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Evaluation of Critical Speed of Generator Rotor with external load

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Abstract—*As a result of continuing demand for increased performance, modern generators are sometimes designed to operate near critical speed, which causes difficulty in maintaining the rotor balance required to ensure acceptable vibration levels. Forced vibration due to the external load acting on the shaft line is not considered in evaluation of critical speeds of multi-section shaft. The importance of the current work carried out is to evaluate the critical speed through an automated process. By this process any configuration of rotor can be used to evaluate the critical speed of multi-section rotor. Thus for evaluation of critical speed and to study the rotor behavior in dynamic condition, a script is generated in order to automate the process, thereby reducing the time consumed & increasing the accuracy. The uniqueness of this methodology adopted is that with the help of one script the input like diameter, length of each step of rotor, load acting on the rotor is accessed as variable into the script. The script is generated in ANSYS using Ansys Programming Design Language (APDL). The generated script results are correlated with theoretical results and the measured results, which are well within the acceptable limits. Important conclusion of the study is achieving complete automation for evaluating the critical speed by considering external load & for any configuration or combination of rotors.*

Keywords—*Rotor generator, Critical speed, Ansys programming design language, Natural frequency*

I. INTRODUCTION

Industrial machinery is said to be optimized when it results from a state-of-the-art mathematical simulation and design, comprehensive prototype, preproduction testing and manufacturing with minimal costs. A design is stated to be satisfactory if it satisfies the high technical requirements and low production costs simultaneously. Hence the machinery design involves an element of compromise in the many requirements of secondary importance. The success of a design, is not simply achieved by looking at one parameter in isolation, but is a complex process, with various parameters in interaction. Generally, as the objectives of the design can be formulated in terms of a few global variables, there may be a great many variables also defined and handled within each subsystem. The evaluation of how the variables of primary importance affect the overall quality of the design is of significant importance. Choosing an optimal design based on the particular requirements of the application and at the same time keeping in view the manufacturing costs is a challenge for any design.

An analysis of rotor bearing dynamics is critical to the design of any high-speed machinery that has rotating parts. Such analysis has taken a giant leap forward with the development of standard computer programs for determining various system characteristics. When the tedious and error prone modeling process is automated and data input is consistently shared among the analysis processors and graphic postprocessors, the analysis would be significantly improved.

Early in the development of rotor dynamics, it was a commonly prevailing notion that operation above the first critical speed was impossible. Nascent stages of rotor dynamic study began with Rankine (1869), whose work resulted in his conclusion that a shaft would be stable under the first critical speed and would always be dynamically unstable above that. Thus, the unbounded increase in the vibration in the vicinity of the critical speed was seen as an unstable condition. This misconception was corroborated by Greenhill (1883), who stated that the shaft inertia contributes to its buckling, thus reinforcing Rankine's concept. Later Dunkerley (1895), using the Reynolds's eigen value concept, could calculate critical speeds of wide variety of shaft disc systems. The turn of the century saw the strong endorsement of Rankine's concepts.

P. H. Mathuria [1] presented number of methods to find the natural frequencies of undamped system through simple methods suitable for hand calculations and the transfer matrix method. The transfer matrix technique may be regarded as extension of holzer method. The transfer matrix relates the state vector of one station to the next. Hence the more complex problem can be solved using these transfer matrix method as there exist a recurrence formula.

Dr. Heinrich Spryl & Dr. Gunter Ebi [2] presented a paper on finite element method on modal analysis of shaft line of hydro power plant. In this, the shaft line was designed to have a first bending critical speed between load rejection speed and runaway speed. The critical speed is determined by structural properties of the shaft line and the bearings which are independent of speed and the speed dependent characteristics, e.g. oil film elasticity, damping, gyroscopic effect and load due to unbalance magnetic pull to the non-symmetry of rotors. In general, the structural properties of the shaft line are well defined in terms of material elasticity and mass distribution.

B. Gurudatt, S. Seetharamu, P. S. Sampathkumaran and Vikram Krishna [3] presented a paper on determination of critical speed of a multi-section rotor using Ansys Parametric Design Language. In this paper the methodology has been presented to use the unbalance force as excitation force to produce the force vibration in the system to obtain the response of the considered system in the form of amplitude and phase angle from which the critical speed can be determined. ANSYS Parametric Design Language (APDL) is a scripting language that has been used here to build the model and automate tasks by using parameters (variables). A sequence of ANSYS commands can be recorded in a macro file which enables the user to create a customized ANSYS command that executes all of the commands required for a particular analysis.

