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Signal and Harmonic Analysis of a Compressor Blade for Identification of the Nonlinear Frequency Vibration

Farhad Asadi, Gholamhasan Payganeh

***Abstract*—**High-speed turbomachine can experience significant centrifugal and gas bending loads. As a result, the compressor blades must be able to resist high-frequency oscillations due to surge or stall condition in flow field dynamics. In this paper, vibration characteristics of the 6th stage blade compressor have been examined in detail with, using 3-D finite element (FE) methods. The primary aim of this article is to gain an understanding of nonlinear vibration induced in the blade against different loading conditions. The results indicate the nonlinear behavior of the blade as a result of the amplitude of resonances or material properties. Since one of the leading causes of turbine blade failure is high cycle fatigue, simulations were started by specifying the stress distribution in the blade due to the centrifugal rotation. Next, resonant frequencies and critical speeds of the blade were defined by modal analysis. Finally, the harmonic analysis was simulated on the blades.

***Keywords*—**Nonlinear vibration, modal analysis, resonance, frequency response, compressor blade.

I. INTRODUCTION

MANY dynamical systems display nonlinear behavior that is status and loading dependent. Strictly speaking, all oscillatory systems in real engineering application are inherently nonlinear. Linear systems are only a localized linear representation of an actual nonlinear system and can approximate the nonlinear behavior which occurs in most real dynamical systems. In some applications, a linear approximation is acceptable; however, linear assumptions often vary significantly from reality and can provide misleading information about the system. Nonlinear systems show behaviors which are different from linear systems such as jump phenomena, internal resonances and chaos or multiple steady-state solutions [1]- [3].

The stiffness and vibratory properties of the vibrational system can be changed suddenly depends on the status frequency of the system. The stiffness of a given system can often be used to determine whether the system is linear or nonlinear. Stiffness can be influenced by a number of factors such as the shape or material conditions.

In our real world, many systems are subject to excessive oscillatory motion due to the resonance. Resonance is mainly caused by a complex interaction between the inertial and elastic properties of the material of the system. In order to understand vibration-related issues, it is essential to identify resonant frequencies. However, in reality, there are always random deviations in blade properties, due to factors such as manufacturing, material tolerances, and in-operation wear. This phenomenon is identified as vibration localization. In order to understand vibration-related issues, it is important to identify resonant frequencies. Thereby, modal analysis is widely used to determine the natural frequencies and mode shapes of engineering systems [4]-[7]. Also, the detection of nonlinear response of system completely and obtaining the transition frequencies which are found in the system are very important for detailed identification of system and also modified design of system.

In this paper, a detailed numerical investigation was initiated on compressor blade. The 6th stage blade compressor model was finely meshed in root and disk connection zone to accurately predict static and dynamic stresses. Once the three dimensional (3-D) ANSYS Finite Element Analysis (FEA) model was prepared and boundary conditions on blade applied, the simulation was run at maximum operating case to determine the highest static loading due to centrifugal loads. Afterwards, operational stresses in the fracture and critical area determine the most probable resonant mode that would produce the highest vibratory stress and deflection in compressor blade. The last harmonic analysis of blade was done respectively and nonlinear behavior detection of system and transition of frequency in this condition is obtained.

A. Computational Setup and Material Properties of Model

Fig. 1 shows the blade geometry and the details of the local regions of compressor blade. A 3-D FE model of a blade was created by CATIA software, as depicted in Fig. 1.

The geometry was then meshed using a tetrahedron mesh, with using 110,000 nodes, to avoid ambiguity along the blade attachment and roots and finer mesh was designated for this region, as seen in Fig. 1. Furthermore, blade is made out of titanium Ti-6Al-4v and also Table I Outlines the material properties of model.

TABLE I
MATERIAL PROPERTIES -TITANIUM TI-6AL-4V OF MODEL

Attribute	Value	Units
Young's Modulus	96	Gpa
Density	4620	$\frac{kg}{m^3}$
Poisson's Ratio	0.342	--

Farhad Asadi has Master degree in Mechanical engineering in dynamic and control, K. N. Toosi, Tehran, Iran (phone: +989357430949, e-mail: farhadasadi1990@gmail.com).

Gholamhasan Payganeh is Assistant professor in Mechanical Engineering at Shahid Rajaee Teacher Training University (e-mail: g.payganeh@srttu.edu).

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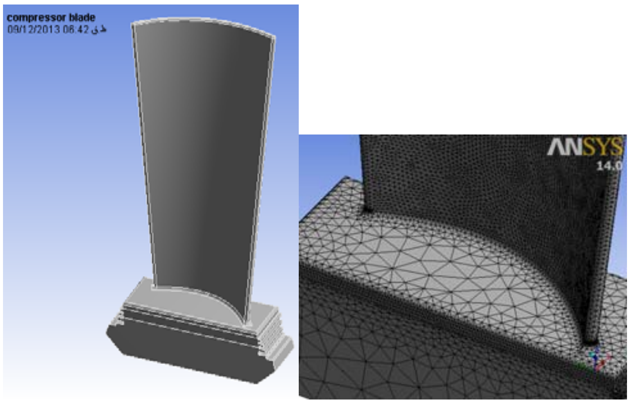


Fig. 1 Geometry and FEM model and meshing of compressor blade

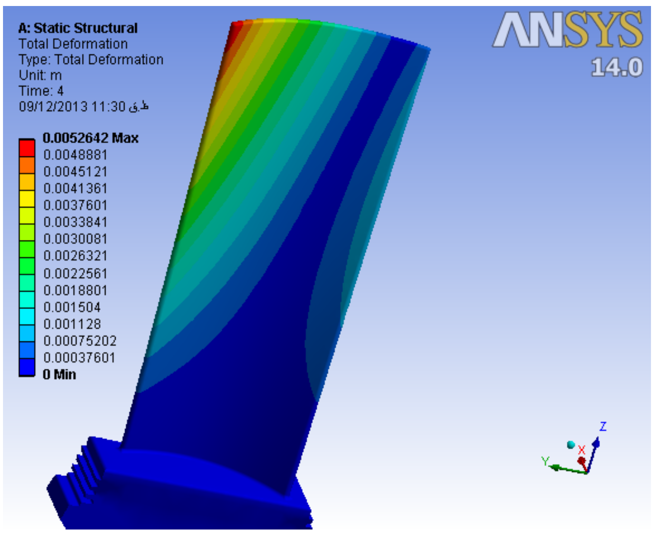
II. Stress Initialization in Model Due to the Rotation

A centrifugal stiffening effect is caused by the centrifugal force field on compressor blade and generated in rotation of shaft, such that the superficial blade stiffness rises with the blade rotational speed [8]-[10]. The effect of higher temperature in the blade change the mechanical properties of blade but it is not consider in this analysis and also it has little effect in compressor stiffness and its respected signal analysis techniques [11]-[14]. Centrifugal forces were simulated by applying an angular velocity to all elements in the model. Centrifugal forces are expected to have influence on blade resonant frequencies. The deformation of any rotating object under dynamic loads is effected by the inertial forces that they experience because of rotation. This leads to a phenomenon in system called centrifugal stiffening and then natural frequencies may significantly increase with rotor speed. This is an important parameter in the design of high-speed compressors and turbines since changing natural frequencies can cause to the increase and create unpredictable resonances.

