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DESIGN AND VIBRATION ANALYSIS OF ROTOR BLADE DISK AND INTEGRATED BLADED DISK OF A GAS TURBINE COMPRESSOR

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All the components in a gas turbine engine undergo overhauling, maintenance and assembly after a specified time duration. Some of these are completely replaced irrespective of their condition reducing the chances of failure at any point of time. The same is applicable to compressor rotor blades where removable blade setup is replaced with a Blisk. A Blisk is an integrated single disk and blade arrangement which is designed ensuring no change in flow path geometry and disk cross section. This resulted in reduction of compressor weight and improvement in stability. Pre-stress modal analysis concept is used to carryout current analysis to find out natural frequencies of the blade at varying angular velocities. Here the model is stressed under the speed of the rotor wheel and the model is then vibrated to find its natural frequency under the stressed condition. Campbell diagrams are then constructed for various nodal diameters to find the interference margins of the fundamental mode frequency like 1st bending and 1st Torsion.

***Keywords*:** Bladed disk, FEA, Pre-Stress modal Analysis, Cyclic Symmetric Analysis.

1. Introduction

The mechanical integrity of aero engine discs and attached blades is crucial to the operational safety and service life of gas turbine engines (GTE). The mechanical joint between compressor blade and the disc represents the most critical load path between the blades and the disc. Contact stresses, interface conditions and the detailed geometry of the joint determine the severity of the resulting stress field. Cracks usually develop in the dovetail region and the designers must understand fully the consequence of varying the geometry or the interface conditions. This stress field during service loads of GTE can induce fatigue. Only centrifugal loads are considered in this study, since it is known that they produce much larger stresses in compressor disc due to thermal, bending, twisting, and vibratory loads. The severe thermal cycling associated with each start-up and shutdown of the engine, together with centrifugal loads and high-frequency excitation problems, causes significant stress intensification at the bore of the rotating compressor disc. Fretting is the damage process associated with the cyclic relative motion of the blade and the disc, resulting in surface damage leading to localized shear traction and micro-slip at the contact surface.

Fig 1: Rotor with disk and blades

Rotor consists of the disk and blades, these blades are connected to the disk on outer diameter of the disk slots. The major disadvantages of rotor with slotted disk, chances for crack initiation in the dovetail slot area due to thermal difference. Number of components in the compressor increased due to individual parts of blades. Rotor with disk and blades geometry shown in Figure 1

Fig 2: Integrated Bladed Rotor (BLISK)

BLISK is fabricated as a single piece component by combining disk and blades. BLISKs are called integrated bladed rotors (IBR), meaning that blade roots and blade locating slots are no longer required. This eliminates the attachment concept of the blades to the disk), this leads to decreasing the number of components in the compressor, while at the same time decreasing drag and increasing efficiency of air compression in the engine. BLISK geometry shown in Figure 2

2. LITERATURE REVIEW

Mr. S. Karthik et al. [1] discussed effective uses of a Compressor blade are limited at its maximum operational junction frequency. The study was done on compressor blade by using different materials with different blade angle profile of 5.5, 8.5, 12.5 and 16.5. A natural frequency was analysed and harmonic analysis is performed for validation. In order to verify the present ANSYS model, the Natural frequency with their modes by using four types of materials are compared with the available experimental results present in the literature. The natural frequency of the compressor blade is compared by using three types of materials and is predicted that at 12.5, 16.5 blade angle profile a carbon fiber reinforced plastic gives better frequencies in different modes.

D. A. Kilpatricks et al. [2] detailed investigation of the stresses in the early stator blade rows of an axial-flow compressor. This study show that stall cell excitation is present in the first and second blade rows and causes blade resonances over a fairly wide part-load condition. Fundamental modes of first and second row stator blades resonances with the fourth and fifth stall cell harmonics give rise to stress peaks in the region of + 10 to + 20 tons/sq in.

Rygiel et al. [3] presented modal analyses, Modes are inherent properties of the structure, and are determined by the material properties (mass, damping, and stiffness), and boundary conditions of the structure. Each mode defined by a natural (modal or resonant) frequency, modal damping, and mode shape. If change in either material properties or the boundary conditions of the structure, its modes will change as well. In few cases with change in mass of structure vibration will change.

3. OBJECTIVES OF THE WORK

The main objective of this work is replacing the life cycle completed rotor blades in the operating engines with Blisk.

- Considered one of the stages for the existing rotor with blade created in Unigraphics.
- Simulate at various operating speeds for rotor with blade disk.
- To consider the natural frequencies for basic mode shapes at various speeds.
- To construct Campbell diagram and calculate the margin of safety.
- Calculating the weight reduced for created blisk geometry, Simulate the blisk at various speeds and construct the Campbell diagram to check the resonance on blisk.

4. DESIGN

The Geometry of Blade profile can be defined in terms of Blade maximum thickness, Position of the blade maximum thickness, Camber line, Position of the blade maximum camber, Thickness of the leading and trailing edges.Data points are provided for the cross section of the Airfoil. These data points considered from NACA 6412 and download from http://airfoiltools.com/airfoil/details?airfoil=naca6412-il. Data points are then connected by using the splines and then joined. The details of the blade creation were presented in step wise

Step 1: Data points are imported in to Unigraphics by keeping the working coordinate system as absolute coordinate system

Step 2: Create splines by connecting points to form a closed loop of curves

Step 3: Move closed section curves in vertical direction

Step 4: Create the airfoil suface by through curve mesh

Step 5: Create blade root cross section based on the radial slot of the disk.

Step 6: Revolve the cross section based on the no of blades in the rotor.

Step 7: Combine the both root and the airfoil, apply smooth blend between the intersection of root and airfoil as shown in Figure 3.

