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Analysis for Calculation of Forced Response of Gas Turbine Blade under Friction Damping

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Abstract: Gas turbine is one of the satisfactory power-developing units among various means of producing mechanical power from units of comparatively small size and weight. Gas turbine technology is widely used both in propulsion engines and for stationary energy conversion in power plants. In the propulsion sector gas turbines are commonly used in military and civil air craft engines and also in marine propulsion and space propulsion engines. The developments may also leads to vehicular gas turbine drives.

In order to reduce the high fatigue risk of blade wheel cycles, the prediction of vibration levels is at an early stage of the design process important. Therefore, the different sources of damping must be modeled accurately. This article discusses the influence of friction in the blade attachments to corrective measures are investigated experimentally. An efficient multiharmonic balance method is used to calculate the forced response of blade disks with contact and friction nonlinearities in blade roots. For experimental validation purposes, a rotating blade disk was tested in a vacuum chamber with the excitation provided by Piezo actuators. A model of the rig was built with a normal load-dependent coefficient of friction and a constant material damping ratio. The nonlinear behavior experimentally observed at resonances was well reproduced. An acceptable correlation was found with experimental resonance frequencies, amplitudes and degree of damping over the entire range the spinning speed and the excitation level range. The proposed experimental method can therefore serve to improve the prediction of the changing stresses in blade disc assemblies.

Keywords: Vibration, damping, friction, blade, disk

I. INTRODUCTION

In this new period of industrialization, vitality or power is said to be the essential need through which any industry can advance along the rails of thriving. Gas turbine is one of the palatable power-creating units among different methods for delivering mechanical power from units of nearly little size and weight. Gas turbine innovation is generally utilized both in impetus motors and for stationary vitality change in power plants. In the impetus area 9 turbines are normally utilized in military and common air create motors and furthermore in marine drive and space drive motors. The improvements may likewise prompts vehicular gas turbine drives.

Most segments in gas turbine motor are presented to vibrations brought about by precarious powers because of relative movements of pivoting and non-turning parts. Because of vibrations, weakness stresses will be created in the sharp edges and in the rotor and lessens its execution. So the productivity of the Gas Turbine will be expanded by lessening the vibrations. As of now, look into is centered on vibration of turbine sharp edges.

Gas turbines (GT) are one of the huge parts of current industry. They assume a key job in aeronautical industry, control age, and primary mechanical drivers for vast siphons and blowers. Displaying and reproduction of gas turbines has dependably been an integral asset for execution advancement of this sort of hardware. Astounding examination exercises have been done in this field and an assortment of investigative and trial models has been fabricated so far to get inside and out comprehension of the nonlinear conduct and complex elements of these frameworks. Notwithstanding, the need to create exact and solid models of gas turbines for various goals and applications has been a solid inspiration for scientists to keep on working in this interesting territory of research. The investigation in this field incorporates white-box and black-box based models and their applications in control frameworks. Counterfeit neural systems (ANNs) as a discovery strategy have been viewed as appropriate and integral assets for information preparing, displaying, and control of very nonlinear frameworks, for example, gas turbines. Moreover, on account of the extreme interest of the power showcase, the power makers are anxious to consistently explore new techniques for streamlining for configuration, assembling, control and support of gas turbines. Gas turbine is as an inside burning motor which utilizes the vaporous vitality of air to change over concoction vitality of fuel to mechanical vitality. It is intended to extricate however much as could be expected of the vitality from the fuel [1]. The administration of gas turbines in modern hardware and utilities situated on power

plants and seaward stages has been expanded in the previous 50 years. This intense interest is a result of their low weight, smallness and various fuel applications [2]. In spite of the fact that the narrative of gas turbines has taken a root ever, it was not until 1930s that the primary down to earth GT was created by Frank Whittle and his associates in Britain for a stream air ship motor. Gas turbines were created quickly after World War II and turned into the essential decision for some applications. That was particularly a direct result of upgrade in various zones of science, for example, optimal design, cooling frameworks, and high-temperature materials which essentially enhanced the motor proficiency.

