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PIPING FORCES AND MOMENTS ON COMPRESSORS AND STEAM TURBINES

by
Katsuki Yagi
Senior Research Engineer
Mitsubishi Heavy Industries, Ltd.
Takasago, Japan

Atsushi Higashio
Senior Engineer
and
Hirokazu Kawashima
Senior Engineer
Mitsubishi Heavy Industries, Ltd.
Hiroshima, Japan

Katsuki Yagi is a Senior Research Engineer of the Turbomachinery Laboratory at Mitsubishi Heavy Industries, Ltd., in Takasago, Japan. He has been involved in the design and development of turbomachinery compressors and expanders as well as combustors and chemical reactors. He is currently engaged in aerodynamic design and testing of special purpose turbomachinery.

Mr. Yagi holds B.S. and M.S. degrees from the University of Tokyo in Japan and an MBA from Bond University in Australia.

Atsushi Higashio is a Senior Engineer of the Compressor Design Section at Mitsubishi Heavy Industries, Ltd., in Hiroshima, Japan. He is engaged in the design and development of centrifugal compressors for use in the petroleum industry in accordance with API 617.

Mr. Higashio holds B.S. and M.S. degrees (Mechanical Engineering) from Kyoto University in Japan.

Hirokazu Kawashima is a Senior Engineer of the Turbine Design Section at Mitsubishi Heavy Industries, Ltd., in Hiroshima, Japan. He has more than 15 years of industrial experience. He is currently responsible for the design and development of special purpose steam turbines for the petroleum industry in accordance with API 612.

Mr. Kawashima holds a B.S. degree (Mechanical Engineering) from Setsunan University in Japan.

ABSTRACT

At recent chemical plants and liquefied natural gas (LNG) plants, the increase of plant capacity has caused the size of turbomachinery to increase and larger nozzle sizes are being utilized. Especially for offshore facilities, a simpler piping arrangement is required despite the larger capacity trend because the space around the turbomachinery is constrained and therefore the piping design allows for less flexibility in its arrangement. Additionally, the heavier total weight of the turbomachinery and the larger size of piping nozzles make it more difficult for plant contractors to handle external piping forces and moments. Although NEMA SM 23 (1994) or API Standard 617 (2002) have been conventionally referred to for allowable loads in turbomachinery piping arrangement designs, the increase in plant capacity and piping size with larger machine installations for offshore facilities has heightened the need to relax this specification.

In this study, the possibility of relaxing the maximum allowable load on centrifugal compressors and steam turbines is investigated based on the general design philosophy of turbomachinery, the authors' experience and analysis of quantitative influence using finite element method (FEM) of typical machines with a piping load of 1.85 times the allowable limit specified by NEMA SM 23 (1994). Various key components including nozzle deformation, stress level, clearance and misalignment of shaft ends are examined and, as a result, a bottleneck (i.e., the limiting factor) which should be taken into account for centrifugal compressors and steam turbines, is clarified in the case of excess piping load. Finally, applicable solutions from the machine manufacturer's side are suggested in order to relax this specification for piping loads on turbomachinery.

INTRODUCTION

Background

The allowable piping load on turbomachinery is an important issue to be agreed upon between plant constructors and machinery manufacturers. Since API Standard 617 (2002) stipulates that 1.85 times the NEMA SM23 (1994) value should apply for the allowable piping load on compressor casing nozzles, this limit has been conventionally proposed as the allowable piping forces and moments on turbomachinery. As is often the case with this topic, relaxation of the criteria is requested at later stages of projects by plant constructors, which causes exhausting discussions. It is not an exaggeration to say that the tendency of the oil and gas industry to value reliability based on past experience has made machine manufacturers quite conservative and cling to international criteria. In addition, it is not preferable to use different limits for various cases from the viewpoint of modular design because it may increase the number of design criteria and cause superfluous complexity.

New applications such as floating LNG (FLNG) or mega plants, however, seem to need higher allowable limits due to limited piping space. It can be a great opportunity as well as an ordeal for our business to meet such challenging requests from our customers. This is the motivation of this study and suggestions to solve this question based on bottleneck analyses are provided in this paper.

Purpose of this Study

The goal is to elicit solutions for the relaxation of the allowable limits of piping forces and moments to meet severer requirements in new fields such as FLNG from a technical point of view instead of disputing contractual agreements. For this purpose, it is important to clarify what should be considered as key issues for this problem, what the bottleneck is and how the manufacturers should modify their turbomachinery designs. It is hoped that this will encourage discussion of practical and viable solutions between manufacturers and plant constructors.