M. Greenhill & A.Cornejo [4] carried out the effect of critical speed resulting from unbalance of backward whirl modes. When considering the synchronous critical speeds of rotor bearing system, the majority of attention in both the technical and practical design has been focused on the unbalance excitation of natural frequency with whirling and spinning in same direction. Common intuition leads most analysts to conclude that since unbalance produces a forward processional force, only modes that are whirling in the same direction should be considered for potential synchronous excitation. Based upon the test results of a typical natural frequency analysis on a fluid film bearing supported rotor system, a unique resonance condition was encountered.

Farouk O.Hamdoon [5] presented a paper on finite element for modeling rotating machinery vibrations. They used beam element to model the shaft with variable cross section, rotating disk and bearing. The natural frequencies of a rotor-bearing system are commonly called whirl speeds. These whirl speed generally change the magnitude as the shaft rotating speed changes because of the gyroscopic effect of the rotating shaft and disk. Whenever the rotating speed coincides with one of the natural frequencies of whirl, a resonant condition is introduced, and this rotating speed is called critical speed. The modal Analysis module of ANSYS is applied to calculate all natural frequency of rotor-bearing system with different rotating speeds and then a Campbell diagram is obtained. From this diagram it's easy to locate the critical speeds of the system, a slope of 1X is drawn when this line intersects with the frequency line it's termed as critical speed.

To summarize the literature review it's been observed that external load has not been incorporated in evaluating critical speed, which when considered tends to vary the critical speeds significantly. In the present study, external load on shaft due to turbine, slip ring and end rings is considered as it represents the real time problem in evaluating critical speeds.

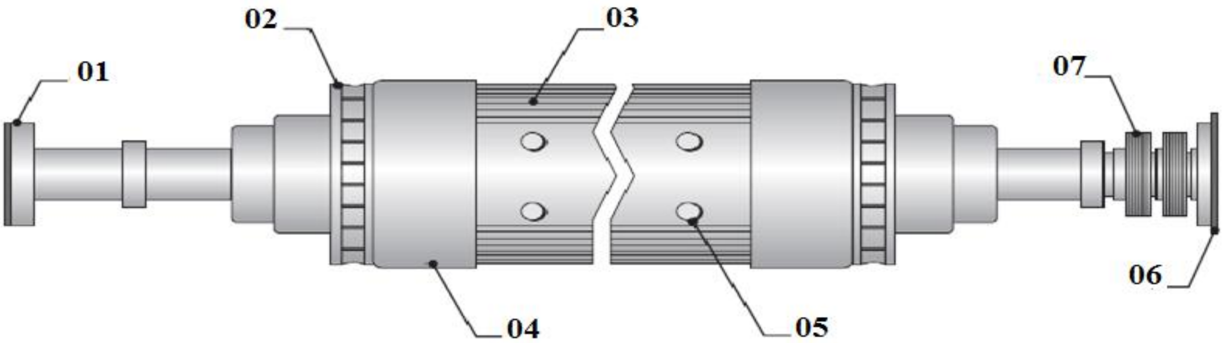

Figure 1. Main components of a rotor

01.	COUPLING	**05.**	BALANCING PLUG
02.	FAN	**06.**	COLLECTOR FAN
03.	COIL SLOT	**07.**	COLLECTOR RING
04.	RETAINING RING		

The main component of the rotor is shown in figure 1. The component like fan, coil slot are mounted on the shaft, complete assembled rotor is mounted between two bearing one being driving end and other being non-driving. The complete rotor is enclosed in a non-rotating enclosure called stator. In the field of generator design, the rotors are subjected to various loading scenarios like load due to turbine, coupling, unbalance magnetic pull, coil slots, retaining rings etc. To analyze the rotor in dynamic behavior, finite element methods are adopted, which provides high computational platform & accurately predict the results.

II. METHODOLOGY

The multi-section of the shaft has been modelled in ANSYS using BEAM 188 element which is linear/quadratic two-node beam element in three dimension with six degree of freedom at each node. These include translation in the X, Y and Z direction and rotations about X, Y and Z axis.

The external load is modelled using MASS 21, it is a point element having up to six degree of freedom i.e. three translational in X, Y & Z axes and three rotational in X,Y and Z axes.

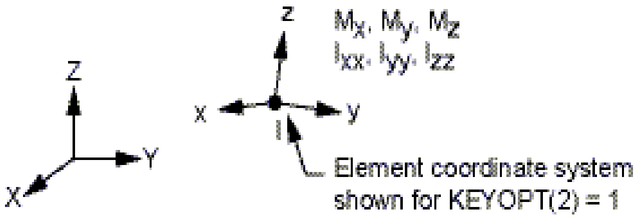

Figure 2 Mass element representation.