A. Gas Turbine Performance

Gas turbines work dependent on Brayton cycle. Figure 1 demonstrates a commonplace single-shaft gas turbine and its principle segments including blower, burning chamber (combustor), and turbine. The arrangement of these segments is called motor center or gas generator (GG). Blower and turbine are associated by the focal shaft and pivot together. Figure 2 shows standard Brayton cycle in weight volume (P-V) and temperature-entropy (T-outlines individually [3]. Air enters the blower at segment 1 and is compacted through passing the blower. The hot and compacted air enters the burning chamber (combustor) at area 2. In combustor, fuel is blended with air and lighted. The hot gases which are the result of burning are constrained into the turbine at segment 3 and turn it. Turbine drives the blower and the GG mechanical yield, which can be a power alternator in a power plant station, an extensive siphon or a substantial blower. The perfect procedures in the blower (1-2) and turbine (3-4) are isentropic. There is likewise an isobaric procedure in the combustor (2-3) and condition (4-1) for the perfect cycle. Be that as it may, the genuine procedures in the blower and turbine are irreversible and non-isentropic. There is additionally weight misfortune amid the procedure in the combustor. Disregarding the weight misfortune noticeable all around channels and the combustor, forms 2-3 and 4-1 can be considered isobaric [4].
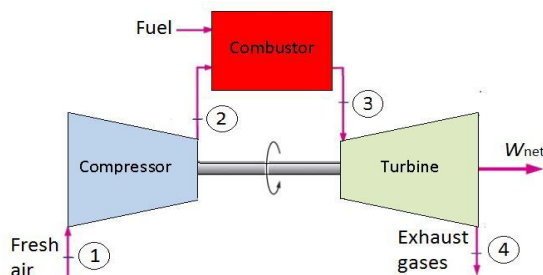

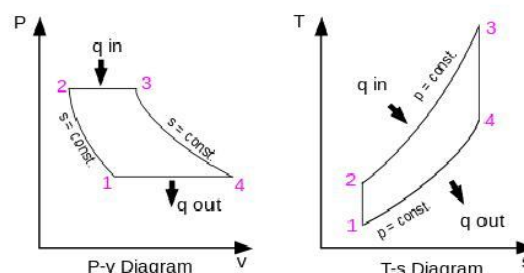Fig. 1 Schematic of a typical single-shaft gas turbine

Fig. 2 Typical Brayton cycle in pressure-volume and temperature-entropy frames [8].

II. IDENTIFICATION OF INDIVIDUAL BLADE VARIATION

When the predictions of a specific numerical code should be compared with collected experimental data, the degree of correlation depends not only on the prediction tool capabilities to capture the dynamic characteristics of the structure, but also in how well the structure is described to the prediction tool. The wrong answer prediction tools are no exception to this. When validating these codes, special attention should be given to the identification of the true mistuning parameters. On the other hand, inherent misidentification may benefit Research efforts that seek to reduce sensitivity to confusion by introducing intentional mistuning proper. Therefore, it is considered appropriate to inform some of the proposed methods for misidentification.

Mignolet et. Alabama and Rivas-Guerra et. Alabama. [5-6] focused on the estimates of the mass and stiffness variations of the blades in a two-part investigation. They the lowest measured frequencies of the isolated blades were used to recover variations in Properties of mass and rigidity two methods are presented. The first assumes variation of the mass matrix is negligible and associates the distortion with the rigidity. Only variations however, in this way it is found that significant errors are introduced in estimates of forced response. In the second method, maximum probability strategy (ML) is used in which a simple Gaussian distribution of Matrices of mass and rigidity are assumed and all the structural parameters are constant. However, this technique is only applicable to sheet discs with separate blades and it is assumed that the individual blades behave in the same way when they are assembled.

A method of misidentification that predicts the individual stiffness of the leaf. Variations of measurements on a complete blade disc are given by Judge et. Alabama. [7] This method, unlike the previous one can be applied to blisks. A very reduced the order model

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[8] in which the modal rigidities of the blade are insulated is used. The variations of these rigidities are calculated based on the measurement data for evaluate the degree of deviation of a tuned reference. The capability of the method is demonstrated in an experimental sheet disk. Although the variation of the intentional distortion imposed in the experimental test is reasonably well predicted, significant deviations are observed in the quantification of mistuning. Feiner and Griffin proposed a completely erroneous experimental identification method [9-10] that requires only a set of flattened disk modes measured and resolves the confusion as a sector-to-sector frequency deviation in relation to corresponding refined reference. Unlike the method proposed by the judge et. Alabama. That It only works for an isolated family of modes. However, the model reduction methodology involved is less complicated and requires less analytically input data generated. His method is based on the so called Fundamentals Disorganization model (FMM) [11] in which a model of sector by sector is very small of a flattened disc is used to predict the vibratory response in an isolated family of modes. The effectiveness of the method is demonstrated experimentally in a test sample.

