

MULTISTAGE COUPLING OF MISTUNED AIRCRAFT ENGINE BLADED DISKS IN A FREE VIBRATION ANALYSIS

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Abstract. *Considered here is the effect of multistage coupling on the dynamics of an aircraft engine rotor with eight mistuned bladed discs on a drum-disc shaft. Each bladed disc had a different number of rotor blades. Free vibrations were examined using finite element models of single rotating blades, bladed discs, and an entire rotor. In this study, the global rotating mode shapes of flexible mistuned bladed discs-shaft assemblies were calculated, taking into account rotational effects, such as centrifugal stiffening. The thus obtained natural frequencies of the blades, the shaft, bladed discs, and the entire shaft with discs were carefully examined and compared with a tuned system to discover resonance conditions and coupling effects. Mistuning caused considerably more intensive multistage coupling than the tuned system and distorted the mode shape nodal diameters.*

1 INTRODUCTION

The influence of multistage coupling on disc flexibility in mistuned bladed disc dynamics were presented for the first time by Bladh et al. [1], who studied single-stage and two-stage rotors of a very simple stage geometry. Shahab and Thomas [2] presented disc flexibility coupling effects on the dynamic behaviour of a multi disc-shaft system. Rzadkowski et al. [3] showed that the coupling of three identical industrial bladed discs on a shaft segment changes the mode shapes of shrouded bladed discs up to the seventh nodal diameter. Sharma et al. [4] analyzed a turbine rotor with 16 discs, with only one of them being bladed, under earthquake-force excitation, but they did not investigate couplings between the shaft and bladed discs. Sinha [5] carried out an analysis of two mistuned bladed discs. Here a system of masses and springs was used to model the bladed disc. Laxalde et al. [6] used the multistage cyclic symmetry method to show the coupling of two bladed discs, each with a different number of blades, mounted on a flexible shaft. Rzadkowski and Drewczynski [7, 8] performed an analysis of eight tuned bladed discs on a shaft with an equal number of blades on each disc to show that coupling between particular bladed discs was visible up to modes with two-nodal diameters. A similar analysis [9] was also carried out on a rotor with a different number of blades on each bladed disc. The results showed that in coupling analysis it is essential to account the different numbers of blades on each disc as occur in real rotors. Rzadkowski and Maurin [10] considered the influence of shaft flexibility on the dynamic characteristics of 4% mistuned and 1% mistuned bladed discs. Free vibration analysis showed that it is important to include the shaft when investigating several mistuned bladed discs since this can considerably change the spectrum of frequencies and mode shapes with zero, one, two and more nodal diameters.

2 DESCRIPTION OF THE MODEL

Figure 1 presents the analyzed aircraft engine rotor model. The main dimensions are as follows: the outer diameter of the largest - turbine disc is 0.512 m and the shaft length is 1.166 m. The number of blades on each disc corresponds to that of a real systems. Therefore there were 28 rotor blades in compressor stage I, 41 in stage II, 41 in stage III, 47 in stage IV, 57 in stage V, 47 in stage VI, 49 in stage VII and 83 in the turbine stage.

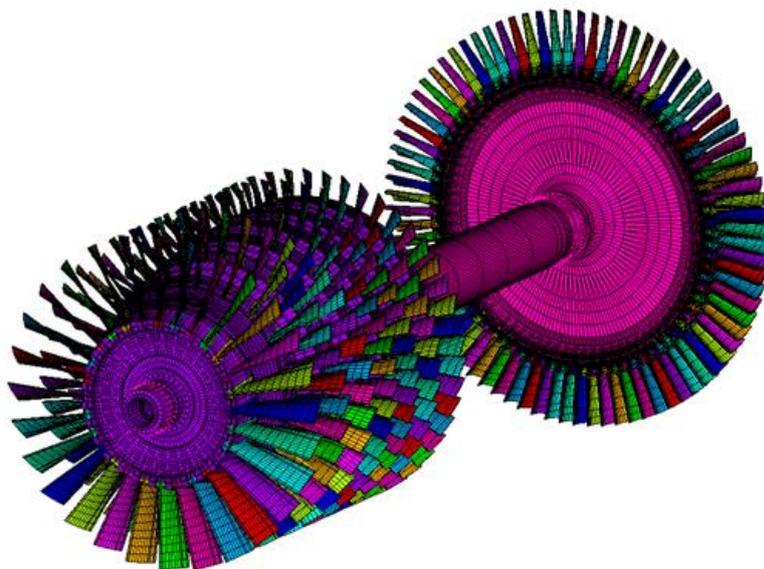


Figure 1: FEM model of the aircraft engine rotor.