Provision is made to the load coming on to the shaft due to turbine, core coil slots, end ring, stop ring etc, which are mounted on to the shaft. The script imports the value from the external file which in turn parameterised each value of the input data. The parameterised values such as diameter and length is utilised to generate the model, the generated model from the script is shown in figure 3.

<u>Input file(1):</u>

```
/PREP7
!!TOTAL NUMBER OF STEPS
S=24
!!FIRST BEARING LOCATION
B1=6
!!SECOND BEARING LOCATION
C1=19
```

***Table1*:** Input data for undamped critical speed analysis with external load.

	OUTER DIA (mm)	INNER DIA (mm)	LENGTH (mm)	LOAD (kg)
1	00	00	00	00
2	101.06	00	19.13	00
3	63.53	00	44.50	500
4	63.53	00	44.50	00
5	101.06	00	19.53	00
6	63.53	00	45.72	00
7	50.80	00	51.33	00
8	50.80	00	76.20	125
9	251.46	00	126.95	125
10	51.05	00	51.33	125
11	51.05	00	76.20	125
12	251.46	00	126.95	125
13	50.93	00	51.10	125
14	50.93	00	76.20	125
15	251.46	00	127.10	125
16	50.93	00	52.71	125
17	50.93	00	76.20	125
18	63.50	00	70.00	00
19	63.50	00	44.45	00
20	63.50	00	44.45	00
21	63.50	00	55.88	00
22	25.65	00	25.40	750
23	25.65	00	25.40	00
24	25.40	00	25.43	00