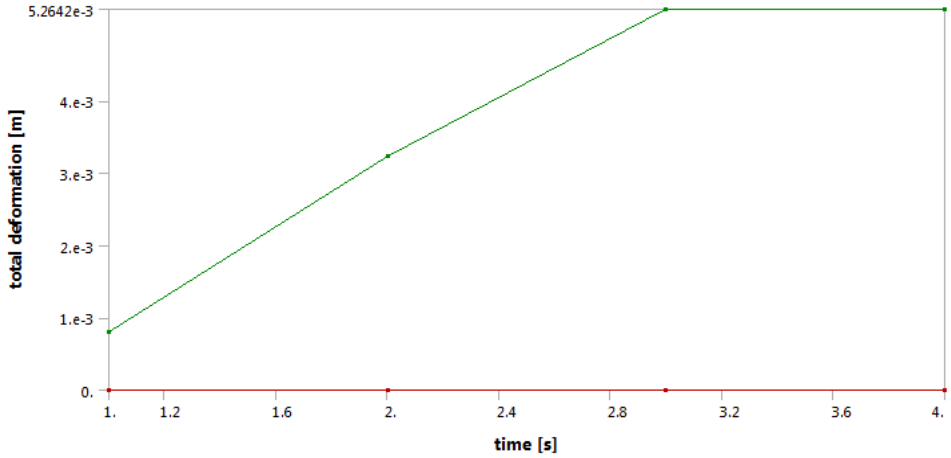
The acceleration of the blade from a status of rest to the determined angular velocity causes transient vibration of the compressor blade, which could potentially complicate the results from an accurate stress determination if not allowed to converge properly [15]. So, it is necessary to identify that the centrifugal loads have converged to a steady-state value in computational process. In this analysis, the centrifugal force is first applied in an implicit pre stress static solution. During the implicit analysis, all the root and disk nodes of the blades are fixed in space, and a velocity is applied to each node of the blade according to a linear function starting from 0 to 900 rad/s. The blade is made to rotate about the x-axis and the analysis is set to finish after four seconds. When the analysis is finished, a solution file is generated which contains stress distribution in the blades due to the application of the angular velocity in compressor blade. Then, this file is used as initial condition in the transient dynamic analysis of the blade rotation in modal analysis and their impact is considered.

The solution file contains the blade deformed geometry and the stress state distribution at the end of the analysis. This is used as initial condition for modal analysis. The stress distribution is varying from 1.096 e8 to 1.316 pa. The maximum stress is observed at the root of the blade which is 1.3161e9 pa. Also, the deformation of blade and its variation are shown in Fig. 2.

After analysis, there is a reduction in the magnitude of twist mode in blade as is shown in Fig. 3. This can be attributed to the centrifugal force acting at radial direction along the blade length. The result shows that there is a drop in the blade twist so; there will be a reduction in the stiffness of the blade. This makes a reduction in the resonant frequencies of the pre-stressed blade.

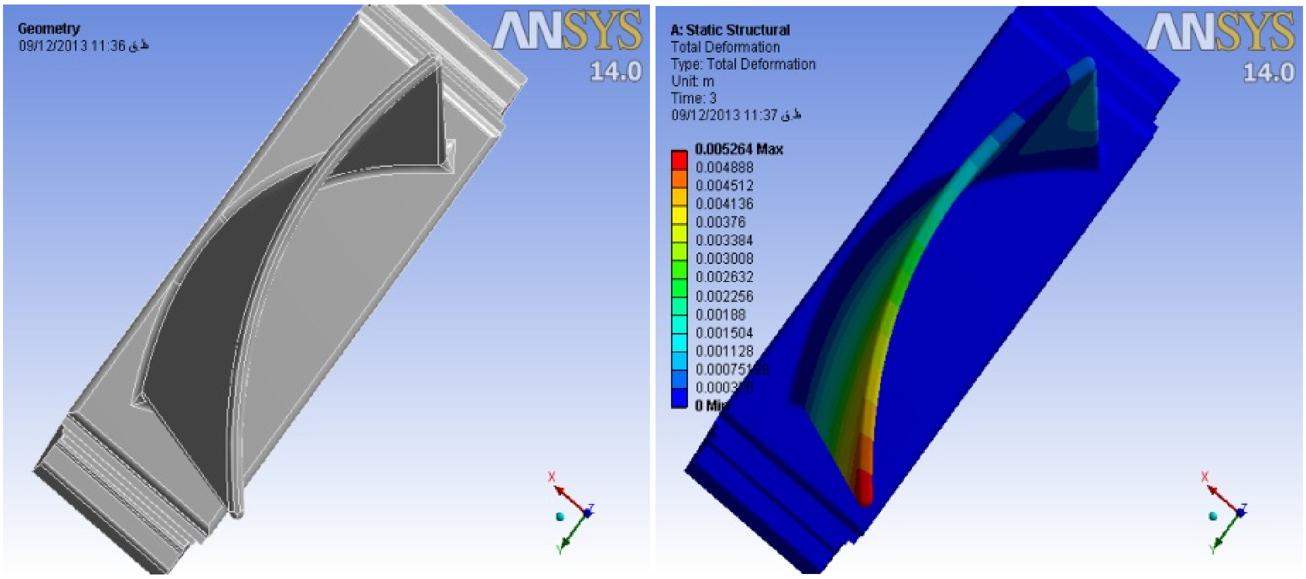


(a)



(b)

Fig. 2 Total deformation of blade due to centrifugal loads from (a) to (b)



(a) (b)

Fig. 3 Blade shape (a) before and (b) after analysis

III. Modal Analysis and Obtaining Campbell and Mode Shapes of Model

Modal analysis was applied to determine natural frequencies and mode shapes. Results were studied in detail against operating frequency of the compressor. After carrying out the modal analysis, harmonic analysis was done to see the response of the compressor under dynamical loading. Afterwards, characteristics and cause of the dynamic loading are also explained in relation to the dynamic behavior. In a gas turbine, the most common sources of excitation are running speed harmonics and vane passing frequencies [16]. Running speed harmonics are multiple frequencies of the compressor. For example a compressor rotor running at 1150 RPM (120 cycles/sec or HZ) would be having running speed harmonics occurring at 240 HZ, 360 HZ, 480 HZ, and so on. The vane passing frequency excitation is caused by air flowing through vane. Vanes are used to direct and control airflow onto the blades. Because of their structure, vanes have flow interruptions cyclic or at regular intervals that cause a cyclic force excitation on the blades. Understanding natural frequency of compressor and periodic forces on blades helps explain resonance. Resonance is a situation where response or amplitude of vibration of blade or any vibratory system is at maximum and resistance to an oscillating force is at minimum state. At this condition, the shape and frequency of a force match with the natural frequency and mode shape of the vibratory system, which is a dangerous condition for system.

