell) self-weight – dead loads. This density times the area gives the self-weight per unit length for the members, while density times plate volume gives the total weight of a given plate (shell) element. Expansion joint mass is calculated and properly distributed to the adjoining (connecting) stack sections (flanges), and consequently do not contribute to the global stiffness of the stack systems.

Internal mass such as the acoustic element (bullet) modeling is typically simplified such that their stiffness and load distribution is adequately represented within the silencer structural shell. This simplification helps to reduce the number of elements used in the model. Figure 3 illustrates the simplified acoustic element (bullet) modeling approach.

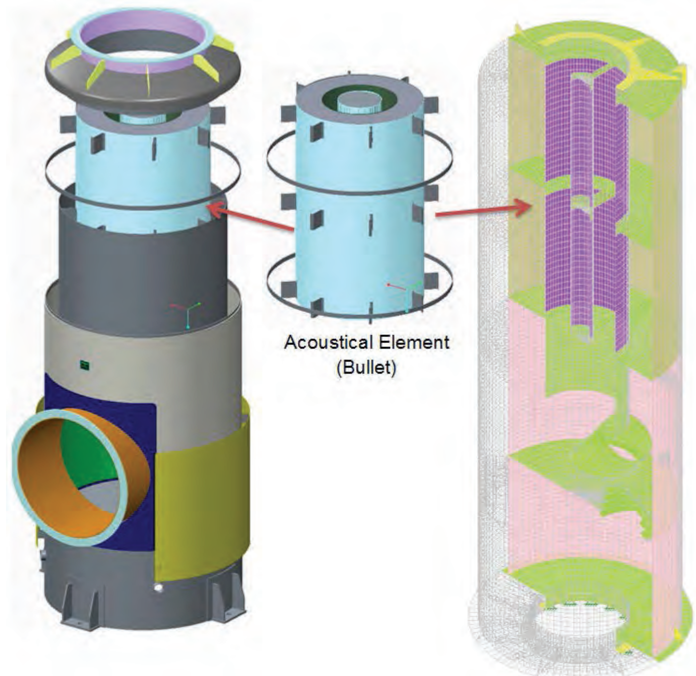

Fig. 3. Isometric view of the typical silencer including the acoustical element (bullet) on the left and middle, and FE modeling example to the right.

The acoustic element (bullet) is supported by vertical "Z" bar supports to the inside shell liner. The shell liner is simply supported at the bottom silencer duct so the whole assembly is free to thermally grow longitudinally with the bullet. This also dynamically decouples the bullet and liner from the duct structure.

The density of the acoustic element's perforated shell and the acoustical packing has been calculated and modeled so the overall mass is accurate and properly distributed to the liner. The bullet shell (skin) is a thin perforated stainless steel sheet, which would result in the low natural frequencies in the model. Since most of the time the bullet is not the object of the analysis, the bullet skin thickness has been modeled to increase the natural frequency so that bullet modes are not included in the dynamic analysis. It is assumed that the duct internal liner does not contribute to the global stiffness of the stack systems. The bottom ring of the base support is modeled using quadrilateral plate elements with boundary conditions including pins (bolts - beam elements) and one way "compression-only" springs attached to the plate joint in the indicated direction. These springs have stiffness for negative (downward) displacements and no stiffness for positive (upward) displacements.

Once the overall FE model of the structure is completed, it is recommended to run the dead load analysis to check for any global and local joint instabilities (Figure 4). Dead load analysis revealed that plate Von Mises stresses are concentrated around the stack support lugs, base ring as well as at the joint between the inlet and silencer. Maximum calculated Von Mises stresses at the base ring are 8 ksi whereas at support lugs are 6 ksi. These calculated stresses are lower than the allowable stresses of 16.1 ksi for base ring, respectively 13.8 ksi at the lined duct support lugs (Figure 4). The allowable stresses are based on the shell temperatures and are calculated using the material properties provided in Table 1, multiplied by the safety factor in accordance with ANSI/AISC 360-05.

