Centrifugal Compressor Train Torsional Vibration Analysis; The Effect of Shaft-Coupling Penetration Ratio

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Abstract

When designing typical rotating equipment, torsional natural frequencies (TNFs) shall be studied to identify the possible resonances. To appropriately compute the TNFs and torsional modes, the effect of the shaft penetration in the coupling hub, known as the shaft penetration factor (SPF), on the torsional stiffness calculation must be considered. In this paper, a comprehensive code for the torsional vibration analysis of a centrifugal compressor train is developed by the rotor-dynamics team of the OTCE company. The torsional analysis of an electrocompressor train is then modeled, based on API 617 criteria in order to examine the effect of the SPF value on the torsional output of the model. Two assumptions regarding the penetration of the shaft in the coupling hub are considered. Assumption #1 is "end-to-end connection of the shaft and the coupling," and assumption #2 is "1/3 penetration of the shaft in the coupling hub." The effects of these assumptions on the torsional behavior of a real centrifugal compressor train, designed by OTCE, are investigated. As the results show, considering 1/3 penetration instead of end-to-end connection causes the value of 2nd and 4th unrigid TNFs to increase by 17% and 12%, respectively. The corresponding mode shapes also show noticeable differences between the two assumptions. From the Campbell diagram, it is observed that the intersections of the 2X line with the 2nd and 4th unrigid TNFs fall near and inside the API separation margin. While modeling the compressor using the 1/3 penetration assumption, the mentioned intersections are kept away from the critical speed range. Consequently, it can be said that the flexible couplings' stiffness precise modeling is a crucial point to correctly address the compressor train torsional natural frequencies. Minor uncertainties in the SPF modeling may lead to noticeable differences in the equipment's torsional behavior.

Keywords: Centrifugal Compressor; Torsional Vibration; Flexible Coupling; Torsional Natural Frequencies.

1. Introduction

A centrifugal electrocompressor train typically consists of an electrical motor as the driver, a gear unit as the speed increaser, and a high-speed compressor as the main equipment. The gear unit is connected to the motor and the compressor by two flexible couplings (as shown in Figure 1). A train torsional natural frequency (TNF) analysis is necessary for a new compressor train design, in order to identify the location of the torsional natural frequencies relative to the excitation, in the system. Torsional natural frequencies play a key role in identifying potential resonances and designing effective vibration control strategies to prevent system failures. When the train is accurately modeled, the undamped torsional natural frequency analysis usually predicts actual train natural frequencies within a small margin of error because most equipment trains generally possess low levels of system torsional damping [1].

Wang et.al. studied the effect of variation in mass-elastic data, and presented a comparison between measured and predicted TNFs for various cases [1]. Meeus et.al. using an experimental study determined the coupling hub-to-shaft connection has a significant influence on the system's first torsional mode. By increasing the shaft penetration factor, and thus decreasing the hub-to-shaft interference, an eigenfrequency drop and a damping increase, was witnessed [2]. Venkataraman et.al. described a methodology to ensure torsional integrity in standardized Electric-motor-driven gas compressor packages [3]. Mishra et.al. analysed torsional vibrations in an open-loop VFD-induction-motor-driven compressor train and provided methods to reduce the vibrations [4]. Ma et.al. proposed a torsional vibration simulation model of the loader with variable damping and stiffness [5]. Chen et.al. conducted an optimization study of the shaft system structure and showed that the optimized shaft system increased the 1st-order natural frequency and reduced the peak velocity of the coupling in the vertical direction, improving the service life and operating reliability of the system [6]. Kapustin and Degtyareva presented main stages of torsion vibration calculation in a multi-row reciprocating compressor including natural and forced vibration analysis [7].