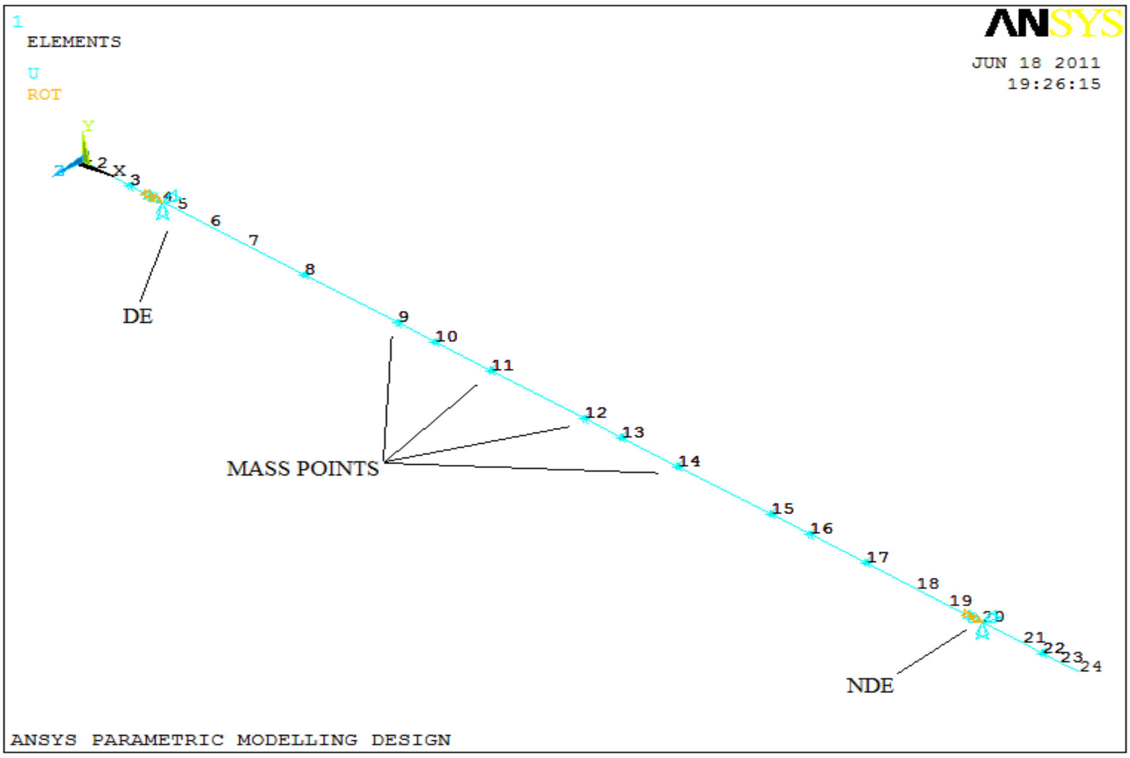

Figure 2: Shaft line for undamped critical speed analysis with external load

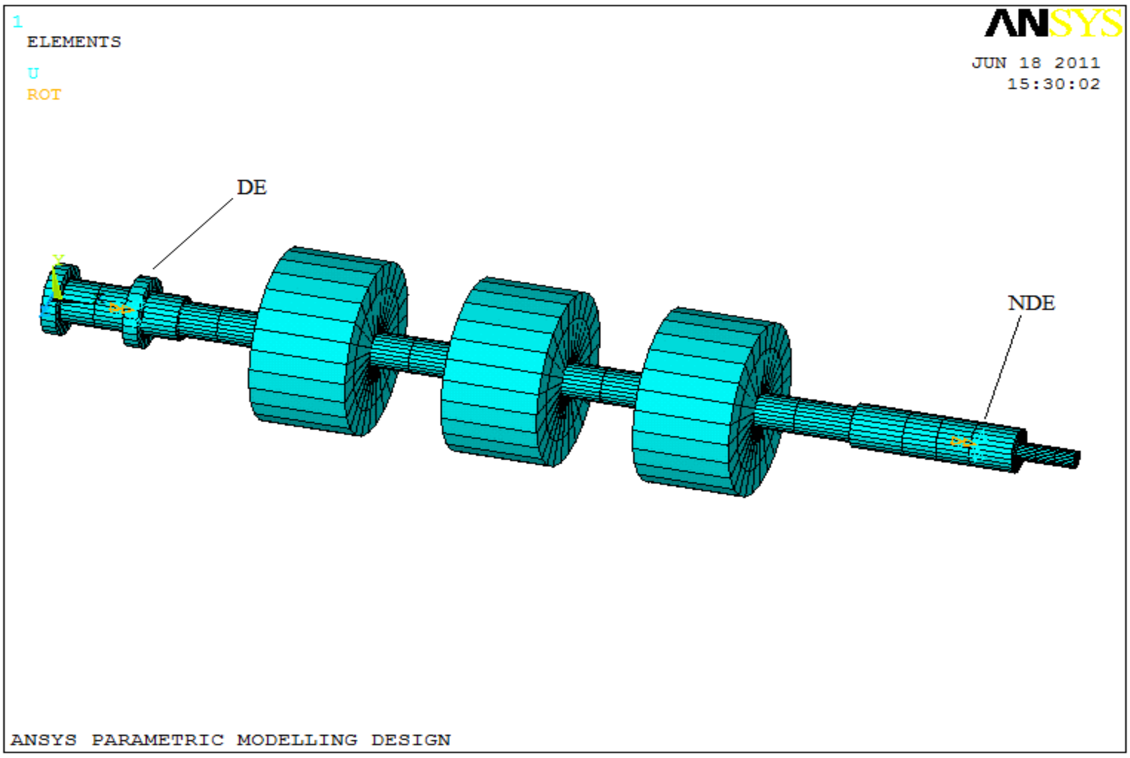

Figure 3: Generated model for undamped critical speed with external load.

Table 2: Output values of the undamped critical speed analysis with external load.

	OUTER DIA (mm)	LENGTH (mm)	LOAD (kg)	DEFLECTION (mm)
1	00	00	00	0.53792
2	101.06	19.13	00	0.44265
3	63.53	44.50	500.0	0.22104
4	63.53	44.50	00	0.00000
5	101.06	19.53	00	-0.98091E-01
6	63.53	45.72	00	-0.32860
7	50.80	51.33	00	-0.58187
8	50.80	76.20	125.0	-0.91792
9	251.46	126.95	125.0	-1.4134
10	51.05	51.33	125.0	-1.5837
11	51.05	76.20	125.0	-1.7125
12	251.46	126.95	125.0	-1.7952
13	50.93	51.10	125.0	-1.7906
14	50.93	76.20	125.0	-1.6479
15	251.46	127.10	125.0	-1.2818
16	50.93	52.71	125.0	-1.1029
17	50.93	76.20	125.0	-0.77125
18	63.50	70.00	00	-0.43196
19	63.50	44.45	00	-0.21443
20	63.50	44.45	00	0.00000
21	63.50	55.88	750.0	0.25779
22	25.65	25.40	00	0.36014
23	25.65	25.40	00	0.46328
24	25.40	25.43	00	0.56655

Table 3: Comparison of results from Ansys & analytical approach

	1^{ST} Critical speed (rpm)	2^{nd} Critical speed (rpm)	3^{rd} Critical speed (rpm)	Shaft Centre of Gravity from origin (mm)	Rotor Centre of Gravity from origin (mm)	Mass moment of inertia of shaft $(K.g/mm^2)$
Ansys Script results	668	2387.15	4346.03	705.11	697.58	1.18
Analytical approach results	677	2452.79	4504.93	705.11	697.58	1.18
Error %	1.35	2.75	3.65			

III. CONCLUSION.

The results obtained from the script shows there is close agreement with the analytical approach. Even though there is a deviation in the Ansys results, the error rate is well within the limits.

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