III.EXPERIMENTAL METHOD

A. Test Bench
The experimental data were obtained with a test facility designed to better understand the friction phenomena in. In the present study, the only friction zones considered the dovetail connections between the disc and the blade. A schematic view of the test stand used can be seen in Figure 3. The rotating disk supports four blades and is placed in a vacuum chamber to minimize the effects of aerodynamic forces. The disc rotates on a hollow shaft, which is supported by two ball bearings and driven by an electric motor. The blades considered here are the compressor blades .
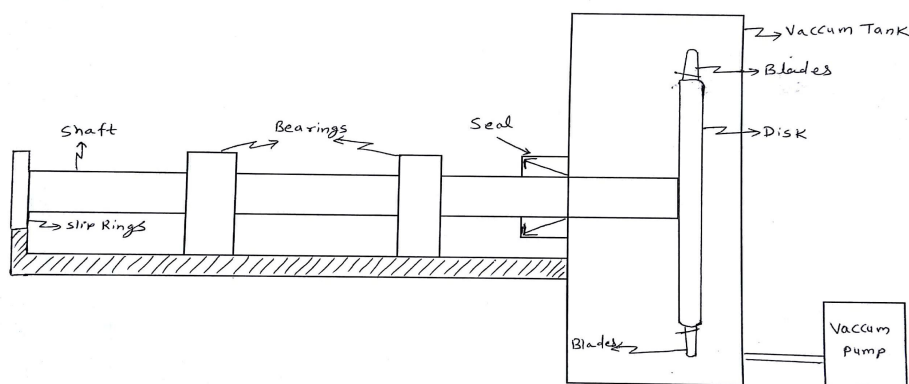

Fig. 3 Schematic view of the test bench.

B. Instrumentation
The four blades are arranged as two diametrically opposite pairs around the disk. Two adjacent blades are equipped with piezoelectric actuators and Strain gauges while the other two remain bare. The two equipped blades, here called blade 1 and blade 2.The lead zirconate titanate (PZT) ceramics 1 cm square flat layers with a thickness of 1 mm are glued to the wings with an insulating epoxy adhesive and adapt to the slightly curved surface. Two ceramics are used for each blade, one on the print side and one on the Suction side. Parallel wiring together with proper orientation of the polarization directions allows them to operate outside of it Phase and generate a flexion movement with a common voltage signal. The PZT layers are located in heavily loaded regions the first bending mode to maximize effectiveness the excitement a slip ring at the extreme end of the wave is used to transmit excitation and measurement signals.

C. Measurements
The first step of the experimental procedure is the pumping of the Air from the vacuum chamber. The results are shown here Paper was obtained with a stabilized pressure of 20 mbar. Then the blade is rotated. Five rotation Speeds were examined from 1000 rpm to 5000 rpm. In order to study the frequency response around the modes of interest, the actuators were provided with a swept sinus excitation sufficiently slow sweep rate to avoid artificial distortion of responses at low damping. For every spider Speed, voltage levels up to 100 V can be applied to the piezoelectric ceramic. In this study, the retained values were 10V, 20 V, 40 V and 80 V. Frequency response functions were provided only by measuring the fundamental component and the half power bandwidth method was used in isolation Resonance peak to evaluate the global damping extent in the system.

IV. EXPERIMENTAL RESULTS

A. Results

This section compares the experimental responses with numerical frequency responses by shown by D. Charleux et. al. [12]. Damping is mainly focused on a specific mode of the blade . A comparison between experimental and numerical frequency responses is shown in Fig. 4 (a), 4 (c) are the results for blade 1. and Fig. 5 (b), 5 (d) represent those for blade 2. A good Agreement between numerical and experimental results observed in both amplitude and phase curves. Resonance from mode 1 is visible in the answer to sheet 1. By sharing the amplitudes by the excitation level, as this resonance a linear behavior could be observed. These was also the case with the simulations, since all contact elements remained in the bar state. The observed discrepancy therefore most likely means that the material damping was underestimated for this particular speed. For mode 2 The experimental peaks are shifted slightly to the left as Ua is increased from 10V to 80V, which is not well reproduced through the simulation. Suspected causes include the level the discretization and the lack of tangential contact stiffness in the model.