Standard Requirement on Piping Load

NEMA SM23 (1994) is the standard code that covers single stage and multistage mechanical drive steam turbines. API Standard 617 (2002), the standard code for centrifugal compressors for petroleum, chemical and gas service industries, quotes NEMA SM23 for the stipulation of piping loads and recommends 1.85 times the allowable limit of NEMA SM23. According to API Standard 617 (2002), the forces and moments acting on compressor casing nozzles should normally be limited as below.

- Total resultant forces and moments should not exceed the limitation of any nozzle:

$$+1.09M_R \leq 54.1D_e \quad (1)$$

- The combined resultants of the forces and moments of nozzles resolved at the centerlines of the largest connection should not exceed the limitations:

For resultants:

$$F_C+1.64M_C \leq 40.4D_C \quad (2)$$

For components:

$$F_x \leq 16.1D_C,\ F_y \leq 40.5D_C,\ F_z \leq 32.4D_C \quad (3)$$

$$M_x \leq 24.6D_C,\ M_y \leq 12.3D_C,\ M_z \leq 12.3D_C \quad (4)$$

where, F_R and M_R mean resultant force and moment, F_C and M_C are the combined resultant of inlet, sidestream, and discharge forces and moments, D_e is equivalent pipe diameter of the connection, D_C is diameter of one circular opening equal to the total areas of openings. F_x, F_y, F_z, M_x, M_y and M_z show the individual components of resultants. The constants are "1.85 times NEMA."

Since there is no specified stipulation on relaxation for steam turbines like compressors, the same, 1.85 times NEMA criteria, have been proposed for mechanical drive steam turbines as compressors due to the uniformity of train design.

APPROACH

What determines the allowable limit of piping forces and moments on turbomachinery? It seems quite difficult to answer this question because the bottleneck may depend on various factors. Although the products belong to the categorized design models, every machine is different because there are many different combinations of machine type, size, design pressure and temperature, nozzle location, service and so on. In addition, all the directions and distributions of piping forces and moments during the detailed design of the turbomachinery are assumptions as the directions and distributions are normally specified by the plant contractor at the later stages of projects.

In this study, all the possible risks are listed and examined to know how much influence each risk has on turbomachinery design, which risk can be a bottleneck and how the bottleneck can be removed. The following five possible risks are considered in this study:

1. Lift condition
2. Contact condition
3. Excess misalignment
4. Excess stress of casing
5. Gas leakage

First of all, no lift condition of turbomachinery shall be one of the criteria because no vertical movement upward is presumed in the design. If lift condition should happen, a lot of unexpected phenomena would arise and reliability would be no longer assured. Secondly, since piping load causes deformation of the casing, contact between moving parts and static parts should also be avoided. Contact condition is most likely to occur at labyrinth seals. Thirdly, misalignment should be within the mechanical limit of coupling. Fourthly, the stress of casing materials caused by piping load must be considered because the limit depends on the material used. Finally, the possibility of gas leakage at flanges should be investigated for horizontal split type compressors.

For lift condition evaluation, a statistical approach is adopted with a simplified model and thus the result can be used as empirical knowledge. For the other four risks, however, it is difficult to check all the possible variations. Therefore, FEM analyses for typical machines were performed as the first step in order to capture the influence of those risks and elicit the key issues to be examined carefully during further investigation.

FEM ANALYSIS

Two things should be underlined in the FEM analysis; one is the fact that this analysis is just a case study. It is important to derive implication applicable for general machines from the few results for specific machines. The other is decoupling the influence of piping loads from that of other thermal and mechanical factors. For instance, the margin for contact condition shall be shared by not only piping load but also the other factors including thermal growth and internal pressure.

The goal is not to form a database by collecting detailed behavior of specific machines, but to provide information about what may become a bottleneck and how it can be removed.

Compressor

The selected type of compressor is double-flow type, which has a fabricated casing, two 60 inch suction nozzles and one 54 inch discharge nozzle. Casing feet as well as the compressor body are considered in the calculation. The underside of the foundation is assumed to be fixed in all directions. The behavior of the compressor body and feet is calculated when piping forces and moments are applied. The direction and distribution of piping forces and moments are shown in Table 1. Individual components of the resultants are fixed at 1.85 times NEMA. The forces and moments at each nozzle are allocated in proportion to each nozzle area. The FEM analysis of the compressor was performed with the following approach (Figure 1). First, deformation characteristics were examined with single force or moment on a single nozzle. Second, several cases for combined forces and moments were simulated to check if linearity of the behavior is applicable. It is helpful to capture the whole picture of deformation in order to estimate the quantitative influence of piping load. Finally, the actual operating conditions were simulated considering the machine's own weight, thermal growth, bolting, internal pressure and piping forces and moments.