Requirements of a torsional system design are normally based on the API Standards (API 617 [8] and API 684 [9]). These standards require torsional natural frequencies (TNFs) to have at least 10% separation margin (SM) from any excitation frequency [1], [8], [9]. In order to appropriately compute the torsional natural frequencies and torsional modes, the penetration effect of the shaft in the coupling hub, denoted by the shaft penetration factor (SPF), on the torsional stiffness calculation must be taken into consideration. There are numerous published criteria and presumptions for the flexible couplings' stiffness modeling. Modeling flexible couplings' stiffness is essential to accurately simulate the interaction between shafts and components [2].

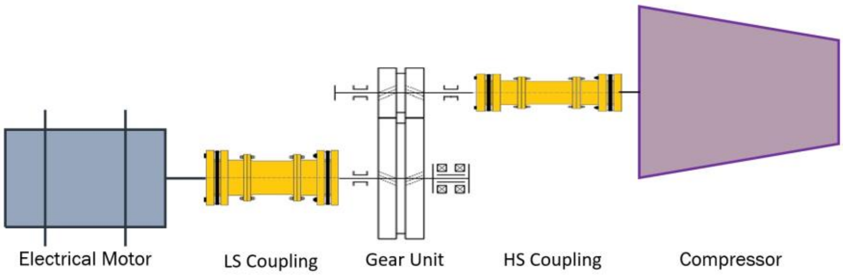

Figure 1. A schematic view of a centrifugal electrocompressor train. The Low Speed (LS) coupling connects the driver to the gear unit, and the High Speed (HS) coupling connects the gear unit to the compressor.

In this paper a comprehensive code for torsional vibration analysis of a compressor train is developed, by the rotor-dynamics department of Oil Turbo Compressor Equipment (OTCE) company. The code is developed and validated based on the literature on the physical modeling of the rotary equipment torsional behavior [10]–[13]. Then, in order to study the effects of SPF value on the equipment torsional behavior, the torsional analysis of an electrocompressor train is performed, using two assumptions for the penetration effect of the shaft in the coupling hub, as follows:

- Assumption #1: “End-to-end connection of the shaft and the coupling”
- Assumption #2: “1/3 penetration of the shaft in the coupling hub”.

The effects of these assumptions on the torsional behavior of designed compressor train, i.e., undamped torsional natural frequencies and the corresponding mode shapes are investigated.

2. Materials and Methods

2.1 Shaft-coupling connection

When conducting train torsional analysis of rotating equipment, using the lumped inertia-stiffness model, one common concern is the modeling of shaft end connection to the coupling hub. Different vendors provide the torsional stiffness differently. Figure 2 shows different ways that torsional stiffness is provided. Figure 2 (a) shows the use of torsional stiffness: from the last inertia location to the start of the coupling. Figure 2 (b)-(e) show different ways that torsional stiffness is provided. Most coupling manufacturers consider the shaft end as part of coupling when calculating the coupling stiffness. This is called the 1/3 penetration rule, as shown in Figure 2 (e) [1]. The SPF of 1/3, illustrated in Figure 2 (f), is the percentage of the shaft length within the confines of the coupling hub that is assumed to be free from restraint at the hub-to-shaft interface. Therefore, the shaft length is effectively increased by the amount of the SPF. Conversely, the length of the coupling hub bore is reduced by the same amount. Based on the hub-to-shaft connection, part of the shaft can twist unrestrained in the coupling hub bore forming a friction interface [2].