A mathematical discussion of the condition of resonance is expressed briefly here and more explanations are available in [17]. In each revolution of compressor, blades pass through a field of pressure fluctuation in flow field due to nozzle or any other interruptions. This variation in pressure induces a time

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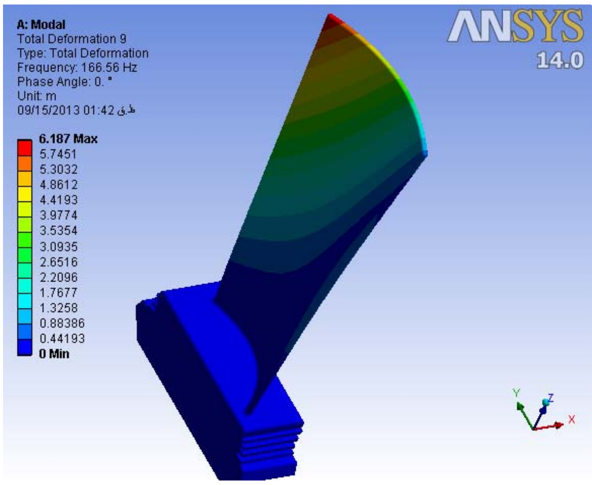
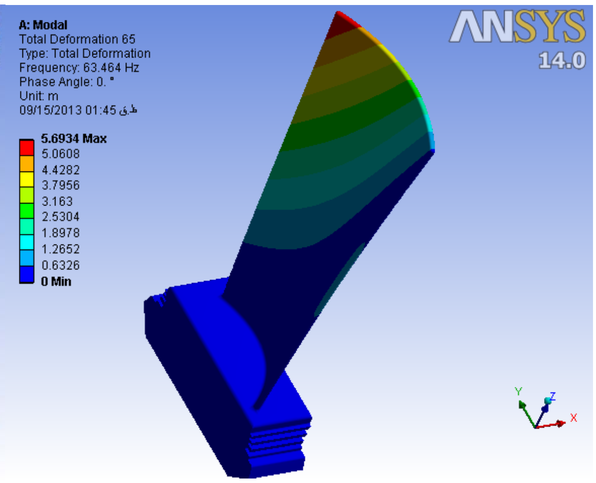
varying force field on the blades. In general form, such forces can be broken into harmonic components using Fourier analysis as follows

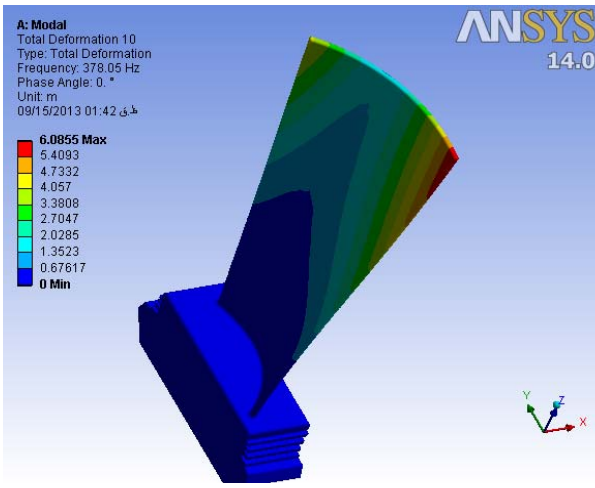
$$F = F_0 + F_1 \text{Sin}\,(\omega_1 t + \theta_1) + \ldots + F_n \text{Sin}\,(\omega_n t + \theta_n) + \ldots \quad (1)$$