Table 1. Piping Load on Compressor Nozzles.

	Nozzle 1	Nozzle 2	Nozzle 3
F_x	5,522 N (1,241 lbf)	4,466 N (1,004 lbf)	5,522 N (1,241 lbf)
F_y	13,891 N (3,123 lbf)	11,234 N (2,526 lbf)	13,891 N (3,123 lbf)
F_z	11,113 N (2,498 lbf)	8,988 N (2,021 lbf)	11,113 N (2,498 lbf)
M_x	8,437 Nm (6,222 lbf-ft)	6,824 Nm (5,032 lbf-ft)	8,437 Nm (6,222 lbf-ft)
M_y	4,219 Nm (3,111 lbf-ft)	3,412 Nm (2,516 lbf-ft)	4,219 Nm (3,111 lbf-ft)
M_z	4,219 Nm (3,111 lbf-ft)	3,412 Nm (2,516 lbf-ft)	4,219 Nm (3,111 lbf-ft)

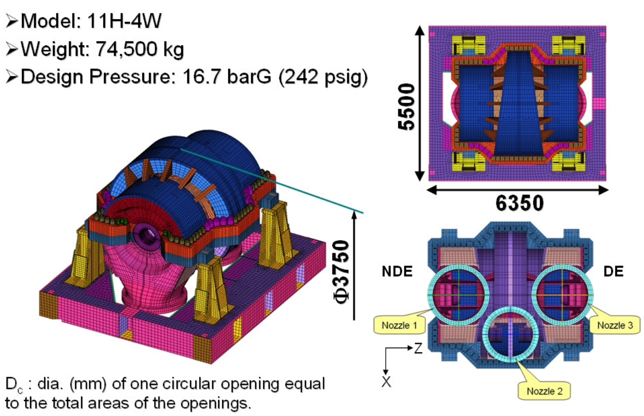

Figure 1. FEM Model for Compressor.

Steam Turbine

Two types of steam turbines were considered with an FEM analysis, those being back pressure type and condensing type. The condensing type, which is typical of larger turbines, is subject to attracting force due to vacuum pressure on the exhaust side (Table 2). This vacuum condition is different from a back pressure type, which is applied mainly to small size and lightweight turbines (Table 3).

Table 2. Piping Load on Steam Turbine (Back Pressure Type).

	Inlet	Exhaust
Fx (N)	2259.9	2607.8
Fy (N)	5589.9	6450.4
Fz (N)	4400	5078.4
Mx (Nm)	12740	3842.6
My (Nm)	5958.4	1921.8
Mz (Nm)	1043.7	1921.8

Table 3. Piping Load on Steam Turbine (Condensing Type).

	Inlet Mouth 1	Inlet Mouth 2	Extraction	Exhaust
Fx (N)	2432.4	2432.4	3128.2	9031.7
Fy (N)	6016.2	6016.2	7737.1	22342
Fz (N)	4736.3	4736.3	6090.7	17589
Mx (Nm)	-9622.6	13237.8	4609	13310.4
My (Nm)	-2828.3	6413.1	2305	6655.2
Mz (Nm)	880	880	2305	6655.2

The FEM analysis of steam turbine was performed considering turbine weight, heat and pressure load at operating conditions and piping loads simultaneously mainly in order to evaluate contact condition because it is empirically known that contact condition often takes place with an excess piping load. As to piping forces and moments on the nozzles, both distributed and concentrated conditions were simulated. The magnitude of each component was determined as 1.85 times NEMA. Figure 2 and 3 show FEM models for steam turbine.

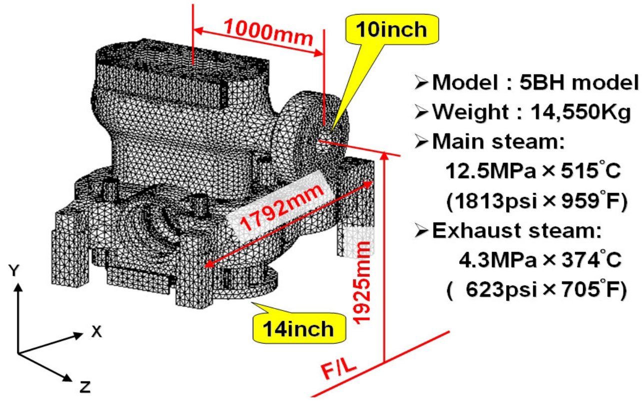

Figure 2. FEM Model for Steam Turbine (Back Pressure Type).