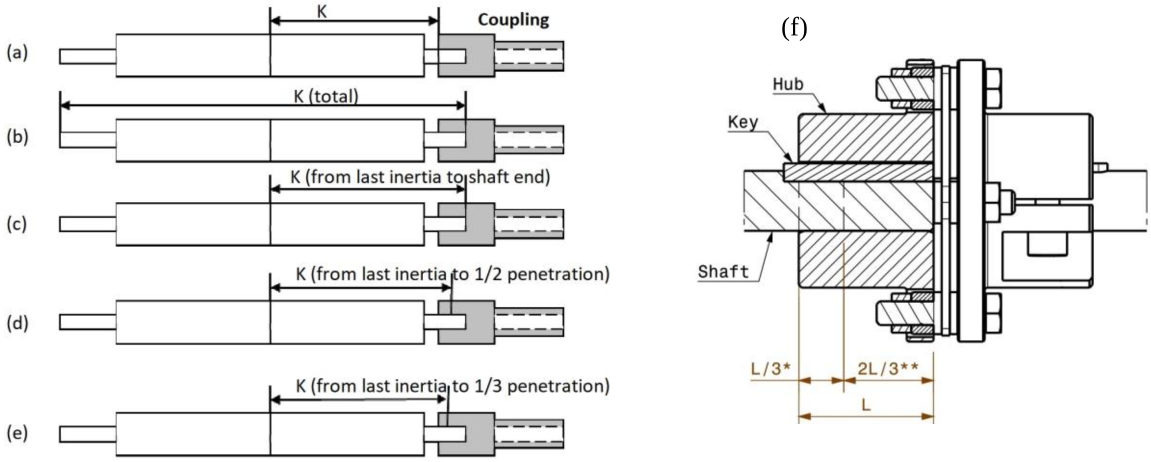

Figure 2. (a) to (e): Different methods for considering shaft to coupling connection torsional stiffness [1]. (f) SPF of 1/3. Shaft twists unrestrained for L/3 and is rigidly connected to the hub for 2L/3 [2].

2.2 Electrocompressor Train

The side view of the compressor rotor, considered in this paper, are shown in Figure 3. This compressor is designed and manufactured at OTCE company. The compressor train torsional model is developed as shown in Figure 4. The train consists of two branches, i.e., low-speed (LS) branch and high-speed (HS) branch. Low-speed branch includes the driver (E-motor), LS coupling, gearbox LS shaft, and the bull gear, as shown in Figure 4. High-speed branch includes pinion shaft, HS coupling, and the compressor elements. Different lumped masses along the rotor (like compressor impellers) are modeled as lumped inertias without stiffness (note the solid circles in Figure 4).

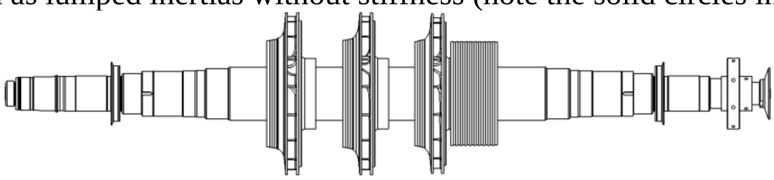

Figure 3. The side view of the compressor rotor, considered in this paper.

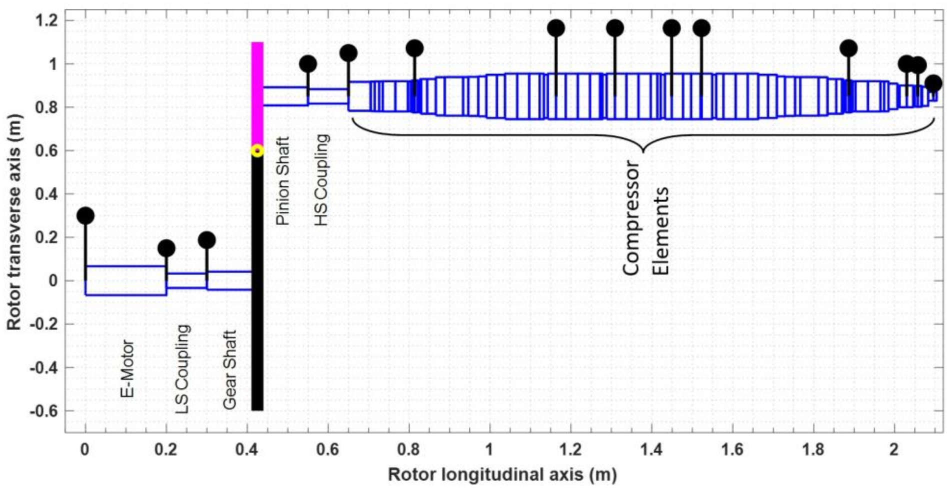

Figure 4. The developed compressor train torsional model, using the developed code at OTCE.

