Torsional Vibration Analysis Of Large Rotor System Using Finite-Element and MatLab Procedures

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Abstract

In this paper, the finite-element and the matlab procedures are used for the torsional analysis of large rotor system. A large rotor system of 13 discs are considered for the purpose of analysis. As a result, the finite-element and matlab procedures are good tools for the analysis of vibration analysis and design of large rotors and their results are accurate in comparison with other literatures. The normal elastic curve and T-**ω** diagram obtained in this study are an effective illustration for the vibration problems in large rotors, and the developed equation for drawing the normalized elastic curve reduce the need for tabulated calculation of this curve and its very essential for vibration analysts and designers.

تحليل الاهتزازات الالتوائية لمنظومات الدوران الكبيرة باستخدام طرق العناصر المحددة والماتلاب

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الملخص

لقد تم في هذا البحث استخدام طريقتين ، احدهما العناصر المحددة والاخرى برنامج الماتلاب لتحليل الاهتزازات الالتوائية لمنظومة دوران كبيرة. منظومة كبيرة تحتوي على 13 قرص استخدمت لهذا الغرض. كنتيجة، طريقتي العناصر المحددة والماتلاب ادوات جيدة لتحليل الاهتزازات وتصميم منظومات الدوران الكبيرة ونتائجهما دقيقة مقارنة مع طرق اخرى موجودة في المصادر. منحنى المطيلية و منحنى العزم-السرعة المستحصلان من الدراسة يوضحان بشكل فعال مشاكل الاهتزاز في المنظومات الكبيرة وكذلك العلاقة المطورة لرسم منحنى المطيلية تقلل الحاجة لاستخدام الحسابات المجدولة لهذا المنحنى وهي مهمة جدا لمحللي الاهتزازات والمصممين.

1-Introduction

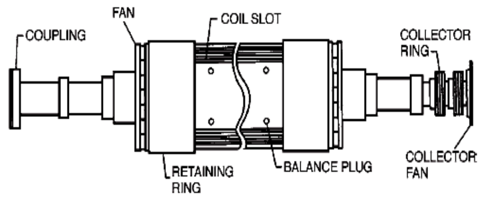
In the design of rotating shaft, torsional vibration analysis is important for ensuring reliable machine operation due to machine passing through that range of vibration from rarely occurring high-level transients to continuous relatively low levels of excitation. If the shaft and rotating component failures occur on these large machines as a result of shaft torsional oscillations, the consequences can be catastrophic.

In the worst case, an entire machine can be wrecked as a result of the large unbalancing forces that can arise following shaft separation and turbine blade failures, and this has actually occurred [1]. For these reasons great attention is generally taken at the design stage to ensure that high-speed rotating machines have the required torsional capability. This can be done by use of numerical techniques such as finite elements and matlab.

This paper present a developed finite element and matlab procedures to analyze torsional vibration problems of large shafts to find the critical natural frequencies and mode shapes, and also to plot the normalized elastic curve of the shaft which shows the critical points of failure. The results have been compared with analytical solutions and other research work.

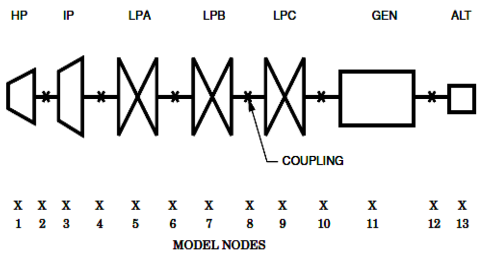
2-Torsional System Modeling

Rotating turbo-machinery shafts usually consists of several rotors that are connected in tandem by coupling. Generally the main body regions of an individual rotor have significantly larger diameter sections than the rotor extensions at each end. These shaft extensions often contain the seals and bearing journals and may terminate with integral or shrunk-on couplings [2]. The following figure schematically illustrates some of these features on a generator rotor [1].



3-Vibration analysis Model

To achieve good accuracy in analysis results, the required number and distribution of elements along the axial length of the machine shaft are often gained through modeling of a particular class of machine. This include the vibration response frequency range of interest, the number of locations that have distinctly different diameters and other geometric discontinuities, and the relative values of stiffness and inertia for discrete spans of the rotor. For a large turbine generator in which the shaft torsional response needs to be estimated following a transient disturbance, the model is strongly influenced by the fact that the shaft response is primarily in the lower order modes of vibration (i.e. less than 60 Hz). The Figure below illustrates such a model, where a total of 13 nodes are placed at the centers of inertia of each main rotor (in this case five turbines, a generator, and a shaft driven alternator rotor) and at the six couplings.