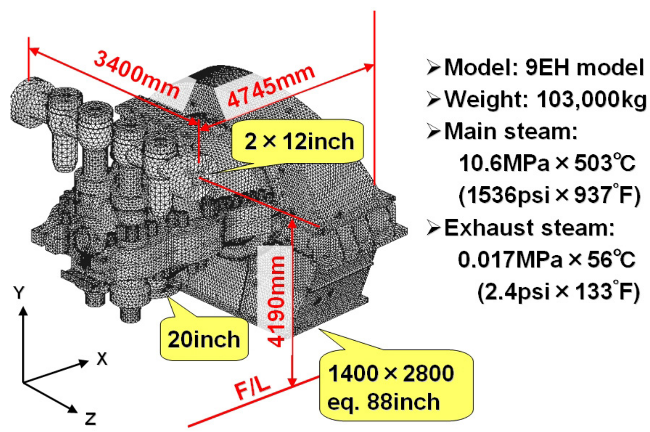

Figure 3. FEM Model for Steam Turbine (Condensing Type).

BOTTLENECK ANALYSIS

Bottleneck analyses for piping forces and moments on compressors and steam turbines were performed as per the above-described approach.

Lift Condition

By comparing machine weight and vertical component of F_C as shown in Figure 4 and 5, a critical value for lift condition can be roughly estimated. For simplification, the distance between the acting vertical force and supporting point is assumed to be twice that between the center of gravity and the same supporting point of the machines. The $21.9D_C$ ($=40.5D_C/1.85$) indicates 1.0 times NEMA of the vertical component of F_C. Multiplication constant number at critical lift condition is calculated with all the past machines. The constant X means that X times NEMA is the allowable limit for lift condition.

Figure 4 and 5 show the allowable limit of lift condition with machine size. The first finding is that larger size machines have a larger allowable limit for lift condition. Needless to say, the business principle should be recognized that machinery be designed with minimized cost to attain the required performance. Another finding is that dispersion of the critical values even for the same size arises due to various design factors including design pressure and machine type. For instance, turbomachinery for higher pressure application should have larger weight and hence a higher limit for lift condition.

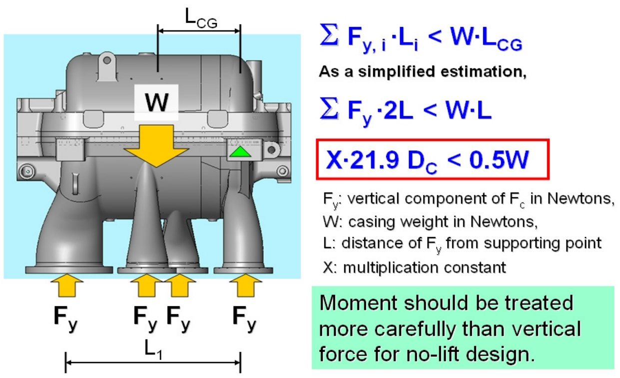

Figure 4. Model of Lift Condition for Compressor.

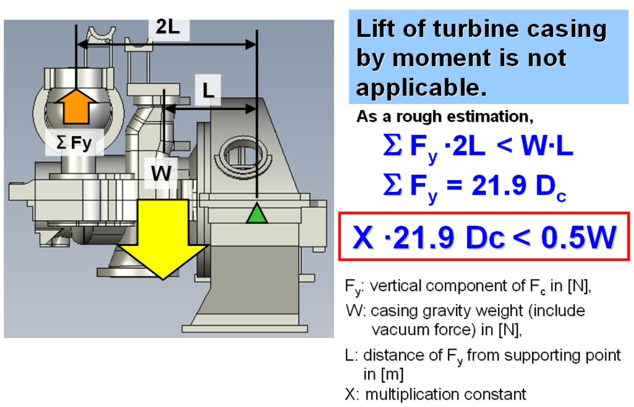

Figure 5. Model of Lift Condition for Steam Turbine.

When the universal value is applied as standardized criteria for all varieties of machine, the criteria shall be determined with small and light machines and therefore the criteria will become quite conservative. The critical values will be around 2.5 times NEMA for both compressor and steam turbine. Setting aside using the standardized criteria, however, there is room for relaxing allowable limits on lift condition especially with large and heavy machines as shown in Figure 6 and 7.