The torsional stiffness of couplings (provided by the manufacturer considering shaft for first 1/3 penetration) and other elements, along with the moment of inertia data are presented in Table 1. Rated speed and operating speed range for each branch is presented in Table 2. The compressor is modeled, based on this data, using the developed code for torsional vibration analysis, at OTCE.

Table 1. Torsional stiffness and moment of inertia of the different components of the compressor train.

Component	Torsional Stiffness (MNm/rad)	Moment of Inertia ($kg.m^2$)
LS coupling	3.2	2.4 (Motor side), 1.1 (Gear side)
HS coupling	1.0	0.2 (Gear side), 0.2 (Compressor side)
Gear shaft	13.8	120.5
Pinion shaft	4.3	0.4
E-motor	6.4	135.4
Compressor	-	0.6

Table 2. Compressor train branches rated speed (operating range is 80% to 105% of these speed values)

No. of Branches	Components	Rated Speed (rpm)	Operating speed range
1	E-motor, LS coupling, Gear shaft	2844	80% to 105% of the rated speed
2	Pinion shaft, HS coupling, Compressor	13392	80% to 105% of the rated speed

3. Results and Discussion

3.1 Torsional Natural frequencies

The compressor undamped torsional natural frequencies are listed in Table 3, based on two mentioned assumptions. As the results show, 1st, 2nd, 4th, and 6th TNFs obtained from two assumptions, are similar to each other. On the other hands, two different assumptions led to different 3rd and 5th TNFs. When the compressor train is modeled considering 1/3 penetration assumption, the value of 3rd and 5th TNFs increased from 6696 rpm and 24928 rpm to 8048 rpm and 28238 rpm, respectively. The reason of increasing these TNF values seems to be that considering 1/3 penetration assumption instead of end-to-end connection between shaft and coupling causes shorter coupling length and hence increasing the stiffness and the corresponding natural frequency.

Table 3. The compressor undamped torsional natural frequencies, based on two different assumptions

#	Compressor train is modeled considering Assumption #1 (End-to-end connection assumption)	Compressor train is modeled considering Assumption #2 (1/3 penetration assumption)	Difference (%)
1st Natural frequency (rpm)	0	0	0
2nd Natural frequency (rpm)	1530	1531	0
3rd Natural frequency (rpm)	6696	8048	17
4th Natural frequency (rpm)	18207	18208	0
5th Natural frequency (rpm)	24928	28238	12
6th Natural frequency (rpm)	38744	38744	0

3.2 Torsional Mode Shapes

The first four unrigid torsional mode shapes are presented in Figure 5 to Figure 8.

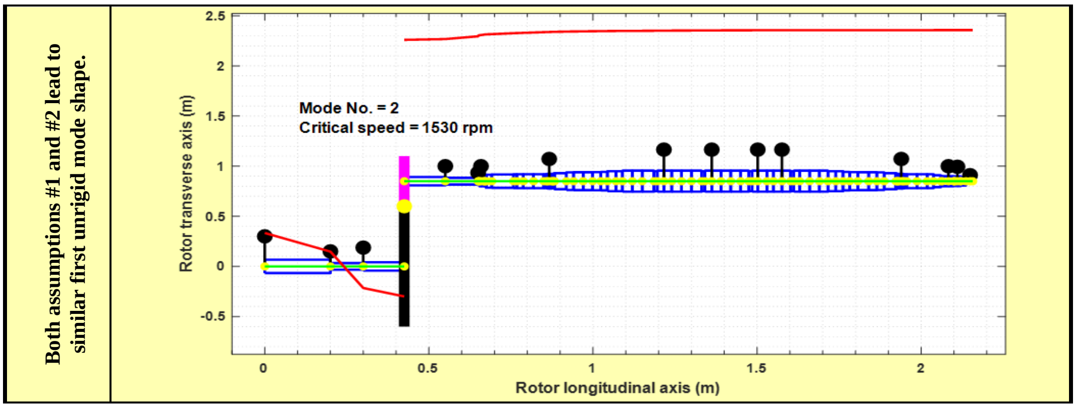

Figure 5. First unrigid torsional mode shape; corresponding to the 2nd TNF of Table 3.