Ansys Solid45 - isoparametric, extended brick elements with 8 nodes were used. The results obtained for these elements are equivalent to those of the Abaqus 20 node isoparametric brick element. Different mesh densities were used and a meshing convergence test based on natural frequencies was conducted. For the final calculations the smallest possible mesh density was applied. The final FE model in its entirety had just over a 1.5 million DOFs.

Two axis-symmetric bearing types were modeled as springs with $k_{xx}=k_{yy}=k_{xy}=k_{yx}=1400\text{N}/\mu\text{m}$ stiffness for two roller bearings and $k_{xx}=k_{yy}=k_{xy}=k_{yx}=1000\text{N}/\mu\text{m}$ for the ball bearing.

In this paper the influence of mistuning will be investigated by comparing numerical calculations with experiments conducted at the Air Force Institute of Technology in Warsaw. The obtained results are compared with those of a steam turbine analysis [10] to provide more general conclusions.

Two rotor models, tuned and mistuned, were developed for the numerical analyses. The blade mistuning was modeled using the modified Young modulus for every blade in each stage according to blade frequencies measured in real aircraft engine. Free vibration analyses were carried out in both models at 15,000 rpm.

3 MODEL DESCRIPTION

The natural frequencies of the rotating mistuned bladed discs were calculated using the ANSYS code. In both the tuned and mistuned models (henceforth referred to as case 1 and case 2, respectively) the natural frequencies for all the bladed discs were computed. The modes of each mistuned bladed disc were classified in an approximate way to those of the corresponding tuned bladed disc. In the case of mistuned systems, mode diameters up to two nodes could be analyzed using nodal line descriptions. However, this was not possible in the case of larger nodal diameter modes as the mistuned blades distorted the nodal lines of the mode shapes. The natural frequencies of tuned bladed discs on the shaft were divided into the modes dominated by the bladed discs and the modes dominated by the shaft with the discs.

The natural frequencies of the mistuned cantilever blades, single mistuned bladed discs and the complete flexible shaft with eight mistuned discs were carefully examined to find resonance conditions and coupling effects.

Figures 2, 5 and 7 present the zero-, one- and two-nodal diameter natural frequencies of tuned (a) and mistuned systems (b). The right axes indicate the natural frequencies of the eight discs (without blades) on the shaft, the middle axes show the natural frequencies of the mistuned bladed discs on the shaft, while the left axes present the uncoupled natural frequencies of single cantilever mistuned bladed discs corresponding to a given nodal diameter. The uncoupled modes of single blades and bladed discs were calculated separately. The frequencies of the turbine bladed disc are marked with the letter T, whereas those of the compressor are marked with the number (1-7) of their particular stage. The letter 'b' indicates single blade vibrations in a given stage. The longitudinal modes are presented in black, the torsional modes in green, the bending bladed disc modes in red and the bending shaft dominated modes in blue. Purple lines indicate couplings between single blades and entire stages. The lines connecting the natural frequencies are divided into two types: continuous lines indicating strong coupling and dashed lines for weaker coupling. Strong coupling occurs when the amplitudes of particular blades are very visible, whereas weak coupling occurs when the amplitude is relatively small. In the latter case we observe the vibrations of a shaft and disc without blades. The frequencies and mode shapes of mistuned bladed discs on a shaft were analyzed, starting from zero-nodal diameter modes.

3.1 Zero-nodal diameter modes

As shown in Figure 2, the first frequency mode corresponding to the zero-nodal diameter (132.46 Hz in case 1 and 132.44 Hz in case 2) was connected with a torsional rotor mode at 180.338 Hz, in both cases causing all the bladed discs to vibrate. The next frequency mode (478.16 Hz in case 1 and 478.02 Hz in case 2), presented in Figure 3, produce coupling between the compressor first stage bladed disc and the turbine disc. These vibrations were also visible on the unbladed rotor at 604.336 Hz, thus the line connecting the two frequencies in Fig 2. Figure 3 shows the influence of mistuning, revealing considerable differences in blade displacement between cases 1 and 2.

Similar mode couplings and distortions also occurred in subsequent zero-nodal diameter modes, up to 1187 Hz (see Figure 2). Some of the mode shapes observed in case 1 (i.e. at 1128.8 Hz etc.) did not appear in case 2 on account of the mistuning.