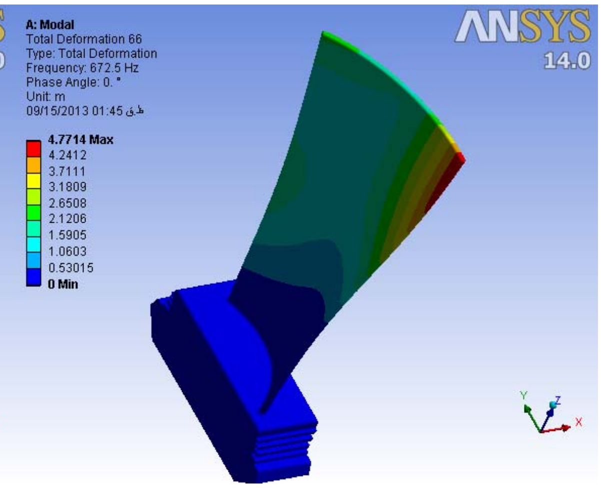
(a) Mode 1 - 166.56 HZ Fundamental



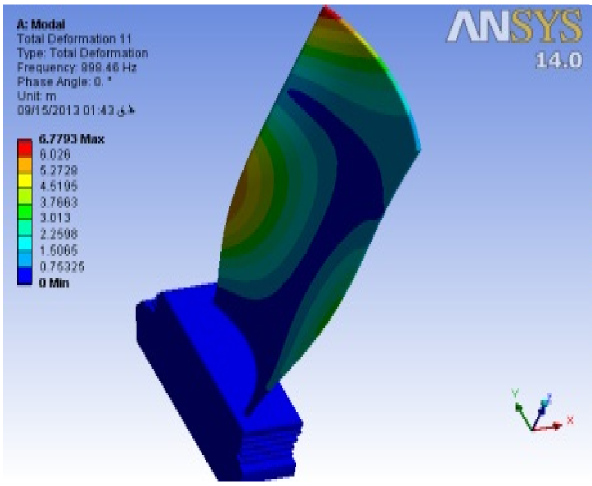
(b) Mode 1 - 63.464 HZ Fundamental



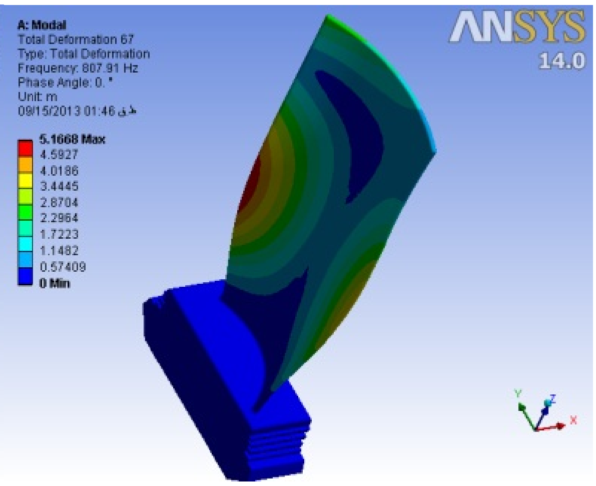
(c) Mode 2 – 378.05 HZ First Torsional



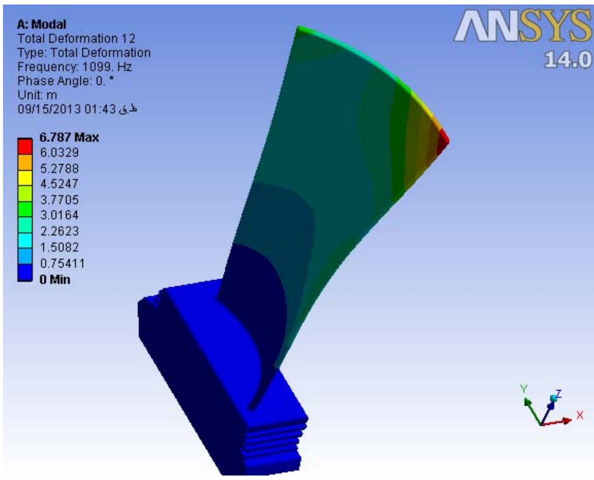
(d) Mode 2 – 672.5 HZ First Torsional



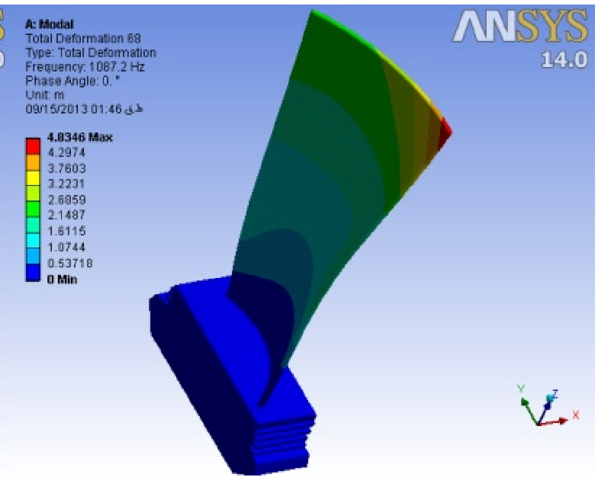
(e) Mode 3 – 898.46 HZFirst Combined Flexural- Torsional



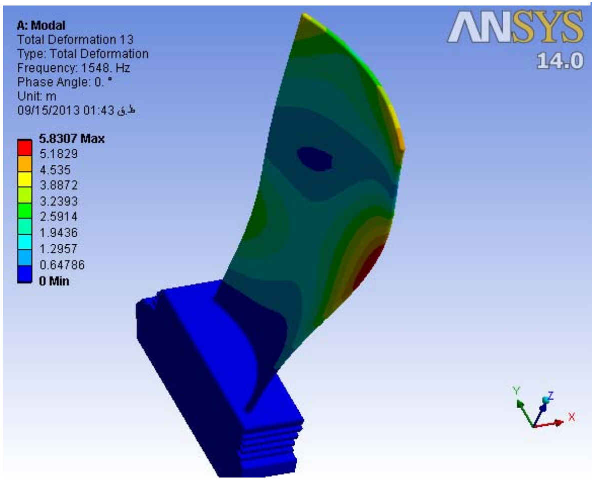
(f) Mode 3 –807.91 HZ First Combined Flexural- Torsional