With the high computing power and memory of modern computers, it is no longer necessary to be frugal in selecting the number of nodes and corresponding degrees of freedom for constructing vibration models. If required, some neighboring sections can be combined to give equivalent properties over the combined length to reduce the number of elements in the model [1].

4-Torsional Analysis Using Finite Elements and MatLab

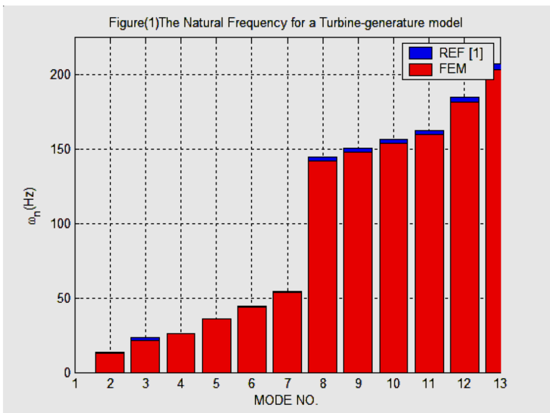
All the required finite element stiffness and inertia matrices (developed from first principles) for calculating natural frequencies and mode shapes can be reviewed in Reference [3].

The case study that follows is a torsional frequency analysis of a large steam turbine tandem compound generator set shown in the previous Figure that has five large steam turbine rotor elements, a large generator rotor element, and a shaft-driven alternator element for providing direct electric current to the generator field winding. HP, IP, LPA, LPB, LPC, GEN, and ALT refer to the high-pressure, intermediate pressure, low-pressure A, B, C turbines and the generator and alternator rotors. Also included in the model are inertias that represent the couplings between each of the rotor elements. As these couplings (at nodes 2, 4, 6, 8, 10, and 12) have significantly lower polar moments of inertia than the main rotor elements. The turbine-generator stiffness and inertia values are given the following table [1].

Table(1) The turbine-generator stiffness and inertia values.

Node number	Inertia, lb-in^2	Shaft span nodes	Stiffness, lbf-in/rad
1	2.0E7	1-2	4.0E9
2	2.0E6	2-3	4.0E9
3	2.0E7	3-4	4.0E9
4	2.5E6	4-5	4.0E9
5	3.0E7	5-6	4.0E9
6	3.25E6	6-7	4.0E9
7	3.5E7	7-8	4.0E9
8	3.5E6	8-9	4.0E9
9	3.5E7	9-10	4.0E9
10	3.75E6	10-11	4.0E9
11	4.0E7	11-12	4.0E8
12	4.0E5	12-13	4.0E8
13	4.0E6		

A finite element program has been developed in this work (based on Holzer method [4]), for the analysis of this case study. The calculated torsional natural frequencies are shown in Figure (1) along side with the results of Reference (1). It is clear from this Figure that there is a good agreements between the finite elements and the results of Reference (1).

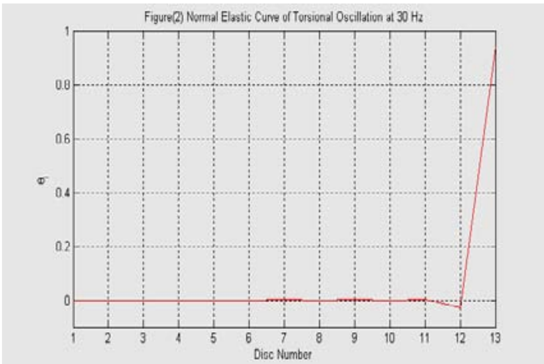


Figure(1) The natural frequency for the turbine-generator model .

The normalized elastic curve at natural frequency of 30 Hz is drawn using a matlab m-file developed in this work based on an equation developed by the current author during his teaching of the vibration subject in the college of engineering, university of basrah which is written as follows:

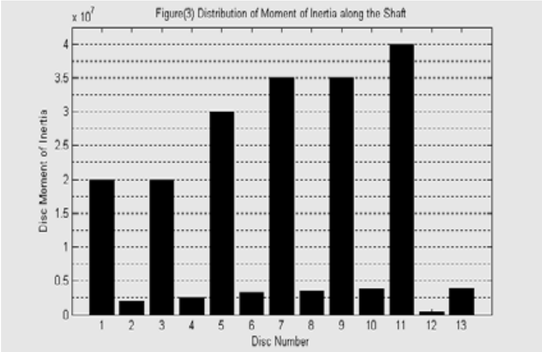
$$\theta_n = \theta_{n-1} - \frac{\omega^2 L_{n-1}}{GJ_{n-1}} \sum_{i=1}^{n-1} I_i \theta_i$$

where $\boldsymbol{\theta}$ is the normalized angular displacement, $\boldsymbol{\omega}$ is the critical speed, **L** is the length of the shaft element, **G** is the modulus of rigidity of material of the shaft element, **J** is the polar second moment of area of the shaft element, **I** is the polar second moment of Inertia of the disc. The results are shown in the following figure :