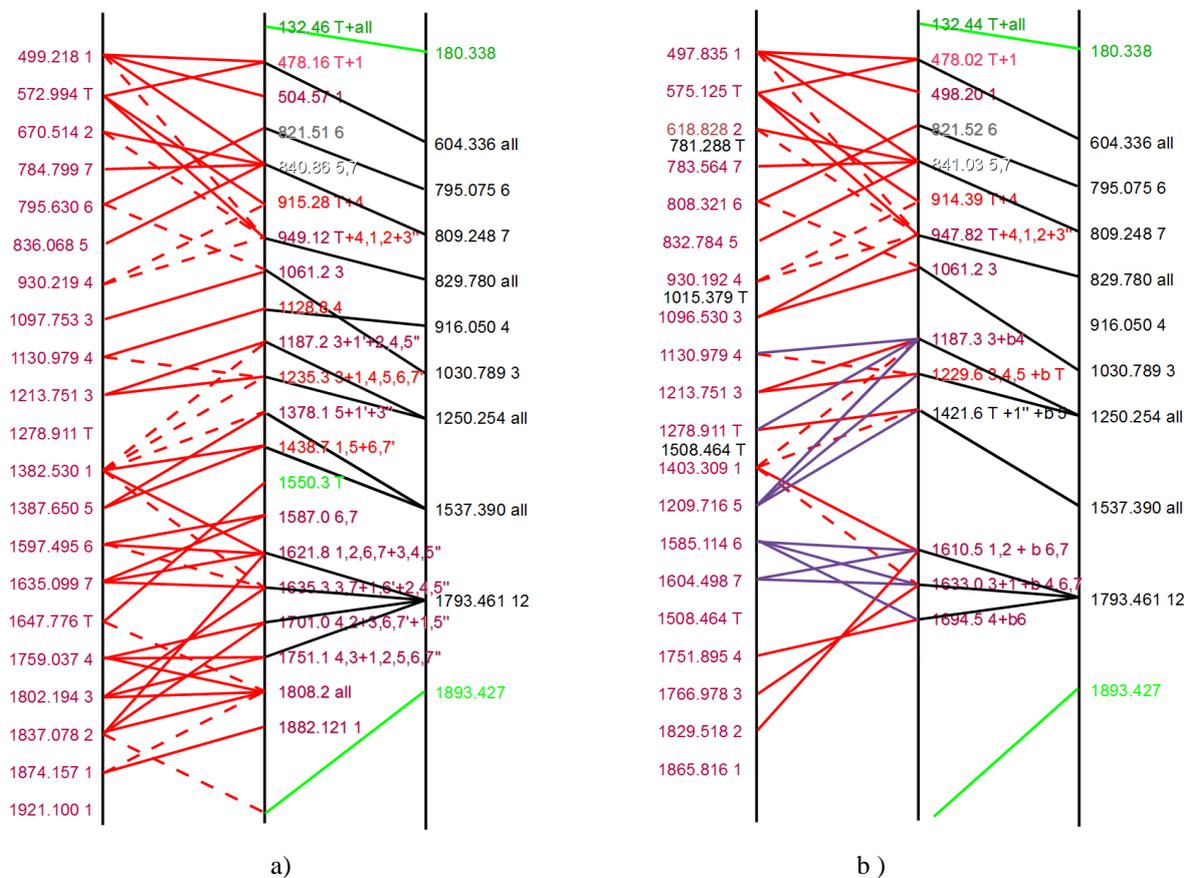


Figure 2: Natural frequencies of tuned and mistuned bladed discs on the shaft corresponding to zero-nodal diameter modes.

Generally we may conclude that mistuning alters the vibration amplitudes in particular blades. Each 1st stage blade in Figure 3 has a slightly different amplitude a mode with an almost zero-nodal diameter. This rule, however, may not hold for other modes.