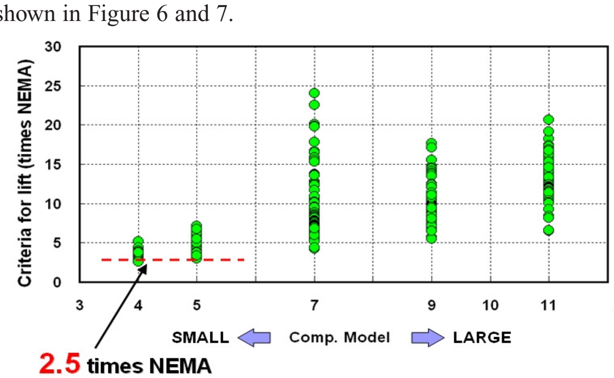

Figure 6. Critical Lift Condition for Compressor.

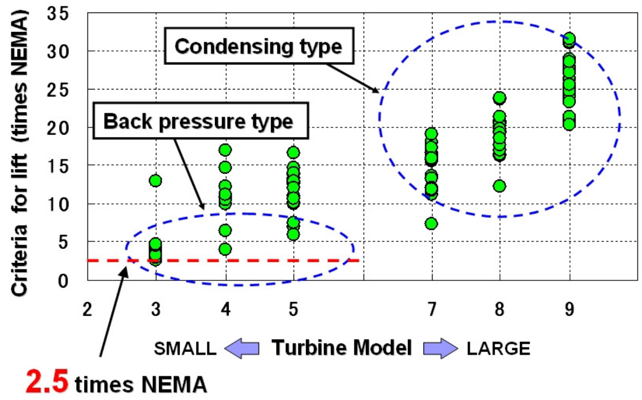

Figure 7. Critical Lift Condition for Steam Turbine.

Contact Condition

Both compressors and steam turbines are carefully designed considering the possible deformation of casing and rotor. Contact during operation would cause unexpected vibration or result in devastating damage of the machines. Contact is most likely to occur at a small clearance of the labyrinth seal structure as shown in Figure 8. The labyrinth seal structure is normally used for gas separation seals, shaft seals and interstage seals.

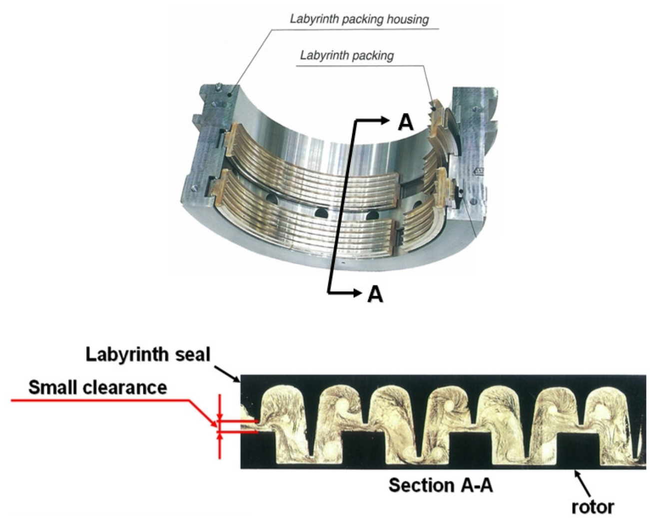

Figure 8. Labyrinth Structure (Typical).

Figure 9 shows the typical deformation characteristics of compressors caused by single piping force. The maximum deformation, which is not equal to change of clearance, seems to demonstrate that deformation depends on the direction of the applied piping load. Change of clearance can be calculated based on an estimation of temperature distribution of the casing and internal parts (diaphragm, impeller and shaft), initial bending due to rotor weight and tolerance of the assembly. The contribution of each factor on the change of clearance is shown in Figure 10. It seems that the influence of piping forces and moments on clearance change is much smaller than those of the other factors and therefore can be regarded as negligible for the centrifugal compressor. In addition, it is possible to estimate the actual stiffness values of each piping nozzle under a postulation of linearity as shown in Figure 9. It would be helpful for plant engineers to use the actual stiffness values instead of a typical stiffness value of 1.8E12 N/mm (1E12 lb/inch), which is generally used in many programs. Since such a high stiffness value means that the turbomachinery nozzle is almost rigid, many problems can be eliminated by using more realistic stiffness values.

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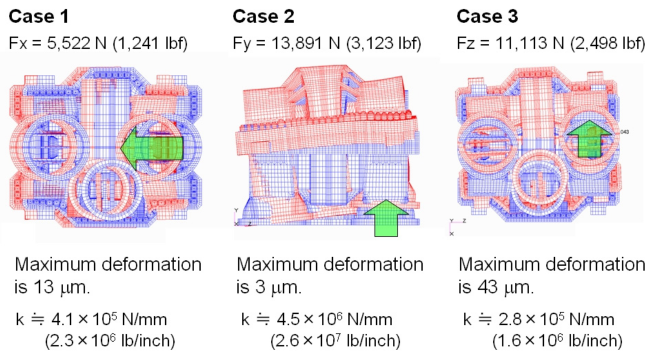


Figure 9. Typical Deformation of Compressor by Single Piping Force (1.85 times NEMA).