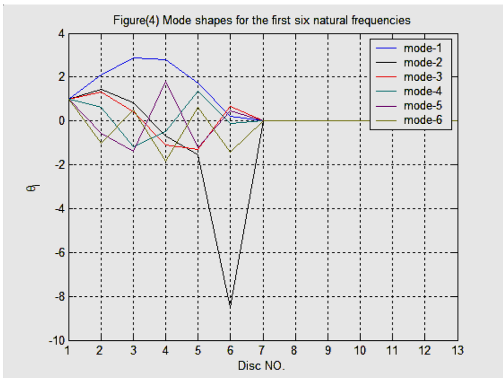
Figure(2) Normal elastic curve of torsional oscillation at 30 Hz.

It is feasible from this figure that the danger's area of vibration of the shaft whole assembly is occurred between disc 11 and disc 13 which is due to the mass distribution along the shaft which clear from the following figure:



Figure(3) Distribution of moment of inertia along the shaft.

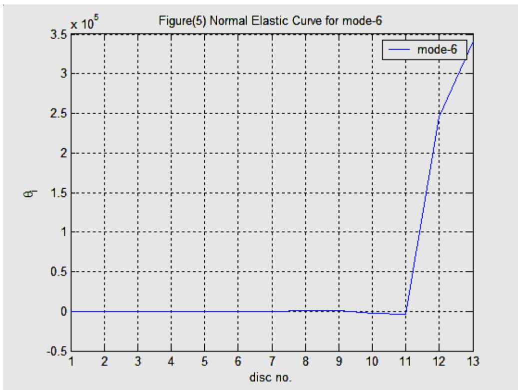
The mode shapes for the first six mode of vibration are drawn in figure below:



Figure(4) Mode shapes for the first six natural frequencies.

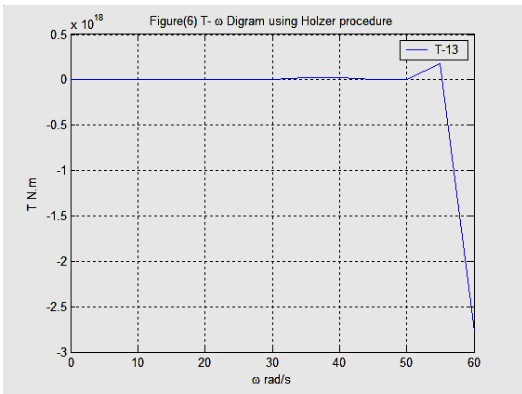
It is clear from this figure that the first six discs of the shaft assembly are the most effective during whole modes of vibration of the shaft and also its feasible that mode-6 is the most effective mode of vibration which is occurs at natural frequency of 43.59 Hz which is less than the operating frequency which 50 Hz in the middle east and 60 Hz in the United State and Europe, and this means that shaft needs special attention during the real operation when its passes this natural frequency during start up and shut down of the turbine-generator unit.

The normal elastic curve for mode-6 have been drawn and shown in Figure(5). It is clear from this figure that the same conclusion of Figure(2) can be considered, that is the danger's area of vibration is between disc 11 and 13.



Figure(5) Normal elastic curve for mode -6.

Using the Holzer procedure given in Reference [5], the T-**ω** diagram for the shaft have been calculated and drawn using a matlab m-file developed in this work the results are shown in Figure(6) for disc number 13:



Figure(6) T-ω diagram using Holzer procedure.

The advantage of drawing the T-**ω** diagram is that the points of intersection of the curve with the zero line represents the natural frequencies of the shaft assembly and also shows the value of the maximum effective torque occur during the operation of the shaft.

4-Conclusions

From the above analysis, the following conclusions remarks can be drawn:

1- The finite-element and the matlab procedures are good tools for the vibration analysis and design of large rotors.

2- The normal elastic curve and the T-**ω** diagram are an effective illustration for the vibration problems in large rotors.

3- The author developed equation for the drawing normal elastic curve is reduce the need for the tabulated calculation of this curve given in Refrence[6].

5- References

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[6] Hanna and Stephens, "Advanced Mechanics of Machines",