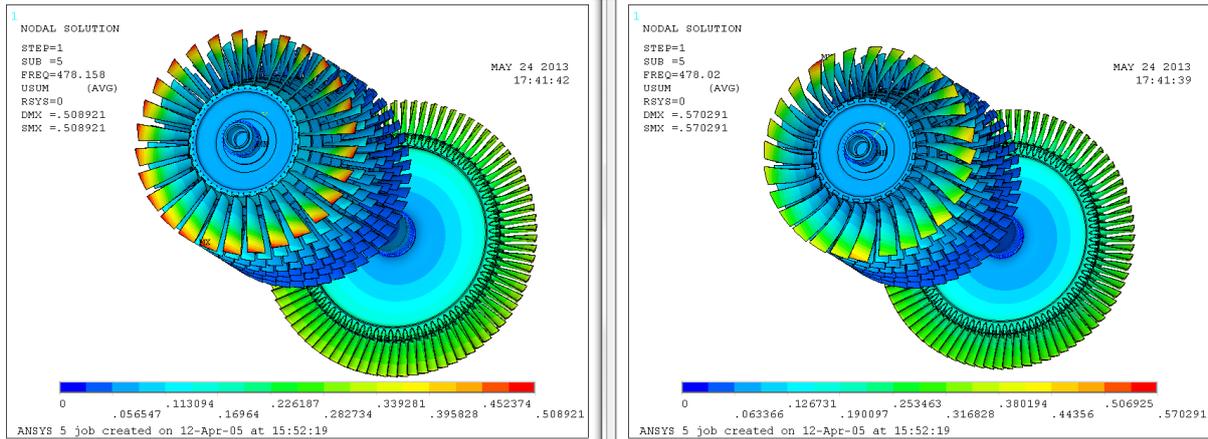


Figure 3: Comparison of mode shapes of tuned (left – 478.16 Hz) and mistuned (right – 478.02 Hz) bladed discs on shaft with a zero-nodal diameter.

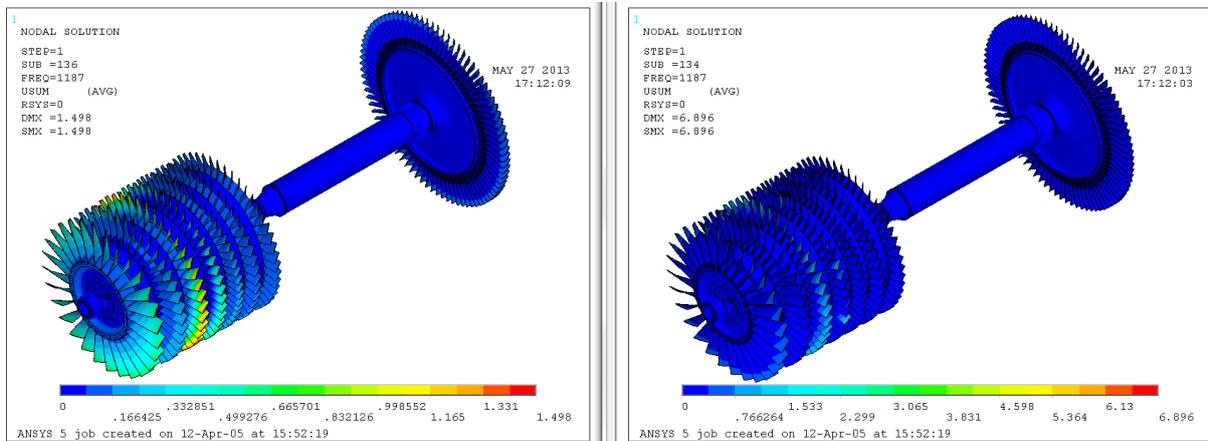


Figure 4: Comparison of mode shapes of tuned (left – 1187.2 Hz) and mistuned (right – 1187.3 Hz) bladed discs on shaft with a zero-nodal diameter.

When the tuned system is vibrating with a 1187.2 Hz mode coupling in the first five stages is visible (Fig. 4). Whereas in the mistuned system at 1187.3 Hz coupling is only visible in the 1st, 3rd and 4th stage with not all the blades vibrating. Moreover, the mode no longer has a zero-nodal diameter.

3.2 One-nodal diameter modes

Figure 5 presents the natural frequencies of tuned and mistuned bladed discs (a and b) on the shaft, vibrating a one-nodal diameter. The splitting of one-nodal diameter double frequencies occurs in the mistuned system as an effect of blade mistuning. The bending shaft dominated modes are presented in blue. The 371.08 Hz mode in Fig. 5a and the 371.01 Hz mode in Fig. 5b are dominated by the bending motion of the shaft caused by the one-nodal diameter mode shape of the turbine disc at 455.871 Hz. The cantilevered turbine disc in Fig. 5a vibrates at 523.866 Hz and at 533.554(603) Hz in Fig. 5b. Next, only the 1st bladed disc vibrates, with modes of 504.27 Hz in case 1, Fig. 5a. In case 2, Fig. 5b this mode cannot be identified.

Similar mode couplings and distortions can be observed in subsequent one-nodal diameter modes, up to the frequency of 1790 Hz (Fig. 5). Some of the mode shapes that appear in Fig. 5a (e.g. at 670.07 Hz etc.) do not occur in Fig. 5b on account of the mistuning.