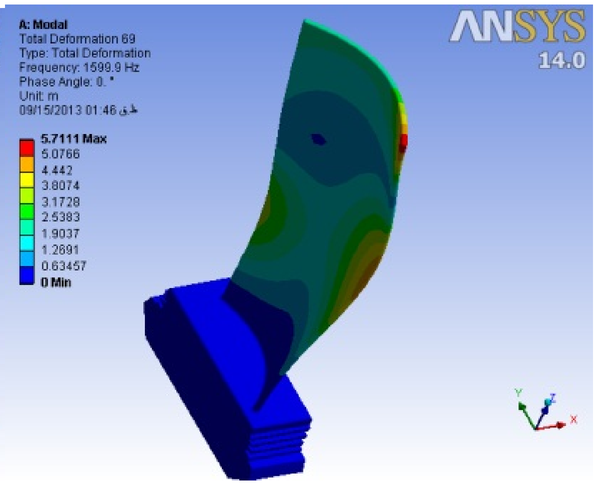
(g) Mode 4 – 1099 HZ Second Torsional



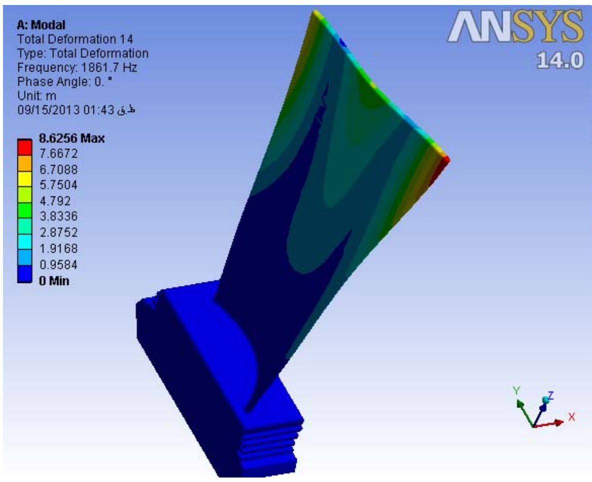
(h) Mode 4 – 1087.2 HZ Second Torsional



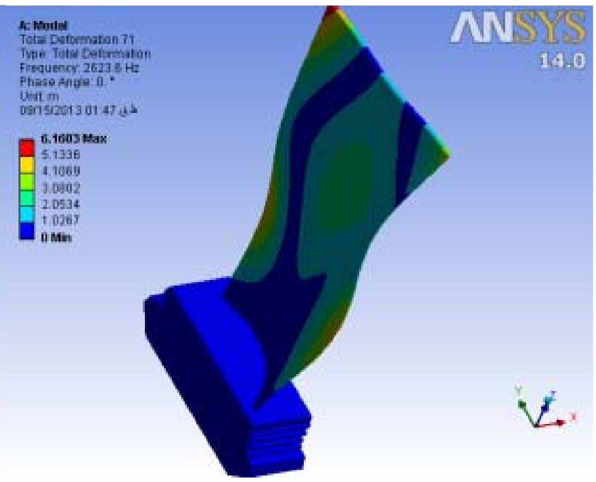
(i) Mode 5 – 1548 HZ Second Flap



(j) Mode 5 – 1599.9 HZ Second Flap



(k) Mode 6 – 1861.7 HZ Tramline



(l) Mode 6 – 2035.2 HZ Tramline

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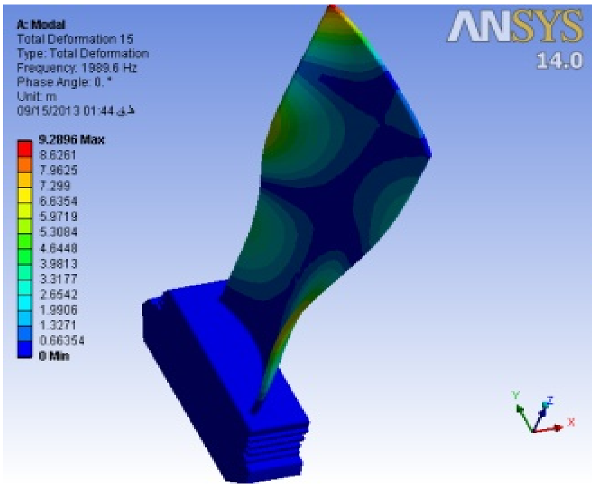
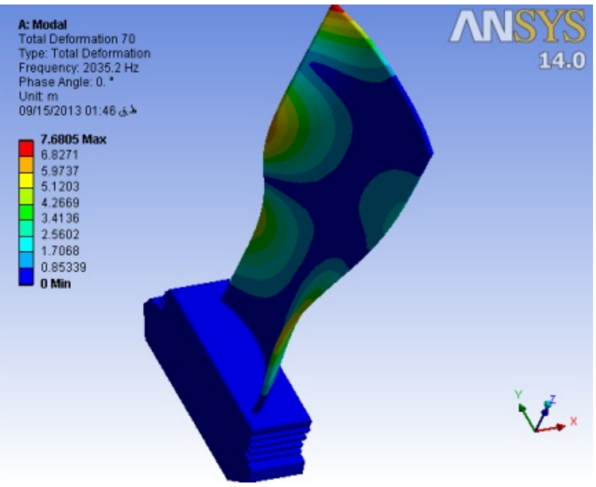
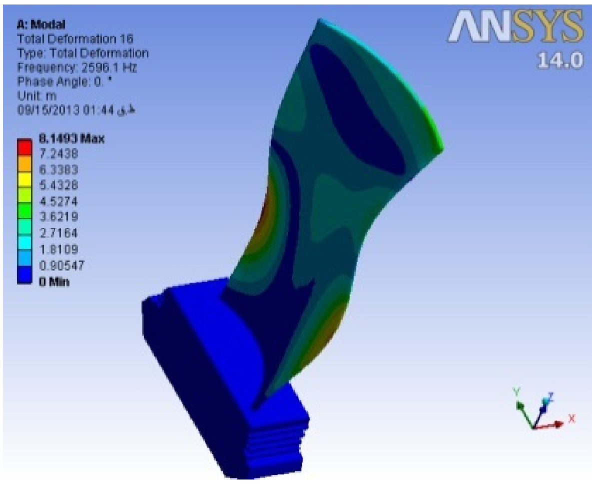


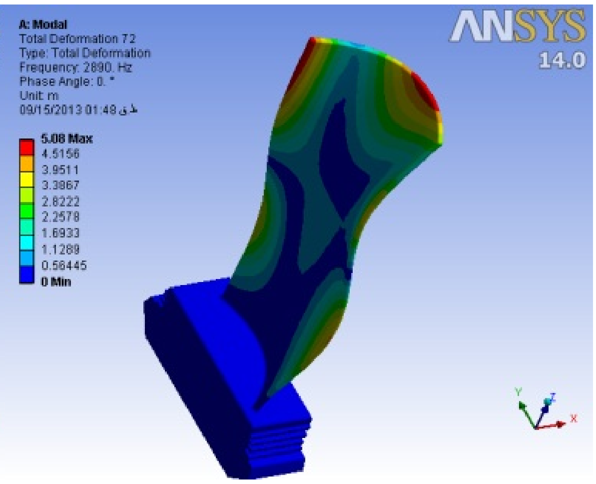
(m) Mode 7 - 1989.6 HZ Third Torsional



(n) Mode 7 - 2623.6 HZThird Torsional



(o) Mode 8 – 2596.1 HZ First Edgewise



(p) Mode 8 – 2890 HZ First Edgewise

Fig. 4 Mode shapes of pre-stressed compressor blade from (a) to (p)

Modal analysis was applied in ANSYS software. Mode shapes for blade are shown in Fig. 4. Two mode shapes are presented per each mode to represent and compare the maximum and minimum vibration of the blade at normal operation (100 rad/s) and high resonant frequency (700 rad/s) respectively in the table format from left to right. Since centrifugal forces on the compressor blade have resulted to lower pre-twist frequency in blade, the torsional stiffness of the blade has reduced and therefore resonant frequencies are expected to decrease in the torsional modes shape. Also, the centrifugal force resonant frequencies are expected to increase flexural modes acting radially outward from the blade which causes an increase in the blade bending stiffness. On the other hand, fatigue on a blade can be related to both frequency of vibration of system and the magnitude of the deflection. Since deflection magnitude reduces significantly with higher order bending modes, some of the least optimal running conditions in terms of blade fatigue can take place at lower order bending mode shapes (large deflection) and high engine orders (frequent deflection) [18].