Table I: Coefficients of friction used in the model

SPINNING (RPM)	NORMAL FORCE RATIO	COEFFICIENT OF FRICTION
1000	1	0.15
2000	4.25	0.092
3000	9.80	0.069
4000	17.67	0.056
5000	27.86	0.048

B. Resonant Frequencies

Figure 6 shows the evolution of the resonant frequency for mode 2 over the speed range. These results are obtained for the lowest excitation level, which minimizes the frequency shift due to the nonlinearity. The numerically predicted centrifugal stiffening is in good agreement with the experiment. But one can notice that the two curves do not have exactly the same shape. The experimental curve is almost straight, while the numerical curve bends a little. One possible explanation for this may be that the model does not include contact stiffnesses. Indeed, experiments reported in Ferrero et al. [13] and Crassous et al. [14] have shown that the tangential contact stiffness increases with normal load. Including such normal load dependent contact stiffnesses in the model would straighten the numerical curve.

C. Amplitudes and Damping

The maximum amplitude at which the blades vibrate is a key information for the designer interested in preventing high cycle fatigue problems. Dynamic stresses together with static stresses are necessary to verify if the blades remain sufficiently under their endurance limit. Figure 7 shows the maximum vibration levels of the bladed disk for the resonant response of mode 2. First of all, it is important to stress that for a speed of 5000 rpm and an excitation of 10 V, the amplitudes obtained numerically are very close to the experimental ones. In this case, energy dissipation in blade roots is minimal and behavior is almost linear. Furthermore, one can note that in Figure 8 (a) the measured and numerical damping ratios are very close. So with the same amount of damping, both computations and experiments give the same amplitude, demonstrating the acceptable accuracy of the simple model used for piezoelectric actuation. Figure 7 also reveals that, to a good extent, the numerical model reproduces the main trends observed in the experiments. Nonetheless, one point of interest seen in (b) is that the slope of the strain versus speed curves obtained with the numerical model can be negative for high speeds and low amplitudes, whereas the experimental curves always exhibit a positive slope. Kielb and Abhari [15] tested a rotating bladed disk in a similar facility at speeds up to 20 000 rpm and also found ever increasing amplitudes. The fact that structural damping model does not take into account the speed and prestress influence due to rotating speed can explain the discrepancies between experimental and experimental slopes.
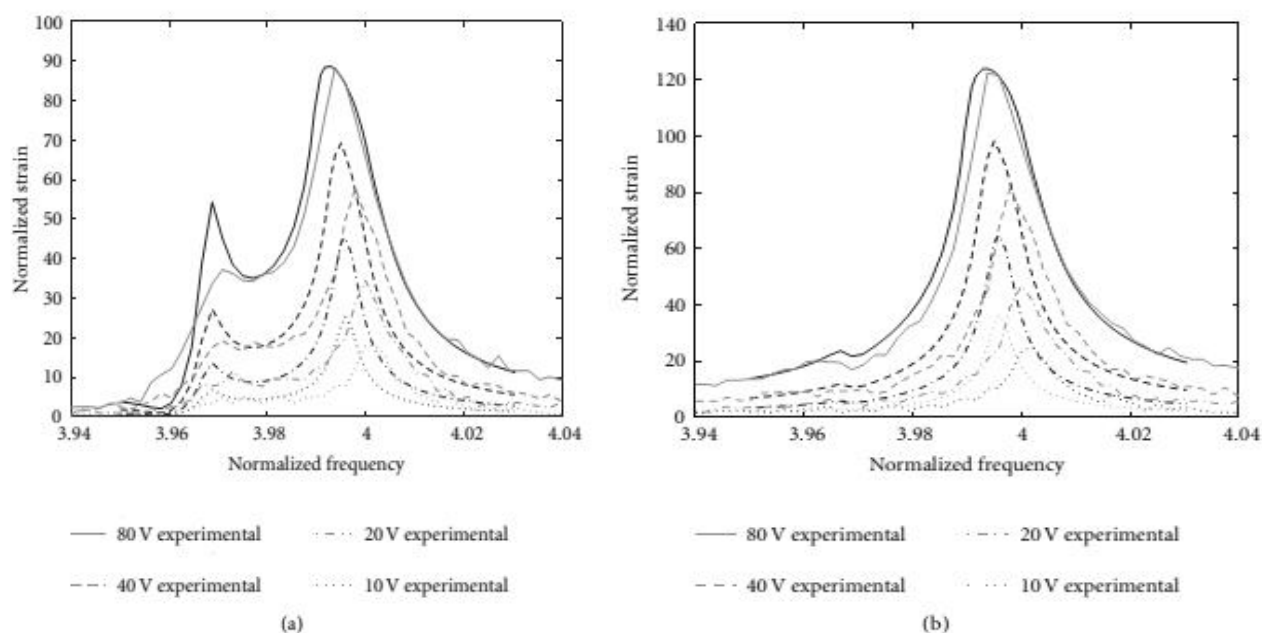
Figure 4 (a) & Figure 5.4 (b): Numerical and experimental frequency responses at 3000 rpm.