Figure 6 presents one-nodal diameter vibrations in the 3rd stage of the tuned system at 1193.6 Hz (Fig. 5) whereas in at 1191.7 Hz in the mistuned system the one-nodal diameter mode is slightly distorted in 3rd stage and additionally individual blades also vibrate in the 4th bladed disc and turbine disc.

Figure 7 present one-nodal diameter modes in the tuned system at 1743.4 Hz and at 1736.0 Hz in the mistuned system (Fig. 5). In the tuned system all the compressor bladed discs are coupled, whereas in the mistuned system coupling occurs only between the 4th bladed disc and individual blades of the 6th and 7th bladed discs. A similar situation occurs in mistuned modes at 1468.5 (1469.7) Hz, 1522.4 (1523.1) Hz, 1665.1(1666.9) Hz, and 1861.4 (1863.4) Hz.

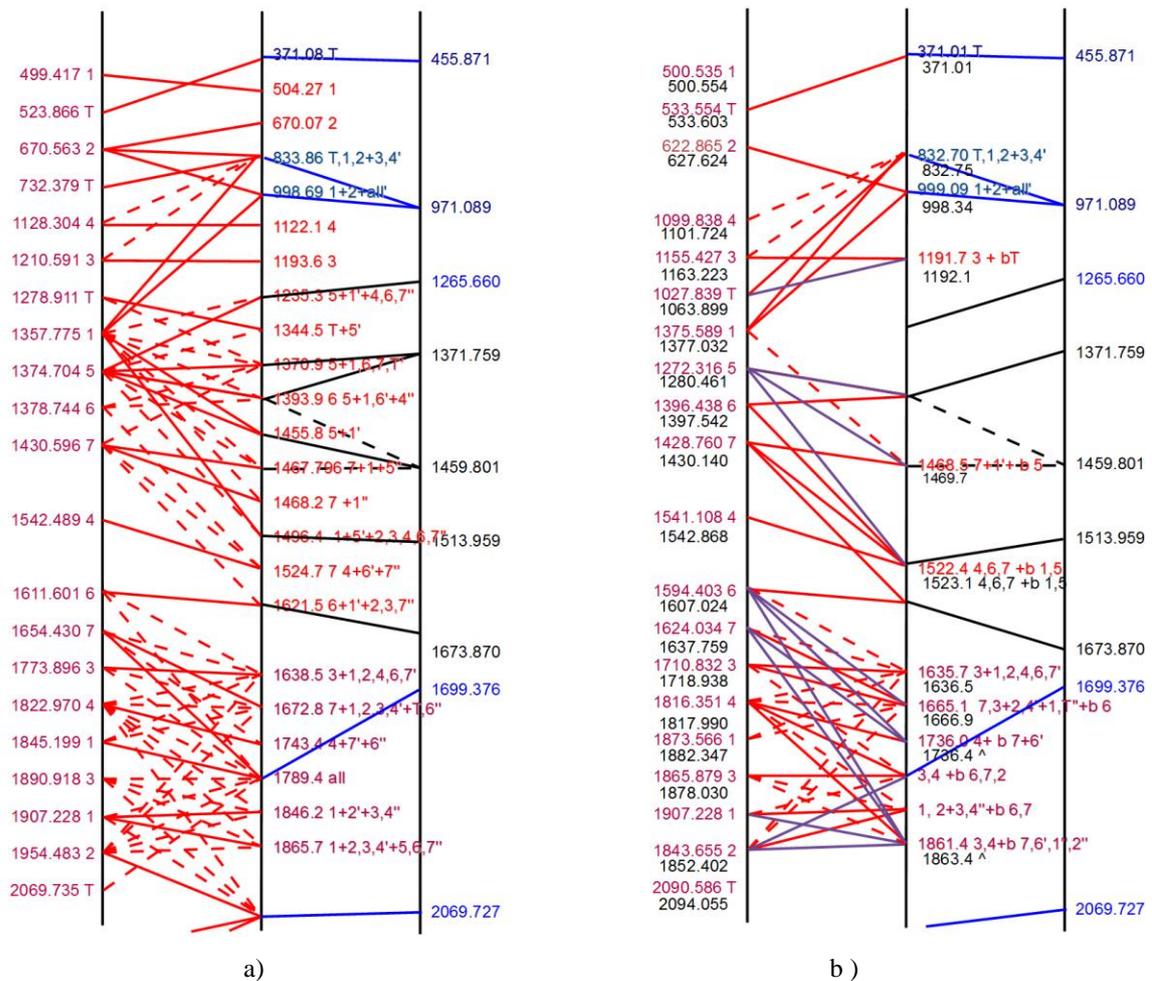


Figure 5: Natural frequencies of tuned and mistuned bladed discs on the shaft with one-nodal diameter modes.