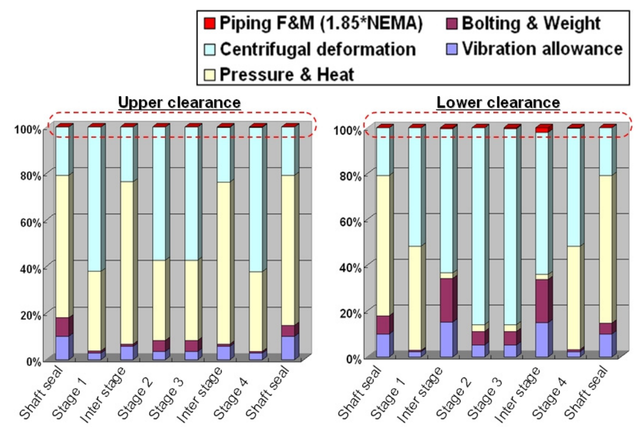


Figure 10. Contribution to the Change of Clearance (Compressor Case: Influence of Piping Forces and Moments Is Highlighted with a Dotted Line).

Evaluation of the risk of contact condition was performed for two types of steam turbine. According to the simulation results, it is found that vertical deformation is significant and thus should be observed on back pressure type steam turbines; on the other hand, horizontal deformation is more significant for the condensing type because vertical deformation is constrained by the attracting force brought on by vacuum pressure.

Figure 11 and 12 are the evaluation results for contact condition of a back pressure type steam turbine. Figure 11 demonstrates that clearance change at front gland is largest. The critical contact condition can be estimated by linear extrapolation of the result of 1.85 times NEMA. The result is presented in Figure 12. Relaxation of the allowable limit of piping load on a back pressure type steam turbine will be preceded by no lift condition first.

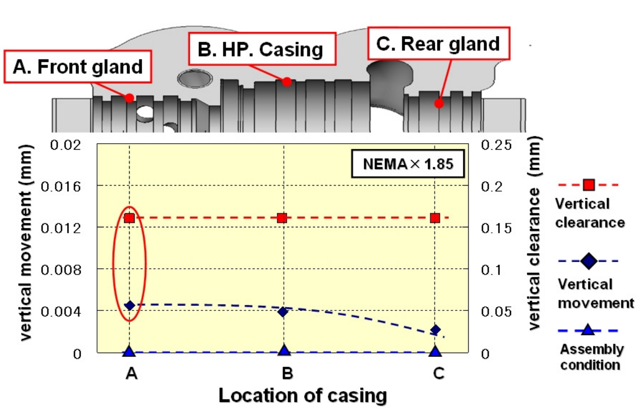


Figure 11. Clearance and its Change Due to Piping Load (Back Pressure Type Steam Turbine Case).

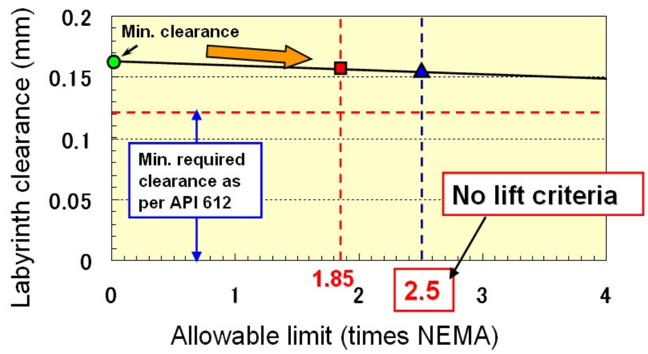


Figure 12. Critical Contact Condition (Back Pressure Type Steam Turbine Case).

Figure 13 and 14 show the results of evaluation for a large size condensing type steam turbine. The change of clearance at MP (mid-pressure) stage is the largest. In the severest case (when a concentrated load is applied), the relaxation limit appears to be up to 6 times NEMA according to linear extrapolation when the minimum required clearance is considered as per ANSI/API Standard 612 (2005).