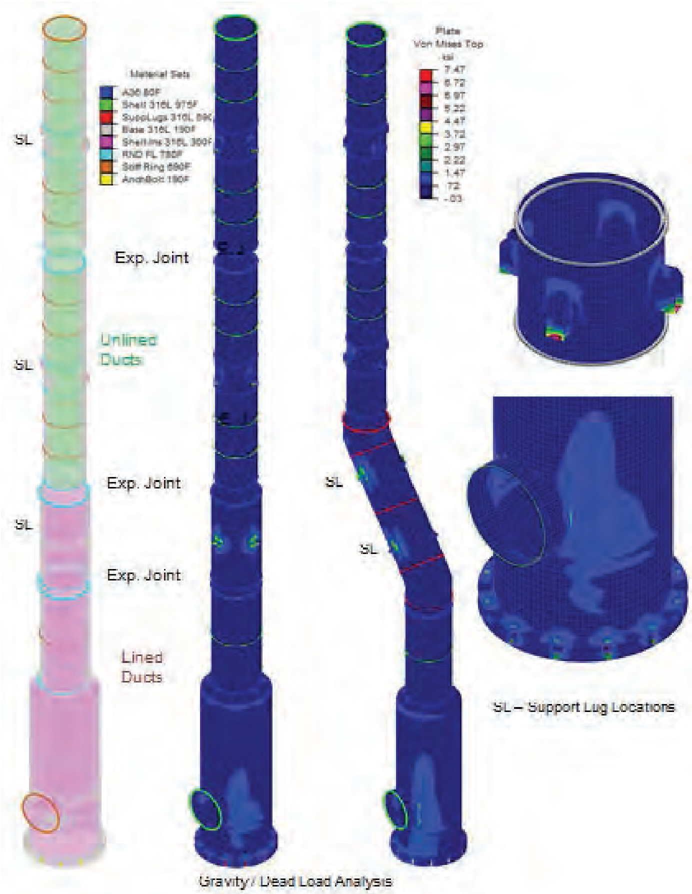

Fig. 4. Schematic view of the stack components and dead load analysis results for straight and partially slanted vertical stack systems.

Second step is to perform the wind load analysis by first running the dynamic analysis of the system.

Dynamic analysis will provide mode shapes, frequencies and periods of vibration needed for the response spectra analysis (RSA) used to calculate forces, stresses and deflections from wind loads. Since steel stacks are typically lightweight and flexible structures with low inherent structural damping, the analysis of wind induced vibrations is essential.

The dynamic characteristics of natural frequencies, corresponding mode shapes, and damping characteristics shall all be considered first in the wind load analysis. Likewise, all modes of vibration that may potentially occur due to wind loading shall be examined. Wind responses include vortex shedding and ovalling. Vortex shedding is related to the drag force caused by the actual wind flow pattern around the cylinder and the pressure distribution.

The wind flow-lines do not follow the body contour downwind creating a lower atmospheric pressure zone and a larger internal pressure, which creates unbalanced pressures between the upwind and downwind sides of the cylinder.

The drag force consists of both a friction and pressure component.

After the modes and frequencies are known determined, the critical wind speed for the stack segment can be calculated. Per ASME (American Society of Mechanical Engineers), STS-1-2011 (Steel Stacks), if the critical wind speed for the observed segment is less than the average hourly design speed at the critical height, vortex shedding must be considered.

Prevention or reduction of excessive resonant vibration is typically accomplished using damping or stiffening methods. Disruption of wind generated vortices, that can induce low frequency vibration, is accomplished by installing curved plate helical strakes attached to the outer surface of the stack (Figure 5).