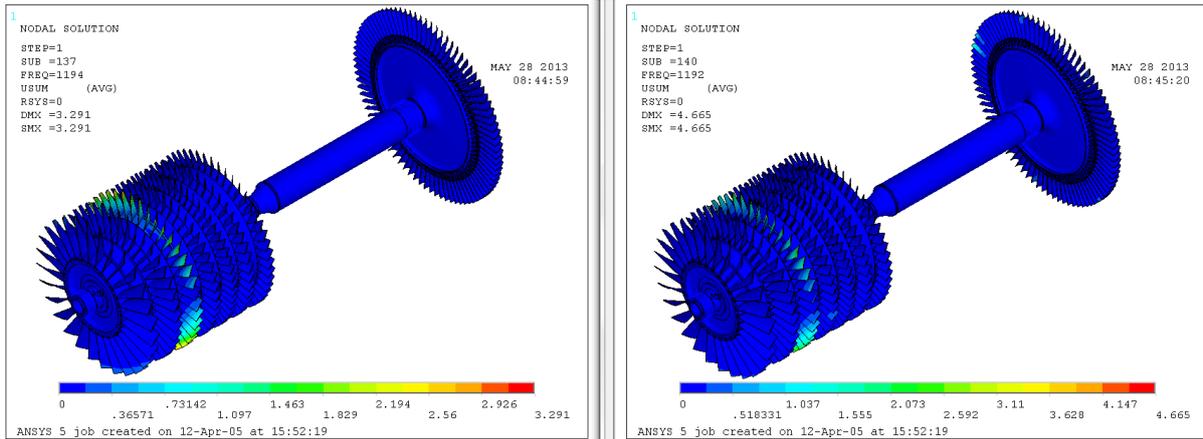


Figure 6: Mode shapes of tuned (left – 1193.6 Hz) and mistuned (right - 1191.7 Hz) bladed discs on shaft with one-nodal diameter.

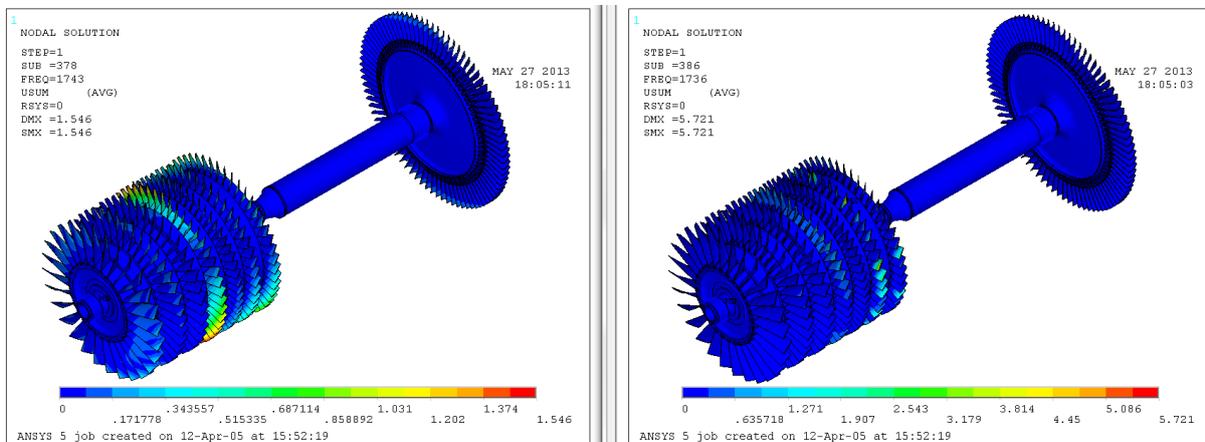


Figure 7: Mode shapes of tuned (left – 1743.4 Hz) and mistuned (right – 1736.0 Hz) bladed discs on shaft with one-nodal diameter.

We may conclude that the higher the mode, the greater the influence of mistuning.

3.3 Two-nodal diameter modes

Figure 8 presents the natural frequencies of tuned and mistuned bladed discs on a shaft with two-nodal diameter mode shapes. The splitting into two two-nodal diameter frequencies appears in the mistuned system on account of the mistuning.