For the highest recorded amplitude (sheet 2, 5000 rpm 80 V), the maximum calculated alternative von Mises stress in The airfoil portion of the blade is about 20 MPa. This is more than twenty times below the duration limit. However, the excitation level was sufficient to produce significant nonlinearity for each velocity. The results shown in Fig. 8 confirm that the overall attenuation decreases with increasing spinning speed. The The attenuation values are of the same order of magnitude as these found by Tokar 'et al. [16] and Kielb and Abhari [15].
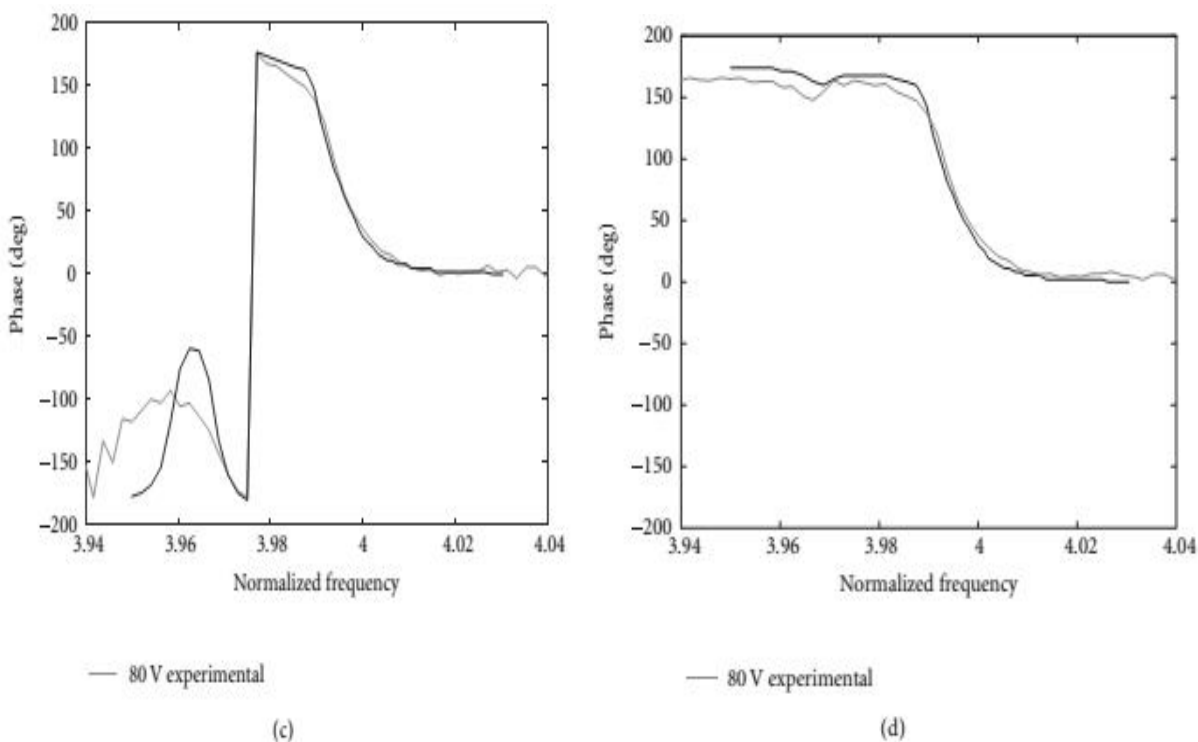
Figure 5 (c) & Figure 5.4 (d): Numerical and experimental frequency responses at 3000 rpm.