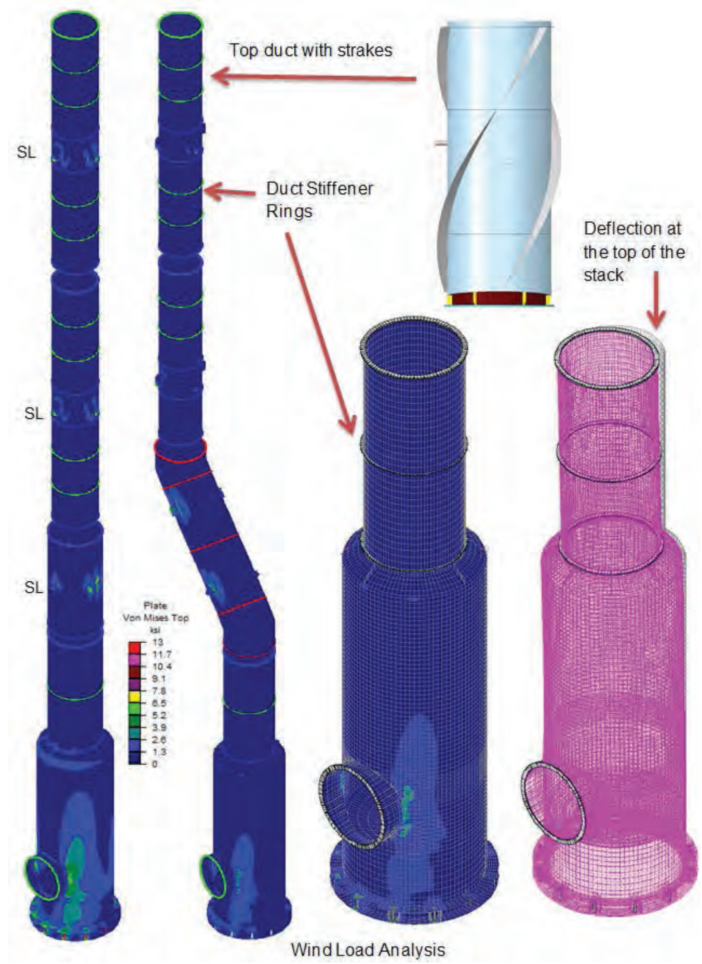

Fig. 5. Schematic view of the wind load analysis results for the straight and partially slanted vertical stack systems. Note the top duct with strakes per ASME STS-1 recommendations.

A three-start set of strakes with adequate structural thickness per ASME STS-1 are provided. Since strakes are used, the stack/system is reanalyzed with 40% increase in the wind load on the critical section caused by the drag force due to inclusion of strakes. The intermediate application of vortex forces on the stack may cause ovalling resonance.

The unlined stacks have very little inherent damping to restrict ovalling and may experience excessive stresses and deflections at the critical wind velocity, while the lined stacks are less susceptible to the ovalling resonance due to the contribution of the liner stiffness in increasing the stack overall natural frequency. Since ovalling depends on the critical wind velocity, natural frequency, shell thickness and duct diameter, the deficiency can be prevented by increasing the number of ring stiffeners which has been applied in these 2 stacks.

After determining the wind dynamic response, stack systems are loaded with calculated wind loads including the velocity pressure exposure coefficients. Wind loading and load combinations for these stacks is in accordance with ASCE 7-05 (10) and it is considered omnidirectional. This analysis revealed that maximum stack deflections are between 0.25 and 0.30in and are within the acceptable limits for both lined and unlined stack sections. Maximum stack deflections are limited to 1in per every 600in of the stack height/length as recommended by ASME STS-1 and industry practice (Bednar, *Pressure Vessel Design Handbook*). Maximum Von Mises stresses (maximum distortion energy of the plate elements) are calculated between 13 and 14 ksi, and are located at the lined duct support lug sections. These stresses are less, respectively equal to the allowable stresses of 14 ksi, which are calculated from the material design properties provided in Table 1. Similarly, maximum calculated unity check (UC) in bending and shear is observed at flanges (beam elements). These values are 0.35 for bending and 0.40 for shear, and are less than the allowable UC of 1.0 in accordance with ANSI/AISC 360-05. The maximum UC check represents the value produced by the interaction of the axial and bending stresses, which is the factored ratio of actual to allowable stress, or demand versus capacity. The shear UC represents a similar ratio based on the shear provisions of the design code.

The seismic analysis of these stacks used response spectra analysis method (RSA). In the RSA procedure, each of the model's modes is considered to be an independent single degree of freedom (SDOF) system. These modal responses are then combined to obtain the model's overall response to the applied spectra. The analysis results such as forces, deflections and reactions are determined based on the load combination including the directional RSA results. For these stacks, seismic accelerations have been applied simultaneously in three orthogonal directions, X, Y and Z, with Y-axis being a vertical axis aligned with the stack longitudinal axis.

The three loading components are combined with other load cases (dead load) assuming that one component is at its maximum while the other two components are at 35% of their respective values. In addition, the sign of each component has been selected such that the combination provides the most critical seismic loading case. The seismic analysis for the above deck equipment is typically based on the extreme level acceleration loads. This means that all the connecting parts, specifically the hold-down anchorage components should resist the fracture at this level of acceleration.

Seismic analysis showed that maximum duct deflections are between 0.22 and 0.25in, which are within the acceptable limits for both lined and unlined stack sections. Maximum allowable deflections are limited to 1in per every 600in of the stack height, as recommended by ASME STS-1, and industry practice (Bednar, Pressure Vessel Design Handbook).