At 499.47 Hz only the 1st bladed disc of tuned system vibrates with two-nodal diameter mode. A equivalent mode shape cannot be found in the mistuned system on account of the mistuning. The turbine bladed disc vibrates at 640.86 Hz in the tuned system and at 637.63(637.67) Hz in the mistuned system.

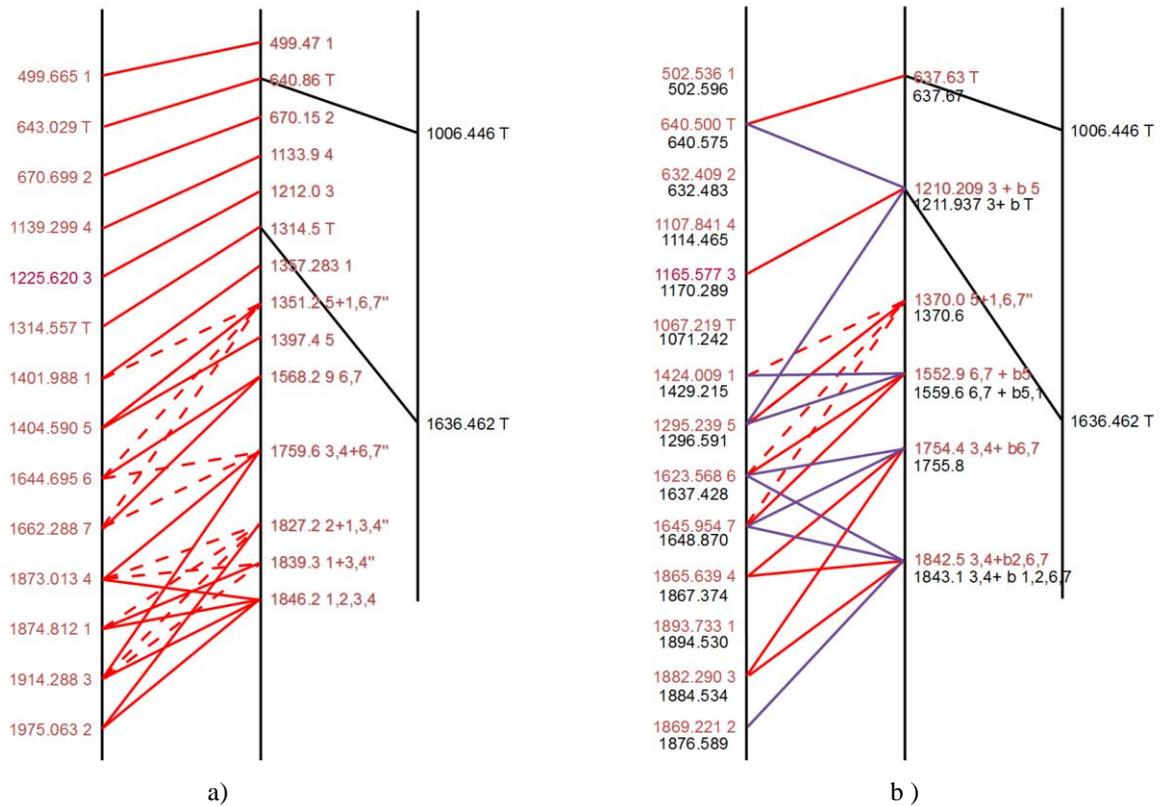


Figure 8: Natural frequencies of tuned and mistuned bladed discs on a shaft with two-nodal diameter modes.

Similar mode couplings can be observed in subsequent two-nodal diameter modes, up to 1760 Hz (Fig. 8). Some of the mode shapes observed in the tuned system (e.g. at 670.15 Hz) do not appear in the mistuned system on account of the mistuning.

Figure 9 shows that the two-nodal diameter vibration mode only appears in the 1st bladed disc of the tuned system at 1351.2 Hz (Fig. 5), whereas in the mistuned system coupling with this mode is observed in the 1st and 5th stages at 1370.0 Hz. In the latter case, however, only some of the blades in the 5th stage vibrate, while in the 1st stage all the blades vibrate with an almost two-nodal diameter.