IV. Campbell Diagram

For a rotating compressor blade, the gyroscopic effects from rotational velocities in compressor can change and vary system's damping. Such effects are commonly investigated in rotor dynamic analysis. The changes in eigencharacteristics of vibratory dynamical system at different duration of rotational velocity can be obtained with the help of Campbell diagram results. Campbell diagrams are used to explain the interference between natural frequencies and common exciting forces of system described earlier [19]. Engine orders for compressor have been computed for this article for speeds between 0 rad/s to 900 rad/s. In a Campbell diagram, the natural frequencies of the system are plotted against engine speed. Resonances can be triggered in the system if the blade natural frequency crosses an engine order, between compressor operations. The Campbell diagram is important for this article because it shows the key resonances which can be triggered in the system. Fig. 5 shows the Campbell diagram for compressor blade during startups and shutdowns of turbo machine. These modes would get excited but the time spent at resonance

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duration is important. In other words, if resonance is excited during short time of engine acceleration or deceleration then it has a small effect, but if there is a long time to be spent at a specific speed then we should care to avoid resonance [20]. From the Campbell diagram, it can be seen that the fundamental resonance can be triggered in the fundamental mode, first torsional, first combined flexural-torsional and second torsional, respectively. These results show that the first mode (166.56 Hz) at 104.1 rad/s and first torsional mode (423.49 HZ) at 300.94 rad/s-combined flexural-torsional mode (846.24 Hz) at 528.77 rad/s and finally second torsional mode (1082.1 HZ) at 683.84 rad/s can be triggered during engine operation. Respected mode shapes for these critical speeds are shown in Fig. 6 and 7, respectively. Finally, the computation of the BPF (Blade Passing Frequency) of the compressor is another important factor because it is another frequency of excitation on blades which should be avoided during engine operation. BPF of the 68th blade of the compressor can result in unwanted vibrations if it coincides with any of the resonant frequencies of compressor blade at operating conditions. It has been calculated that there are 72 blades on a single compressor disc for the IP stage then BPF is 10.2 KHZ. Since the BPF frequency is 10.2 kHz, it is much higher than the resonant frequencies of system and consequently it is no expected to pose a problem during engine operation.

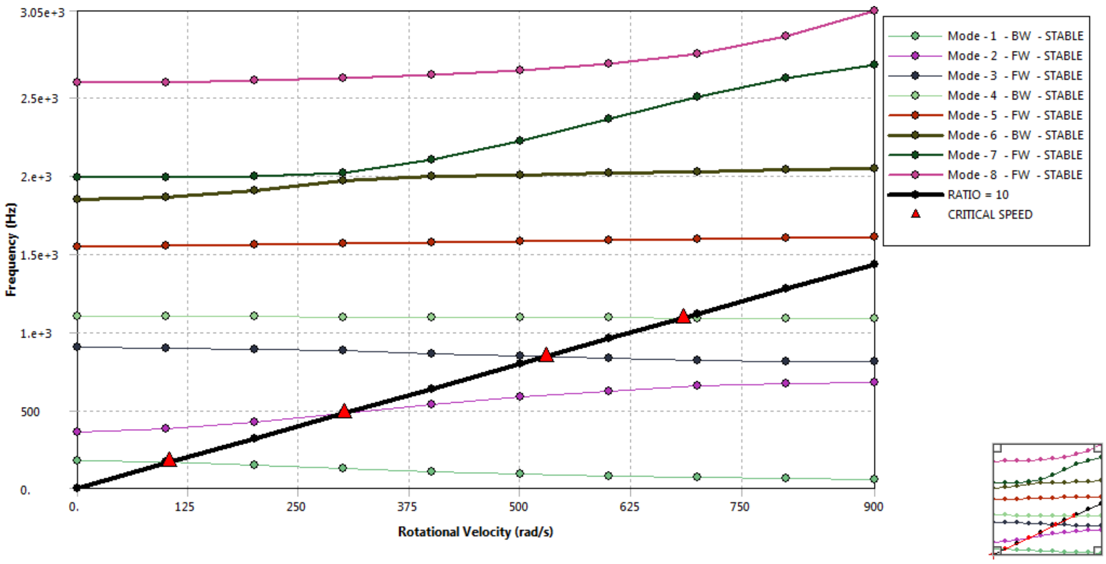

Fig. 5 Mode shapes of pre-stressed compressor blade

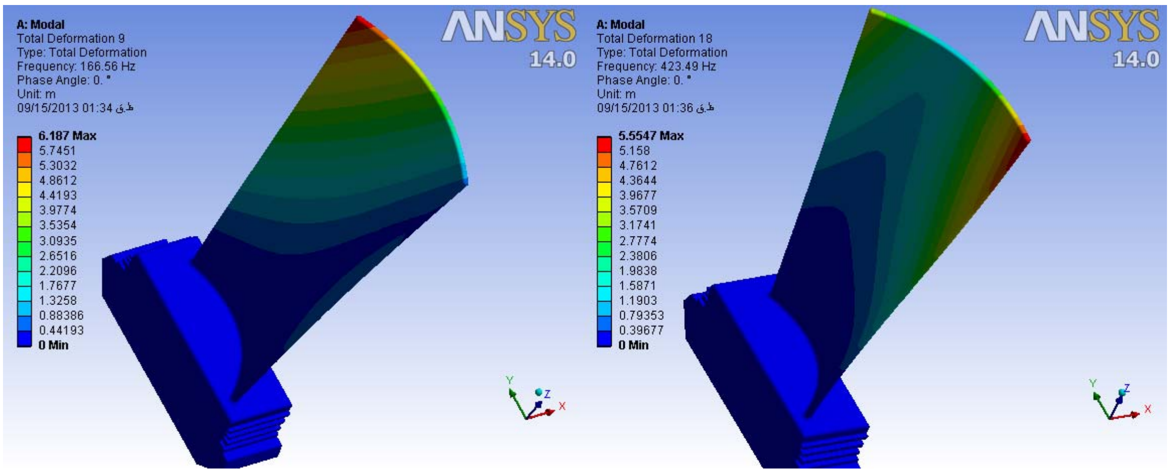

Fig. 6 Fundamental and first torsional mode at 104.1 and 300.94 rad/s

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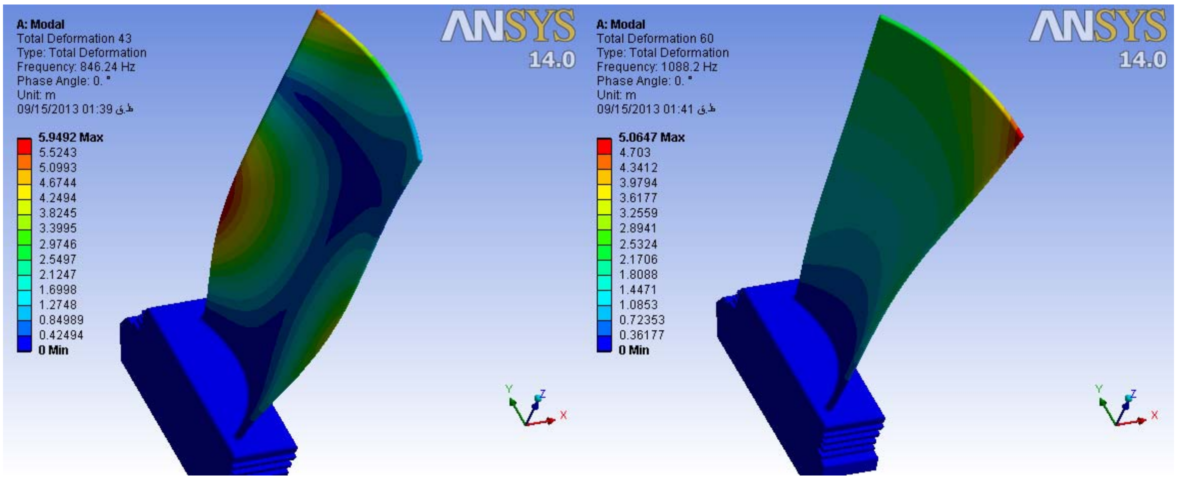