Principal and Von Mises stresses from the seismic analysis were lower than the allowable stresses of the material design properties provided in Table 1. Maximum Von Mises stresses are calculated to be in the range between 9 and 12 ksi and are located at the lined duct support lug sections. These stresses are lower than the allowable stresses of 14 ksi calculated from the material design properties provided in Table 1.

Stress concentrations of 18 to 20 ksi exceeding the allowable stresses (at or adjacent to the support lugs) are mitigated by adding structural re-pads on the shell. The re-pads are designed so they cover sufficient area with peak stresses around the critical support locations, and help to lower these stresses to allowable stresses of 14 ksi. In addition, shell re-pads help with joint distortions and excessive joint rotations during extreme loading events. Maximum calculated unity check (UC) in bending and shear is observed at flanges (beam elements). The observed values are 0.36 respectively 0.42, and are less than the allowable UC of 1.0 in accordance with ANSI/AISC 360-05, (Figure 6).

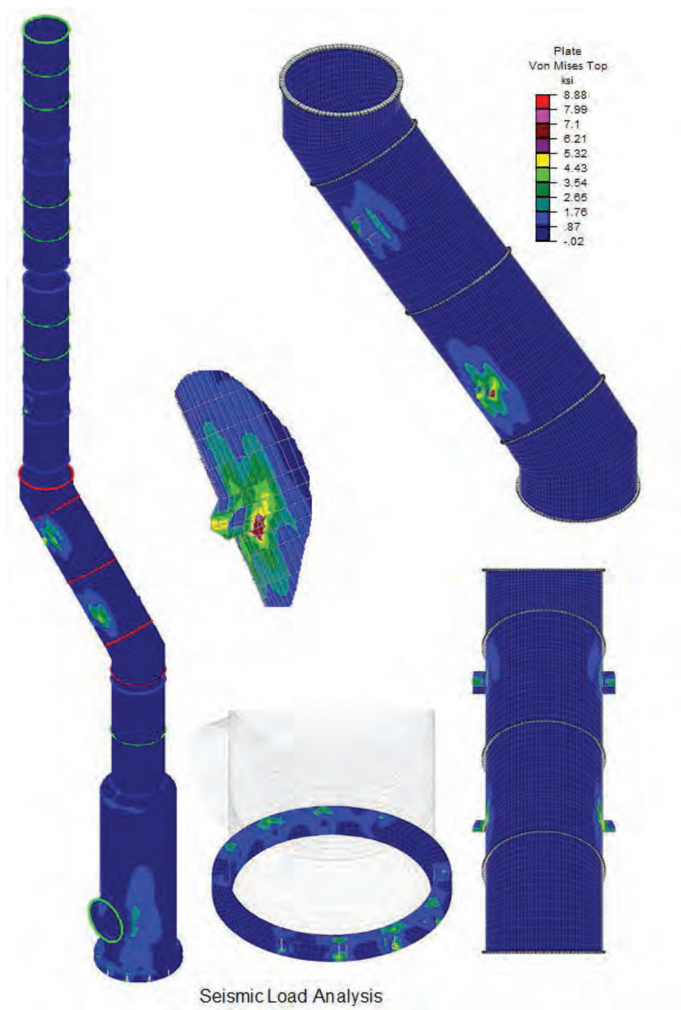

Fig. 6. Schematic view of the seismic analysis results for partially slanted vertical stack system. Note stress concentration at the shell support lugs and base ring anchor bolts.

The boundary conditions for the thermal analysis assumed that sliding can occur at the supports via slotted bolt holes. It is assumed that thermal loads and stresses for the gravity and dynamic analysis (wind and seismic) are relatively small compared to the self-weight and not sufficient to overcome friction forces at the supports, therefore the supports are modeled as pinned, (Figure 7).