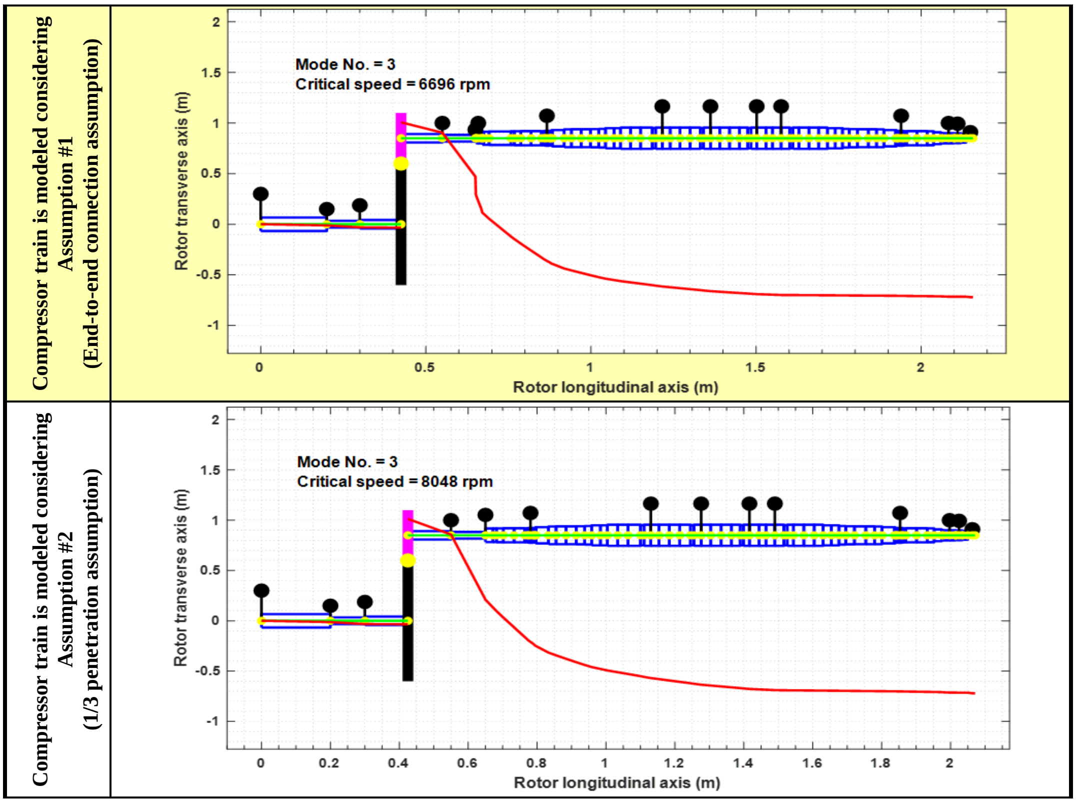

Figure 6. Second unrigid torsional mode shape; corresponding to the 3rd TNF of Table 3.