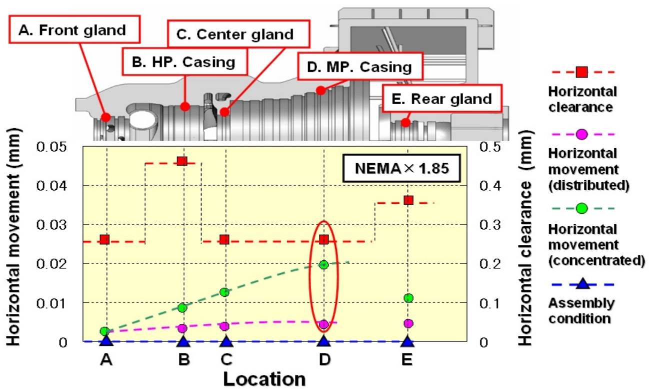


Figure 13. Clearance and its Change Due to Piping Load (Large Size Condensing Type Steam Turbine Case).

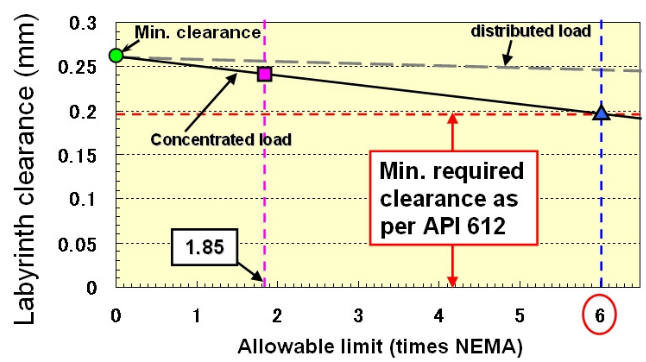


Figure 14. Critical Contact Condition (Large Size Condensing Type Steam Turbine Case).

The case study analysis for contact condition implies that contact condition can be a potential risk with large steam turbines when a relaxation of 1.85 times NEMA is attempted. Since clearance is a controllable design parameter, mitigation of the risk is possible by changing clearances in consideration of the tradeoff between clearance and performance.

Casing Stress, Misalignment, and Gas Leakage

These items are evaluated with FEM analysis for compressors in the following manner. Two cases are calculated: with and without a piping load of 1.85 times NEMA. Comparing the results of the two cases, the sole influence of piping load on turbomachinery can be highlighted. Typical results are shown in Figure 15, 16 and 17. Figure 15 shows additional stress on the compressor casing caused by piping load. Maximum stress is observed at the place where compressor body and foot are in contact. The stress level of 1 MPa is much smaller than the casing strength. Figure 16 shows additional alignment change caused by piping load. The change is not so large as to jeopardize the mechanical limit of the coupling design. Figure 17 demonstrates that surface pressure between a compressor's horizontal mating flanges is not significantly affected by piping load.

As a consequence, the influence of piping load pertaining to these items is much smaller than that of the other factors and thus seems to be negligible for large compressors when piping forces and moments of 1.85 times NEMA are applied. Since an elastic body is presumed in the FEM analysis, the influence of a larger piping load can be inferred with linear extrapolation of the result. It is implied that stress of casing, misalignment and gas leakage do not become critical for the relaxation of allowable limits on not only compressors but also steam turbines in advance of lift and contact conditions.

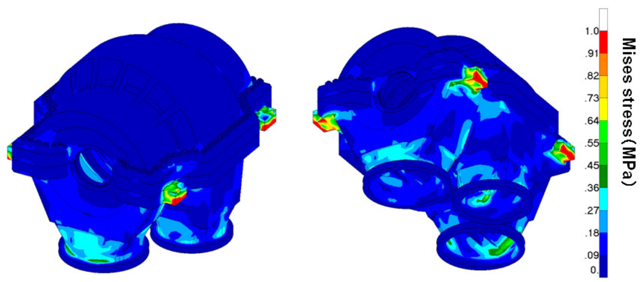
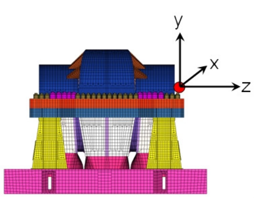


Figure 15. Casing Stress Distribution of Compressor.



	Misalignment (mm) δ_y	Misalignment (mm) δ_x
● Bolting + Weight	-0.141	-0.010
● Bolting + Weight +Pressure	-0.096	0.014
● Bolting + Weight +Pressure + Heat	-0.673	0.066
● Piping forces & moments	0.001	-0.071
Total	0.672	-0.005
Parallel offset	3.570	3.570

Figure 16. Misalignment by Piping Forces and Moments on Compressor.

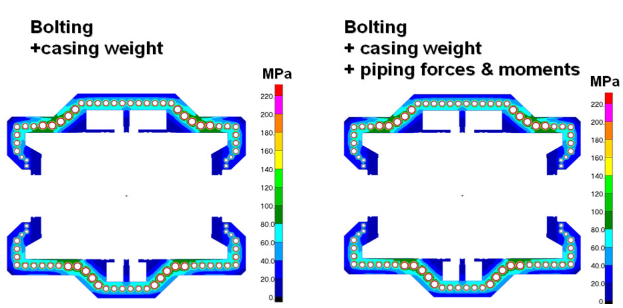


Figure 17. Surface Pressure Between Horizontal Flanges of Compressor.