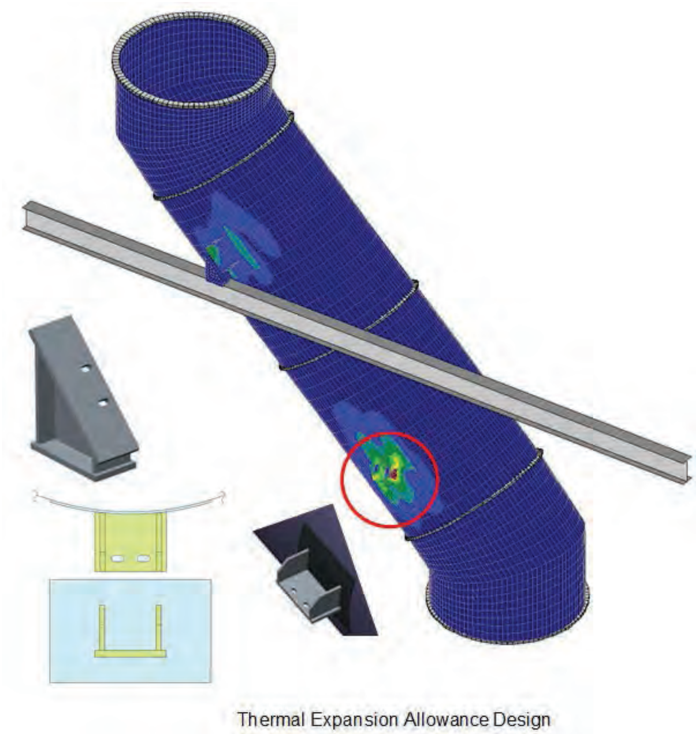

Fig. 7. Schematic view of the slanted duct support lug configurations. Note slotted bolt holes on the support lug and welded "I" beam seat to allow thermal expansion at the support locations.

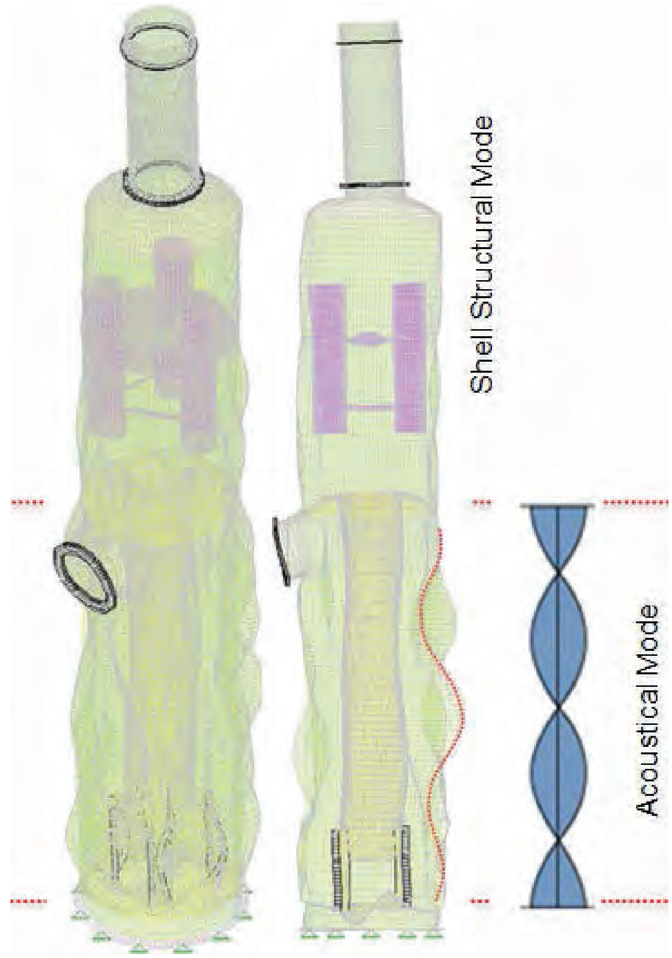

Fig. 8. Silencer structural resonant frequencies close to the dominant firing frequencies and acoustic modal frequencies.

Exhaust Stack System Structural Analysis – Numerical Example 2

The independent dynamic analysis using RSA is a useful tool to analyze systems that are subjected to special loading conditions such as flow induced vibration. A different example of an unlined stack systems experiencing shell cracking due to the combined vibrations caused by the exhaust gas flow, dominant firing frequencies and acoustic modal frequencies are presented. Similar FE modeling and analysis approach as in the previous example has been used in this system as well. In addition to RISA 3D software package, CREO 2 Simulate has been utilized to model and analyze the internal parts of this silencer system. Figure 8 shows an unlined silencer/ stack system where acoustical resonances of the silencer chamber coupled with the significant structural resonances of the shell at the dominant engine order frequencies resulted in violent vibration levels, that ultimately caused shell cracking.

Modal analysis of the system was performed to determine which part(s) of the system were sensitive to shell structural resonant frequencies that were close (aligned) with the dominant firing and acoustic modal frequencies. In order to determine the origin and cause, a detailed FEA modeling and modal analysis of the system including internal components has been performed. The measured engine firing frequencies for this system are 45, 90 and 135 Hz.