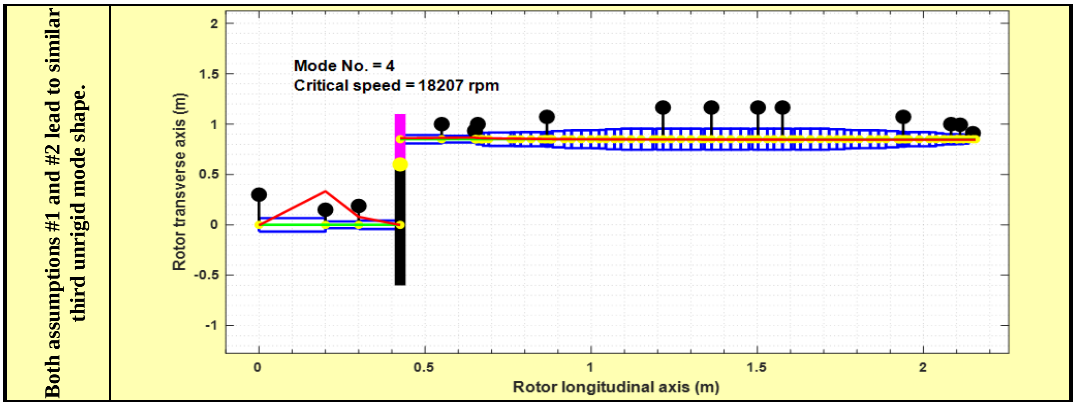

Figure 7. Third unrigid torsional mode shape; corresponding to the 4th TNF of Table 3.

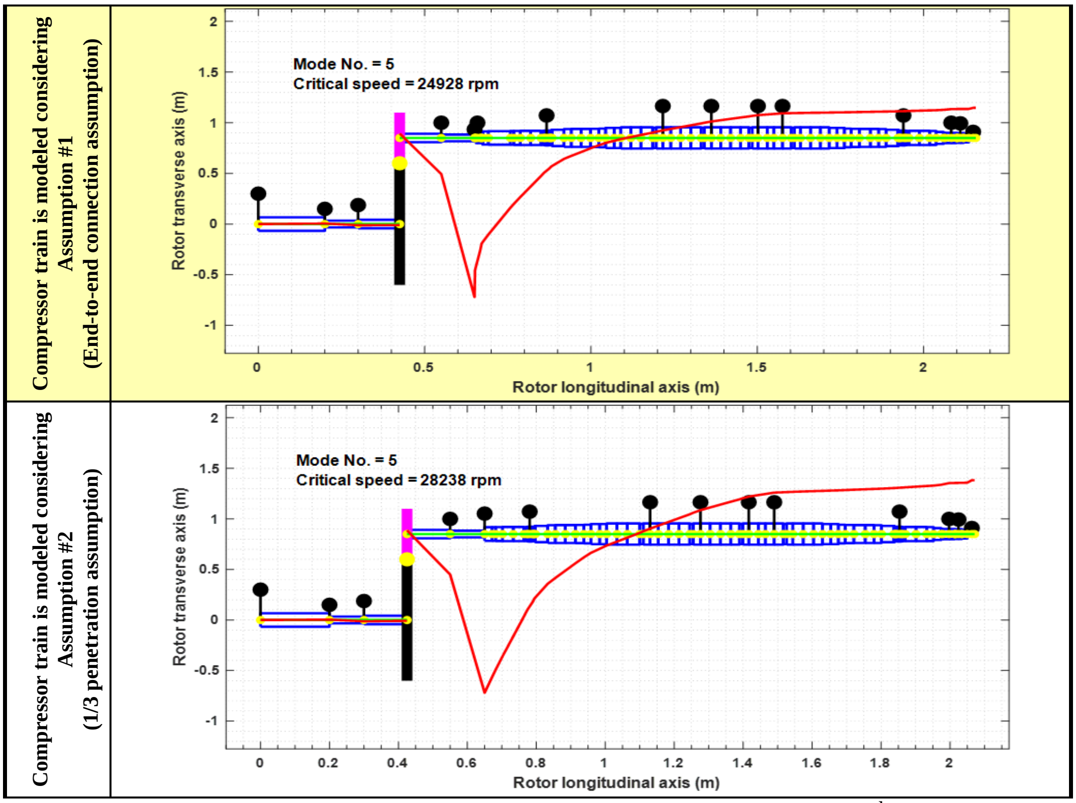

Figure 8. Fourth unrigid torsional mode shape**;** corresponding to the 5th TNF of Table 3.