SUMMARY AND SUGGESTIONS

As potential risks of excess piping forces and moments on centrifugal compressors and steam turbines, five relevant items are investigated in this study and some important information was derived as follows:

• First, it appears that the key issues are lift condition for small-size compressors and steam turbines (back pressure type) and contact condition for large size steam turbines (condensing type). The case study indicates that 2.5 times NEMA is the critical lift condition for small-size machines and 6 times NEMA for large size condensing type steam turbines. Since the FEM analysis is just a case study, it would be wrong to conclude that these results are universally valid. The results of case studies should be used to prioritize the concerned items rather than specifying critical values or preferable directions.

• Second, it is inferred that a directional selection of piping load be effective because deformability is sensitive with direction of the piping load.

Therefore, for the relaxation of the allowable piping load limit from 1.85 times NEMA, lift and contact conditions shall be carefully considered in turbomachinery design and appropriate modifications depending on the target limit shall be applied for actual practice. Such modifications are exemplified as follows:

No lift design

• Additional weight should be put at appropriate positions on the machine casing.

• Casing thickness should be increased to surpass the piping load upward at the worst-case lift condition.

No contact design

• Clearance where contact is likely to occur should be increased. Tradeoff relationships between increase in clearance and decrease in performance should be considered.

• Deformability of the machine casing and rotor is controlled mechanically to avoid worst-case contact condition.

The above-mentioned modifications are not easily accomplished on the manufacturer's side only because of the uncertainty of direction and distribution of piping load stipulated in NEMA SM23. The same problem exists on the plant contractor's side; the directions and distributions of piping load can be fully specified only after finishing entire plant engineering. If this timing mismatching problem is solved with mutual concessions and breakthrough ideas, relaxation of the allowable piping load limit would be much more attainable. With a specified piping load, it would be viable to perform a verification test in the manufacturer's test bed.

It should be added that stiffness values of each piping nozzle can be estimated by a deformation analysis under a postulation of linearity. It would be helpful for plant engineers to use these stiffness values instead of a typical stiffness value of 1.8E12 N/mm (1E12 lb/inch) in their piping design programs because such a high stiffness value is likely to cause excessive reactive forces in the piping system.

CONCLUSIONS

The 1.85 times NEMA (1994) has been conventionally proposed as the allowable limit of piping forces and moments on centrifugal compressors and steam turbines as recommended in API Standard 617 (2002). In reaction to the increasing pressure for drastic alleviation of the allowable limit of piping load on turbomachinery, possible risks were presumed and the possibility of such alleviation was examined with a statistical approach based on the general design philosophy and experience and a case study approach using FEM analyses. As a result, important implications were elicited in this study as follows:

• Lift and contact condition may be bottlenecks and should be examined as priority issues for the drastic relaxation of the allowable piping load limit.

• Directional selection of piping forces and moments would be effective for tailor made design of the turbomachinery.

- If all the directions and distributions of piping forces and moments are specified in advance of the detail design stage, the criteria would be better assured with easier tailor made design.
- Use of the actual stiffness values of piping nozzles for piping design programs would eliminate many problems caused from a postulation of de facto rigidity of piping nozzles.

In order to confirm the critical conditions, FEM analyses should be performed with the actual compressors and steam turbines. The information elicited in this study, however, can be utilized to reduce the number of items to be examined. It is hoped that this study would be of help to provide answers to the frequently asked and challenging questions about piping loads on turbomachinery.

NOMENCLATURE

F_R = Resultant force
M_R = Resultant moment
F_C = Combined resultant of inlet, sidestream and discharge force
M_C = Combined resultant of inlet, sidestream and discharge moment
D_e = Equivalent pipe diameter of the connection
D_C = Diameter of one circular opening equal to the total areas of openings
F_x = Individual component of resultant force in x direction
F_y = Individual component of resultant force in y direction
F_z = Individual component of resultant force in z direction
M_x = Individual component of resultant moment in x direction
M_y = Individual component of resultant moment in y direction
M_z = Individual component of resultant moment in z direction
L = Distance from supporting point
L_{CG} = Distance of center of gravity from supporting point
W = Machine weight
X = Multiplication factor
k = Stiffness value
δ_x = Horizontal movement at shaft end
δ_y = Vertical movement at shaft end

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