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Experimental curves are even fairly well reproduced by simulation when damping is greatly underestimated at low vibrations (Curves (a) and (b)). The advantage of considering the friction in the blade root is obvious in this case as a linear calculation would have generated horizontal lines with a damping ratio from $5.5 \times 10\text{-}4$.Possible causes for the observed deviations it is assumed in the model that the friction is in the dovetail Joints are the only source of nonlinearity. For example as I said Previously, the possible variation of the material damping with the vibration amplitude was not considered. In addition, as the speed increases, the centrifugal stiffening could be responsible for a decrease in material damping. Smith and Wereley reported such a phenomenon with rotating composite supports (Smith and Wereley [17]), but no similar experimental work with titanium alloys was found the literature.
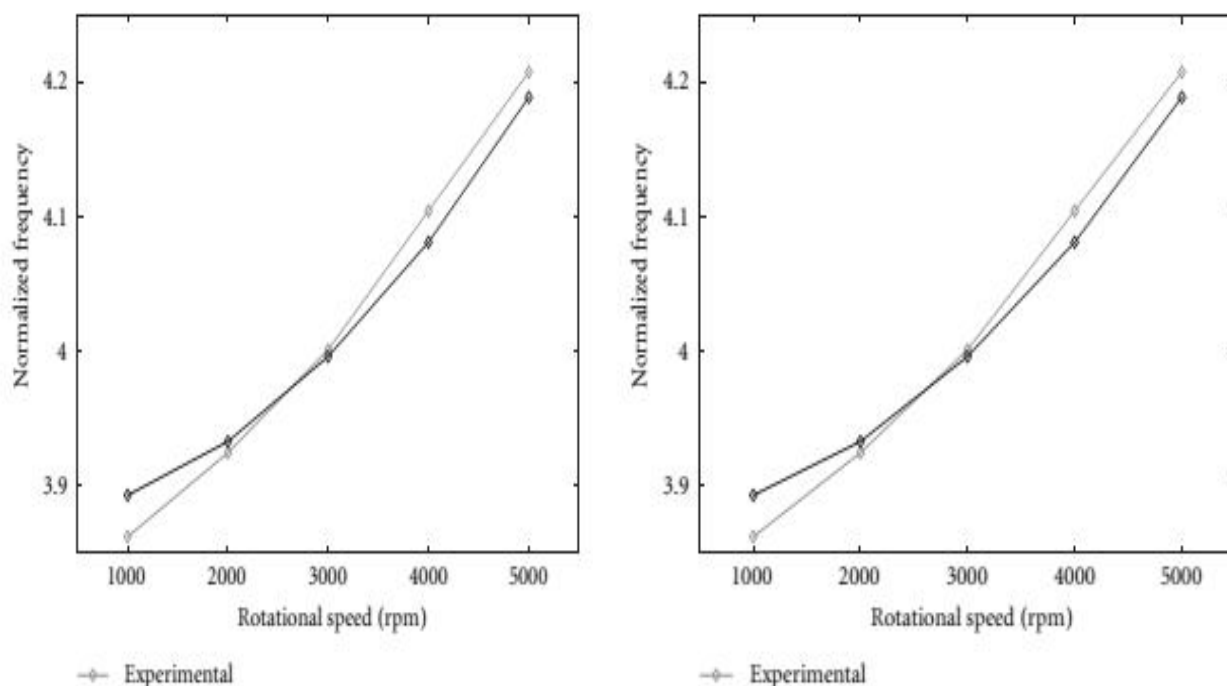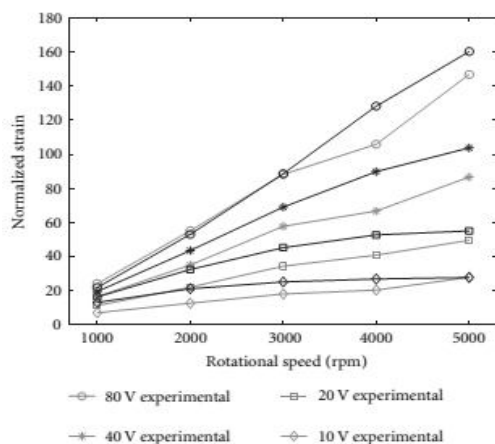
Figure 6: Numerical and experimental resonant frequencies for Ua= 10 V.

Another source of error refers to the number of contact elements for the simulation. Sinclair et al. [18] showed that highly refined gratings are required to achieve converged contact stresses in dovetail joints, but such discretizations are not compatible with dynamic calculations.In this A compromise study was found and 24 contact elements per Flank were used. This is not enough to predict exactly Contact stress distribution and therefore also causes errors in the estimation of slip ranges and slip amplitudes.

V. CONCLUSIONS

An attempt to predict the nonlinear forced response of Discs with friction in the blade fixtures were presented. To calculate the steady-state reaction, a DLFT method was used, taking into account the friction of smooth Coulomb friction and the one-sided contact laws. To check this method, experimental testing of a vacuum rotating blade Chamber was performed. With harmonic excitement The measured frequency response showed a nonlinear behavior near resonances, due to the leaf root friction. A three-dimensional finite element model of the rig was designed and updated to accommodate the system mismatch Sound.