The torsional mode shapes are shown in solid broken lines, along the rotor longitudinal axis. The mode shapes of Figure 5 and 7 show no difference between two assumptions. Their corresponding TNFs were almost similar, as mentioned above. The mode shapes of Figure 6 and 8 show noticeable difference between two assumptions. The corresponding TNFs were different, as well.

3.3 Campbell Diagram

In this section the Campbell diagram is analysed to observe the effects of two assumptions more comprehensively. Figure 9 shows the Campbell diagram, developed by the in-house torsional vibration analysis code. The horizontal axis shows the speed in rpm. The first four unrigid TNFs are specified on the vertical axis (see the figure legend and four horizontal lines in the plot). Two vertical solid lines shows the rated speeds of the LS and HS branches of the compressor train (see Table 2). Two vertical dashed lines specify the operating speed range of LS and HS branch, i.e., 80% to 105% of the rated speed. Two diagonal lines are 1X and 2X compressor operation lines.

The intersection of diagonal lines with each of the horizontal lines, demonstrates a speed, that may be a cause of concern. According to API 617 “the intersection of the primary (coupling) modes with the 1X mechanical excitation shall be at least 10 % above or 10 % below the specified operating speed range”. The range of 10 % above and 10 % below the specified operating speed range is shaded in the Campbell diagram (see the shaded areas in Figure 9). As the results show, neither in the 1st assumption nor in the 2nd assumption, the intersections of 1X line with the first and second natural frequencies (the primary critical speeds) do not fall within the shaded area. It can be interpreted that the compressor train meets API torsional requirements.

To investigate the effect of two assumptions more deeply, we examine 2X line and its intersections with the presented natural frequencies. As the results show, when the compressor train is modeled based on assumption #1, the intersection of 2X line with the second natural frequency falls on the border of the shaded/critical area (note the left arrow in the top image of Figure 9). On the contrary, modeling the compressor using the assumption #2 keeps the intersection of 2X line with the second natural frequency away from the shaded area (note the left arrow in the bottom image of

Figure 9). Moreover, assumption #1 causes the intersection of 2X line and the fourth natural frequency fall in the compressor operating range, while assumption #2 moves this intersection away from the operating range (note the right arrows in the top and bottom images of Figure 9).