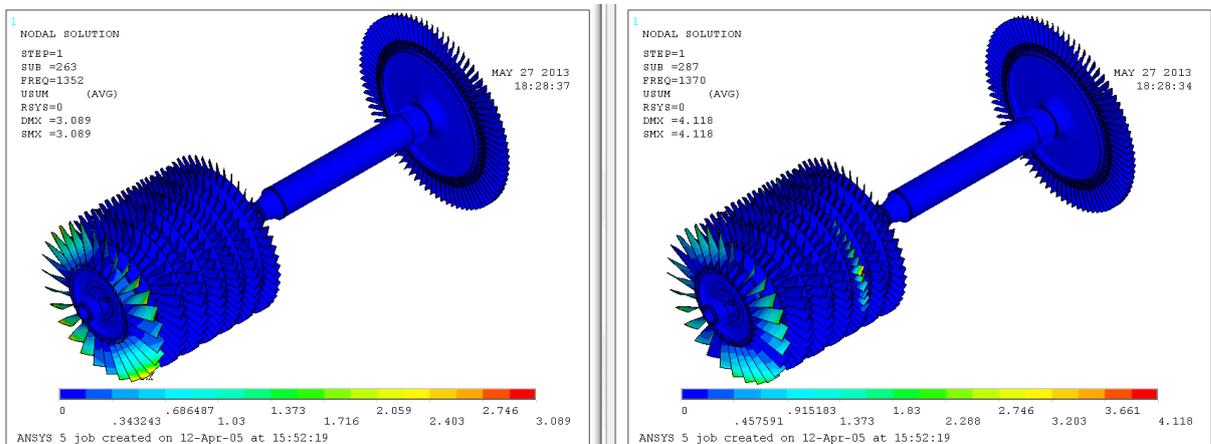


Figure 9: Mode shapes of tuned (left – 1351.2 Hz) and mistuned (right – 1370.0 Hz) bladed discs on a shaft with a two-nodal diameter.

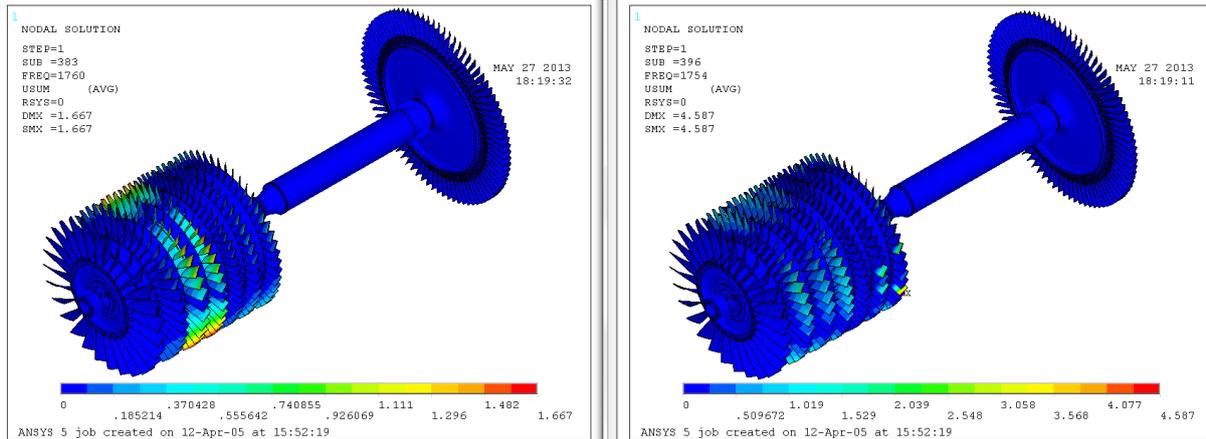


Figure 10: Mode shapes of tuned (left – 1760 Hz) and mistuned (right - 1756 Hz) bladed discs on a shaft with a two-nodal diameter.

In the tuned system at 1759.6 Hz there is strong, two-nodal diameter coupling between the 4th and 5th stage and weak coupling between the 6th and 7th.

In the mistuned system at 1756 Hz there is weak, two-nodal diameter coupling between the 2nd, 3rd and 4th stages as well as individual blades in 6th and 7th stages. Similar multistage coupling also occurs at other frequencies, such as 1842.5 and 1843.1 Hz.

4 CONCLUSIONS

Free vibration analysis of an aircraft rotor with 8 mistuned bladed discs showed that mistuning considerably increases multistage coupling when compared with a tuned rotor system.

The mistuning distorts the nodal diameters of mode shapes and causes multistage coupling between individual blades.

Free vibration analysis has also shown that mistuning changes the mode shapes and number of nodal diameters in particular mistuned bladed discs. Due to mistuning, the number of resonances and couplings in the system may change.

All above conclusions are congruent with the ones presented for steam turbine systems [10].

Free vibration analysis has not sufficiently explained multistage coupling in tuned bladed discs. To better understand this problem, forced vibration analyses should to be carried out.

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