The FE analysis resulted in frequencies of 43, 86 and 130 Hz for the silencer's main chamber and shell. These frequencies are close to the engine firing frequencies, which would potentially produce the resonant frequency that consequently leads to excessive vibrations and failures of system components.

Modal analysis showed that the re-design of the internal acoustical chamber and increase of stack stiffness would eliminate shell cracking due to excessive vibrations. The first measure to eliminate these vibrations included installing an additional domed head to de-tune acoustic modes and increase the system stiffness. This measure also included designing the internal tubes from solid sheet instead of perforated plates which increased the mass and stiffness of internal parts. In addition to reducing the length of the 1st silencer chamber, the shell thickness was increased at the base, a domed head, and gussets were installed between the head and stack contributing significantly to the stiffening of the entire system. Modal analysis showed that the new silencer design has acoustic modes and structural resonant frequencies that are not close (not aligned) with the dominant firing frequencies, which was the objective of this study (Figure 9).

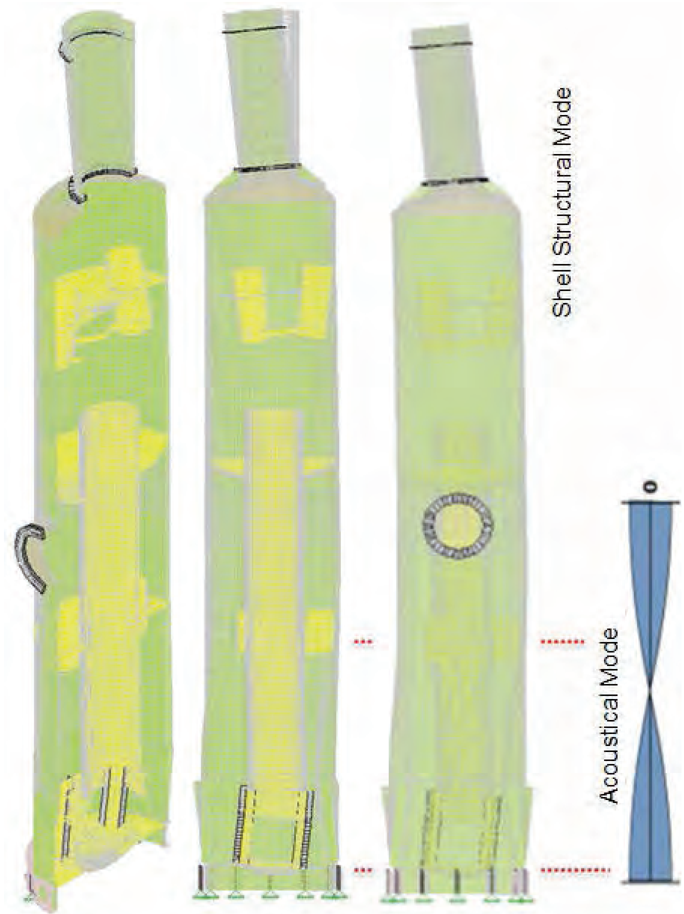

Fig. 9. Improved silencer design including acoustic and structural resonant modes/frequencies not aligned with the dominant firing frequencies.

CONCLUSIONS

This paper details the analysis process and results for development of an optimal silencer and duct system from an internal pressure, wind, seismic and aero-acoustic perspective. The FEA method is utilized to determine the appropriate duct system shell thicknesses, support locations, and expansion joint locations / requirements. Specific consideration for the finite element type (beam or plate), material and mass modeling are discussed. Simulation of various load cases is considered in the evaluation of the exhaust systems response to wind and seismic inputs. In addition, the loads and stresses on the supporting structures have been reviewed.

To demonstrate the proposed modeling and analysis method two numerical examples of common offshore topside stack systems are presented. The first example analysis suggests that the design wind load often exceeds the critical wind speeds due to vortex shedding. In this example the top duct sections experienced wind induced vibrations that required using strakes on the system top sections.

Seismic analysis showed that stress concentrations are highest at the support and connection locations. These stresses require additional strength and stiffness that is accomplished by adding structural re-pads at the critical locations.

Modal analysis can further be performed to examine the impact of wind and seismic loads, as well as the flow induced vibrations. The second example shows utilization of modal analysis to determine the shell cracking due to the resonant frequencies and resolution of the problem. The FE analysis showed that it is possible to eliminate the resonant frequencies by combination of increasing shell thickness and installing an additional domed head to de-tune acoustic modes and increase the system stiffness.

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