The simulations were done with normal load dependent coefficient of friction and a constant material Damping. Under these assumptions experimental damping Layers were reproduced with reasonable accuracy for different speeds and different levels of excitation. The method can therefore be used to better predict this changing stresses in blades. That watched Discrepancies between numerical and experimental results can be due in particular to the number of contact elements retained which was insufficient to accurately predict the distribution of contact stress. Contact stiffnesses, that may vary with normal load, were not included in the model presented in this paper and could also play an important role. There is a need for more experimental data to enhance the level of modeling and the precision of numerical results. In particular, material damping should be determined alone as well as its variation with amplitude and centrifugal load.
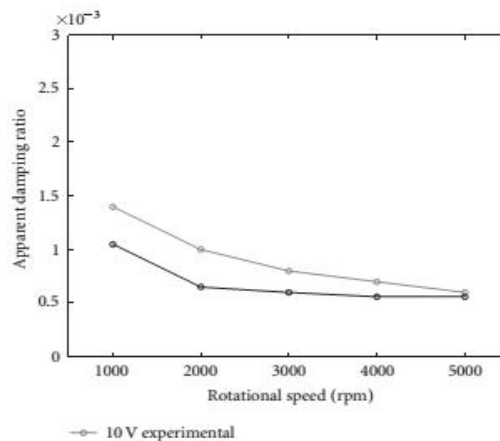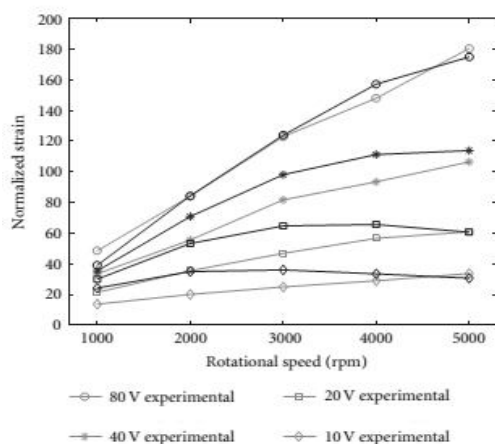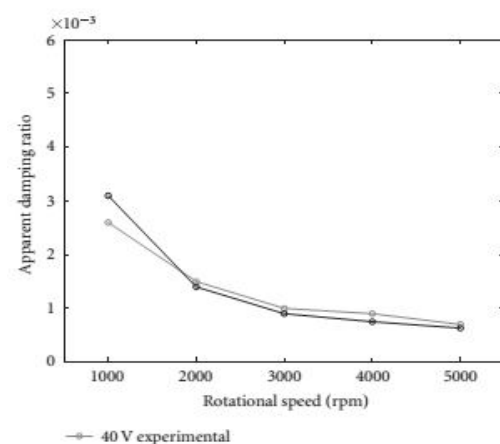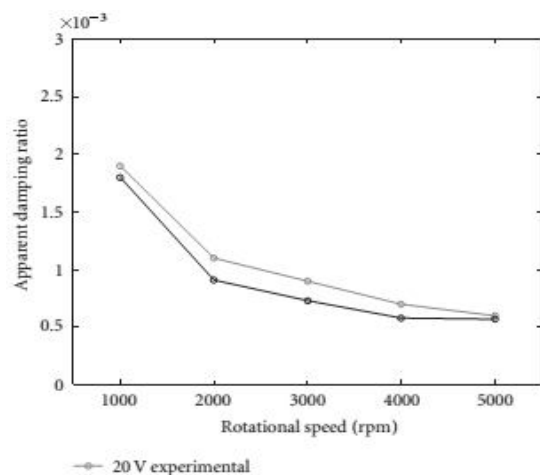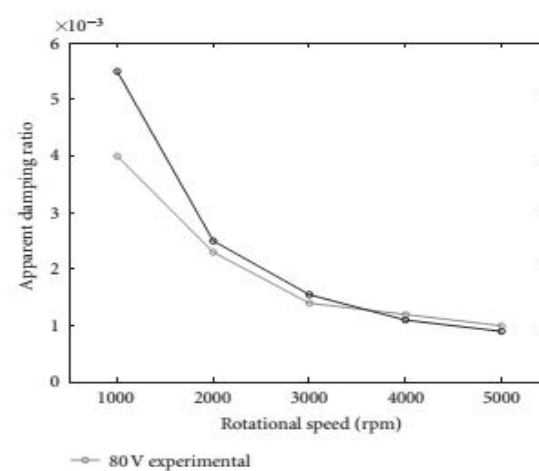
Fig. 7 : Numerical and experimental Figure
maximum vibration levels: (a) blade 1, (b) blade 2.