Fig. 7 Combined flexural-torsional mode and second torsional mode at 528.77 and 683.84 rad/s

V. RESULTS OF FREQUENCY RESPONSE OF MODEL

After obtaining natural frequency and mode shapes, alternating forces must exist to excite a system and make it vibrate. These forces have inherent frequencies and shapes like as compressor blade. A quasi-static gas pressure was also applied over length of the blade. Harmonic analyses usually require cyclic load data for the analysis. It was found out that pressure load in flow field around the 68th compressor blade is the only load which is varying 2.5-4 bar along the length of the blade [20]. Bending stress due to the fluid pressure is:

$$\sigma = \frac{x}{I_{yy}}[M_\theta \cos\varphi - M_\alpha \sin\varphi] - \frac{y}{I_{yy}}[M_\alpha \cos\varphi + M_\theta \sin\varphi] \quad (3)$$

With using the theory of turbo machinery and writing the equation of momentum at two directions, distribution of aerodynamic forces along the blade can be calculated.

$$q_u = \rho_b . C_{2a} . (C_{1u} + C_{2u}) . t_b \quad (4)$$

$$q_u = [\rho_b . C_{2a} . (C_{1a} - C_{2a}) + \Delta P] . t_b \quad (5)$$

Besides, in turbo machine forced vibration of the blades results from the flowing flow from each rotor passes a stator blade and then pressure is generated in a periodic pattern. This pressure is assumed to be sinusoidal with the blades. Then, the blade motion will be sinusoidal with an amplitude β at the frequency of the excited force, this being called the stator BPF as shown by ω_{spf}. This blade motion causes a phase modulation of the pressure peaks associated with each blade. In this study, this is modeled by a hanging function that is attached to each blade in compressor, with the characteristics of the blade's forced vibration. Mathematical representation of this pressure is:

$$P_n = P_a + P . e^{jb[\theta - \Omega t - \beta \sin(\omega_{spf} t + \gamma_r)]} . \left[H\left(\theta - \Omega t - \alpha_r + \frac{\pi}{b}\right) - H\left(\theta - \Omega t - \alpha_r - \frac{\pi}{b}\right)\right] \quad (6)$$

$$\gamma_r = \frac{2\pi(r-1)}{b} - round\left(\frac{s(r-1)}{b}\right)\frac{2\pi}{s} \quad (7)$$

where H is the Heaviside function and r is the r^{th} rotor blade. The mathematical representation of the local moment in flow field is given as:

$$T_P = M_0 \sin(\omega_{spf} t + \gamma_p)\, \delta(\theta - \alpha_p) \left(\frac{1}{R}\right) \quad (8)$$

$$\gamma_P = round\left(\frac{b(p-1)}{s}\right)\frac{2\pi}{b} - \frac{2\pi(p-1)}{s} \quad (9)$$

Harmonic analysis results are represented in the form of a graph commonly identified as FRF (frequency response function). Fig. 8 shows the variation of deformation with respect to operating frequency and Fig. 9 shows the variation of Von-Mises stress with respect to operating frequency and Fig. 10 shows acceleration variation.

The deformation contour is converted from time domain to frequency domain using FFT (Fast Fourier Transform). As can be seen from Figs. 8 and 10, the first nonlinear behavior is determined at 1557 HZ with an amplitude response of 2.11 E-3 m. It is suspected that it is a resonant frequency because of the high amplitude of vibration; however, this nonlinear behavior does not coincide with any of the resonant frequencies that determined from modal analysis but this is close to fifth mode (1599.9 Hz) which is called second flap mode shape. However, it should be remembered that the blade responses are affected as a result of casing and root attachment of blade compressor. Therefore, we can consider contact between these element and analysis our problem more accurately.

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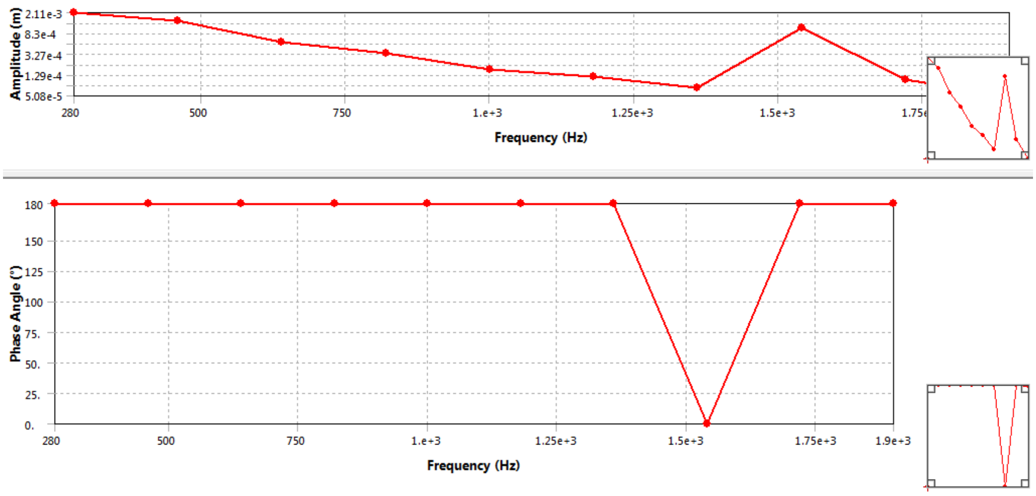

Fig. 8 Frequency response curves of deformation with respect to operating frequency