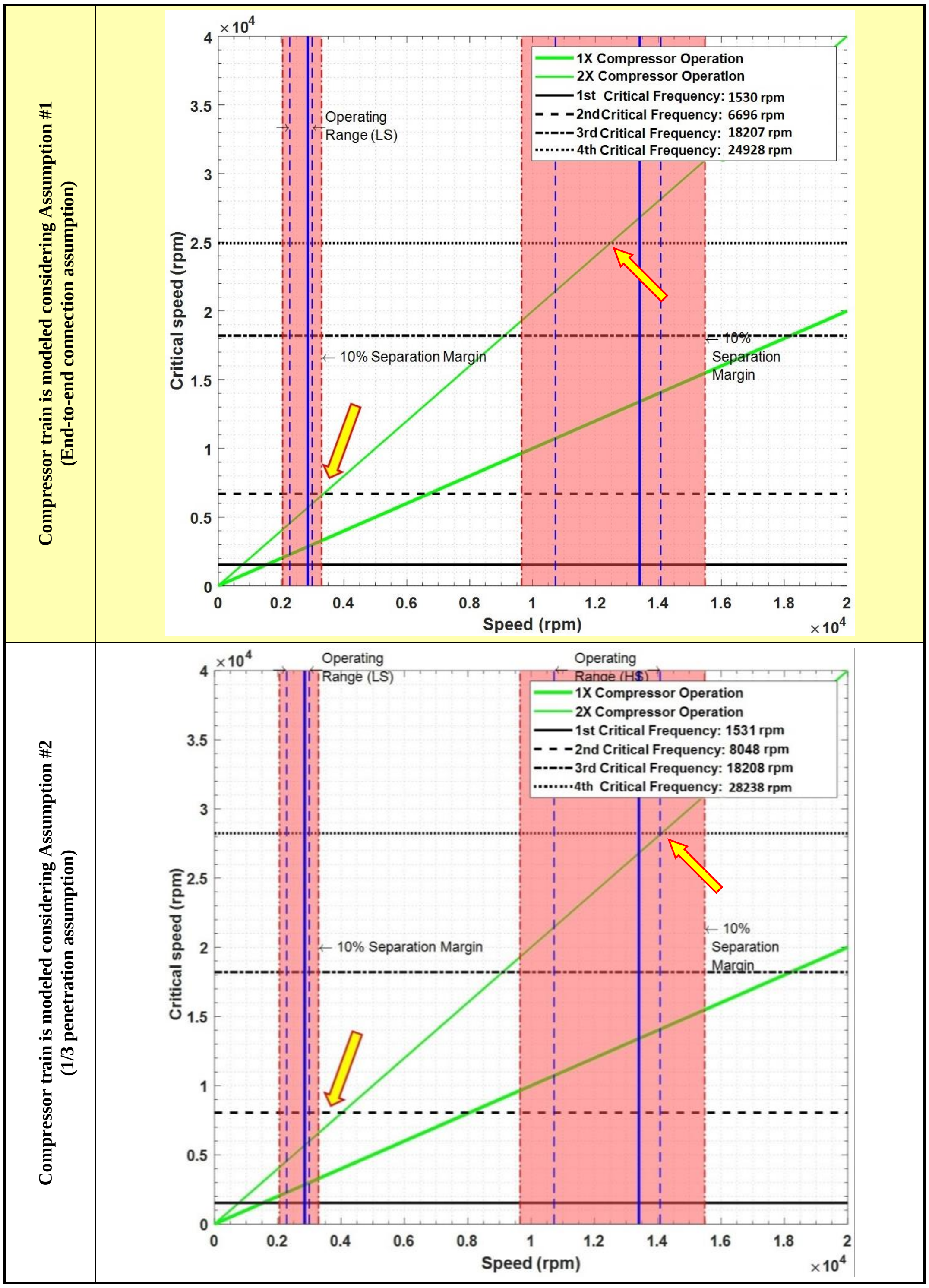

Figure 9. The Campbell diagram showing the torsional behavior of the compressor train.

4. Conclusion

A new compressor train design requires a train torsional natural frequency (TNF) analysis. The rotor-dynamics department of OTCE company developed and validated a comprehensive code for torsional vibration analysis of a compressor train. The torsional analysis of an electrocompressor train is then carried out to investigate the effects of the shaft penetration factor (SPF) on the torsional behavior of a real compressor train, designed by OTCE, based on the API-617 criteria. Two assumptions are made regarding the SPF value. Assumption #1 is “end-to-end shaft-coupling connection” and assumption #2 is “1/3 penetration of the shaft in the coupling hub”. The associated mode shapes and undamped TNFs are investigated. As the results show, when the compressor train is modeled considering 1/3 penetration assumption, the value of 2nd and 4th unrigid TNFs has increased by 17% and 12%, respectively. Considering 1/3 penetration assumption instead of end-to-end connection causes shorter coupling length and hence increases the stiffness and the corresponding TNF. The corresponding mode shapes also show noticeable differences between the two assumptions. Other TNFs and mode shapes were not affected by changing the shaft-to-coupling connection modeling. Campbell diagram shows that when using assumption #1 the intersection of 2X line with the 2nd TNF falls on the border of the API 617 separation margin. While, modeling the compressor using assumption #2 keeps this intersection away from the critical speed range. Moreover, assumption #1 causes the intersection of 2X line with the 4th TNF to fall in the middle of the compressor operating speed range, while assumption #2 moves this intersection away from the operating range. To sum up, it can be said that the correct modeling of a flexible coupling stiffness is very important to correctly address the compressor train torsional natural frequencies. Minor uncertainties may lead to noticeable differences in the equipment’s torsional behavior.

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