Fig. 8: Numerical and experimental equivalent
viscous damping ratios: (a) 10 V, (c) 40 V

Fig. 8 : Numerical and experimental equivalent viscous damping ratios: (b) 20 V,(d) 80 V

REFERENCES

[1] Ewins, D. J., Structural dynamics characteristics of bladed disks AGARD Manual on aeroelasticity in axial-flow turbomachines, Vol. 2, 1988

[2] Ewins, D.J. ,The effect of blade mistuning on vibration response – a survey IFToMM 4th Int. Conf. on Rotordynamics, Prague, Czechoslovakia, 1991

[3] Myhre, M., Moyroud, F.,Fransson, T. H. , Numerical investigation of the sensitivity of forced response characteristics of bladed disks to mistuning ,Proceedings of ASME Turbo Expo 2003, June 16-19, 2003, Atlanta,USA

[4] Kenyon, J.A., Griffin, J. H. Mistuned bladed disk forced response with frequency veering Proceedings of 8th National Turbine Engine HCF conf., 14-16 April 2003, California

[5] Mignolet, M.P., Rivas-Guerra, A.J., and Delor, J.P.Identification of Mistuning Characteristics of Bladed Disks from Free Response Data - Part I Journal of Engineering for Gas Turbines and Power, Vol. 123(2), 2001, pp. 395-403.

[6] Rivas-Guerra, A.J., Mignolet, M.P., and Delor, J.P.Identification of Mistuning Characteristics of Bladed Disks from Free Response Data –Part II Journal of Engineering for Gas Turbines and Power, Vol. 123(2), 2001, pp. 404-411.

[7] Judge, J.A., Pierre, C., and Ceccio, S.L. Mistuning Identification in Bladed Disks Proc. of the Int. Conf. on Structural Dynamics Modelling, 2002, Madeira,Portugal.

[8] Bladh, R., Castanier, M. P., Pierre, C. Component-mode-based reduced order modeling techniques for mistuned bladed disks – Part I: Theoretical models ASME Journal of Eng. for Gas Turbines and Power, Vol. 123, 2001a, p. 89-99.

[9] Feiner, D.M., and Griffin, J. H.Mistuning Identification of Bladed Disks Using a Fundamental Mistuning Model – Part I: Theory ASME Paper GT2003-38648, Int. Gas Turbine Institute Turbo Expo, Atlanta, Georgia.

[10] Feiner, D.M., and Griffin, J. H.Mistuning Identification of Bladed Disks Using a Fundamental Mistuning Model – Part II: Application ASME Paper GT2003-38648, Int. Gas Turbine Institute Turbo Expo, Atlanta, Georgia.

[11] Feiner, D. M., Griffin, J. H. A fundamental mode of mistuning for a single family of modes, Proceedings of ASME TURBO EXPO, 2002, GT-2002-30425.

[12] D. Charleux et.al. , "Numerical and Experimental Study of Friction Damping in Blade Attachments of Rotating Bladed Disks", International Journal of Rotating Machinery, Volume 2006, Article ID 71302, Pages 1–13

[13] J. F. Ferrero, E. Yettou, J. J. Barrau, and S. Rivallant, "Analysis of a dry friction problem under small displacements: application to a bolted joint," Wear, vol. 256, no. 11-12, pp. 1135–1143,2004.

[14] J. Crassous, S. Ciliberto, E. Charlaix, and C. Laroche, "Hysteresis and recovery length in a dry solid friction experiment," Journal de Physique II, vol. 7, no. 12, pp. 1745–1751, 1997.

[15] J. J. Kielb and R. S. Abhari, "Experimental study of aerodynamic and structural damping in a full-scale rotating turbine," ASME Journal of Engineering for Gas Turbines and Power, vol. 125, no. 1, pp. 102–112, 2003.

[16] I. G. Tokar', A. P. Zinkovskii, and V. V. Matveev, "On the problem of improvement of the damping ability of rotor blades of contemporary gas-turbines engines," Strength of Materials,vol. 35, no. 4, pp. 368–375, 2003.

[17] C. B. Smith and N. M. Wereley, "Transient analysis for damping identification in rotating composite beams with integral damping layers," Smart Materials and Structures, vol. 5, no. 5, pp. 540–550, 1996.

[18] G. B. Sinclair, N. G. Cormier, J. H. Griffin, and G. Meda, "Contact stresses in dovetail attachments: finite element modeling," ASME Journal of Engineering for Gas Turbines and Power, vol. 124, no. 1, pp. 182–189, 2002.

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