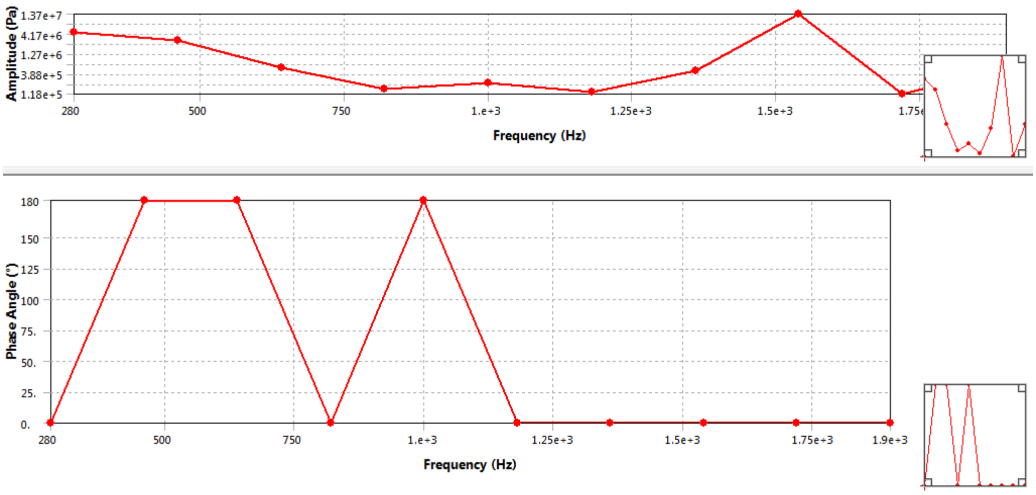

Fig. 9 Frequency response curves of Von-Mises stress with respect to operating frequency

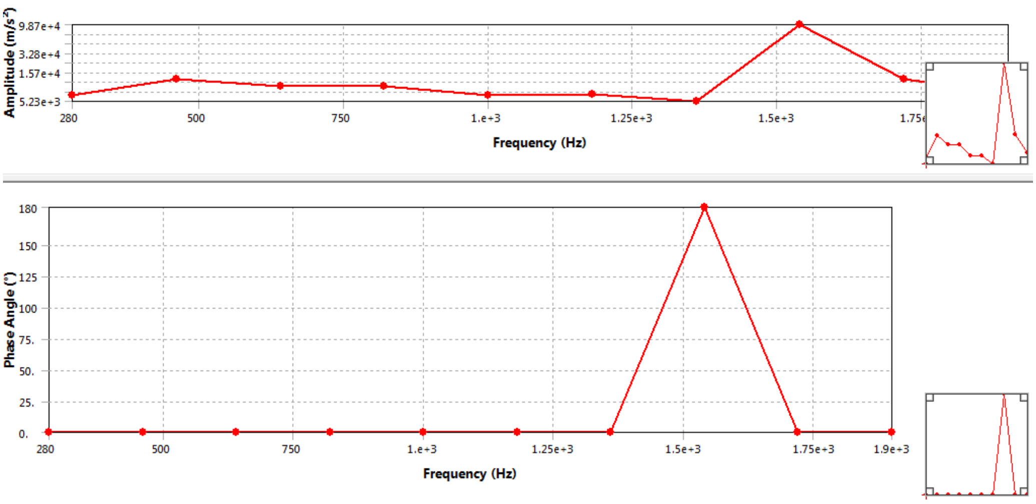

Fig. 10 Frequency response curves of acceleration with respect to operating frequency

VI. Discussion

There are numerous excitations that occur within a turbo machine. These range from the vibrations like intermittent stalls, surges, vane passing, rotor stator rubbing and cavitation to forced vibrations like unbalance. Also, maximum observed blade deformation was proved to occur during the surge event as is shown in compressor map in Fig. 11 resulting in maximum blade fatigue and Von-Mises stress. Surge is the aerodynamic excitation forces because of circumferential fluctuation in the flow field, normally caused by blades passing through the wakes of upstream blades, interaction from upstream or downstream blades, non-uniform inlet flow or from variation in the back pressure. When frequency of the incoming flow disturbances matches a compressor blade mode natural frequency, it can lead to excessive blade stress amplitudes. The frequency of the flow disturbances is proportional to the speed of the rotor. Finally, we must protect blade from this resonance condition and determine it carefully [21].

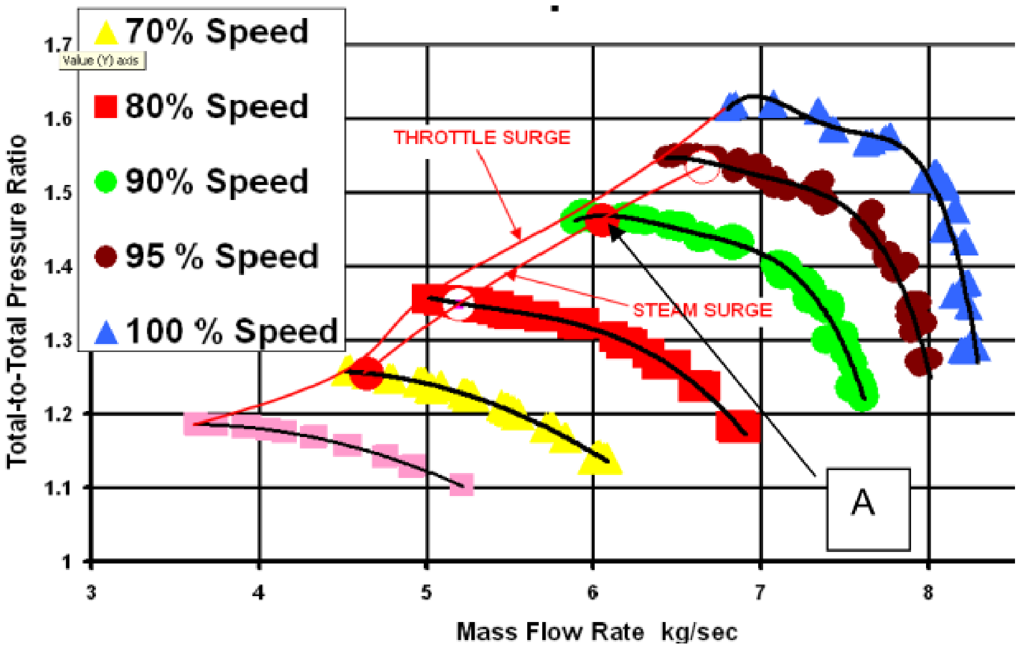

Fig. 11 Compressor performance map

VII. Conclusion

The primary conclusion from this paper is that centrifugal load has important role in changing the stiffness and frequency and mode shape. For instance, resonant frequencies are expected to decrease the torsional mode shapes and also due to the centrifugal force, resonant frequencies are expected to increase in the flexural modes which causes increase the blade bending stiffness. Also, from the Campbell diagram we concluded that time spent at resonance duration is important factor for excitation of nonlinear vibration in the compressor blades. These results are maybe because of the aerodynamic excitation forces in flow field but for determining this issue accurately, we need couple CFD result with nonlinear vibration synchronal. Totally, it is important this resonance condition should be avoided to extend blade life by limiting fatigue.

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