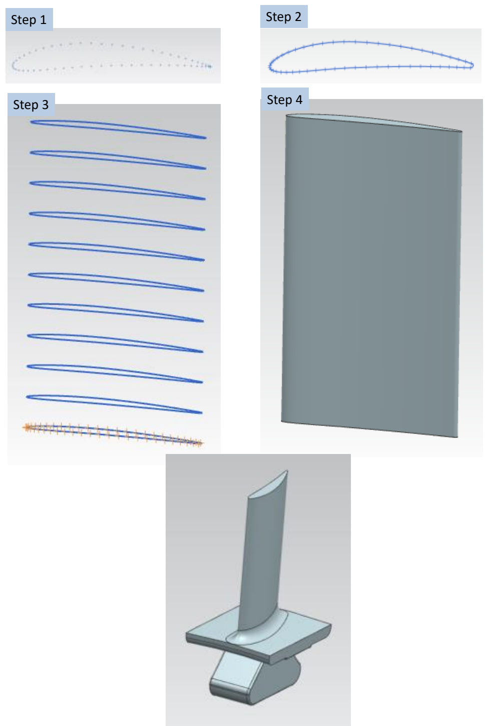

Fig 3: Airfoil Blade

Steps Followed for Rotor Blade Creation:

The details of the rotor blade creation were presented in step wise,
Step 1: Create cross section of the rotor as per the disk cross section.
Step 2: Revolve the disk cross section about z-axis 360°.
Step 3: Assemble the blade into disc slot at correct positon.
Step 4: Repeat the above step for remaining 44 blades, after assemble the blade rotor with disk as shown in Figure 4

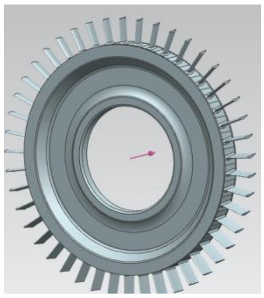

Fig 4: Rotor Blade Disk

Steps Followed for Blisk Creation:

The details of the rotor blade disk creation were presented in step wise,
Step 1: Create cross section of the rotor as similar to the rotor blade disk in such that reduction in area of cross section at blade assembly as shown in figure 4.3.
Step 2: Revolve the disk cross section about z-axis 360°.
Step 3: Assemble the blade on rotor at correct positon.
Step 4: Repeat the above step for remaining 44 blades, combine all the blades with rotor disk.

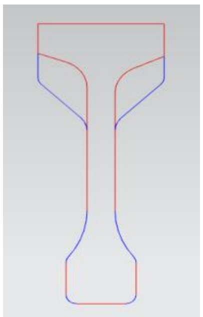

Fig 5: Comparison of Blisk vs Rotor Blade Disk Cross Section

Sector creation for cyclic symmetry analysis:

The structure which is repeated at equally spaced intervals about the axis, If the displacement boundary conditions of all segments are identical with respect to the axis, we can analyzed the structure in terms of the mass and stiffness characteristics of a one segment. This concept is called "cyclic symmetry". A sector represents a pattern that is repeated n times in a cylindrical coordinate system, it yield the complete structure. In the present Rotor Blade disk have 45 blades so create the 45 sectors about the rotation axis. The sector models for Rotor blade disk and Blisk shown in Figure 6

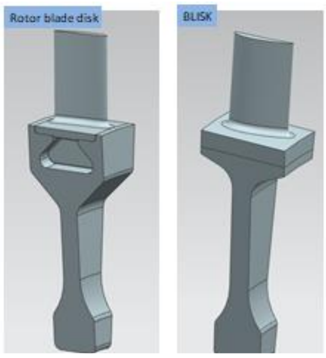

Fig 6: Cyclic Symmetry Section for Blisk and Rotor Blade Disk

5. ANALYSIS

Meshing is the process of discretizing the model in to finite no entities called elements which are connected to each other by nodes.

Boundary Conditions

In the pre-stressed analysis phase couplings are used for the model. Couplings are responsible for the behavior of the model as a 360 degree model. Couplings create constraint equations for the sector ends. Hence couplings doesn't allow for the tangential deformation of the model. But in the Modal analysis phase command "CYCLIC" is used. This option triggers the formation of the constraint equations on the sector ends. Cyclic option allows specifying the harmonic index. Harmonic index allows us to specify the model to run for a particular nodal diameter which is the main cause for the resonance. To prevent the rigid body motion of the component, a line of nodes on the inner diameter surface of the rotor wheel is constrained in all degrees of freedom as shown in Figure 7. Even though the present analysis focuses on the blade, Rotor wheel should be constrained on the inner diameter and not on the outer diameter so that the nodal diameters can be obtained.

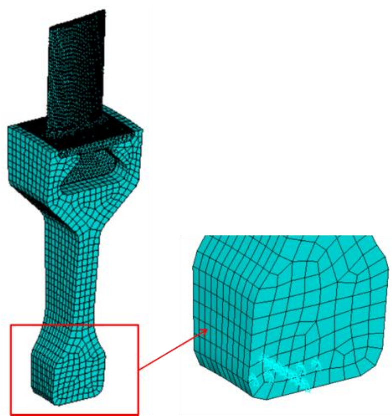

Fig 7: Boundary Condition on the Sector Model

Pressure load:

Pressure applied on the cross section downstream face and upstream face as shown in below Figure. The pressure applied for rotor with blade disk and Blisk are same, it varies from 0.07 to 0.86 N/mm^2. In Figure 8 shows the pressure contour on the blisk.

Temperature load:

Temperature applied on the body of the components. Temperature on the body elements increases along axis of the engine and vertical direction. The temperature applied for rotor with blade disk and Blisk are same, it varies from 38° to 115°c. In Figure 8 shows the temperature contour on the rotor with blade disk.

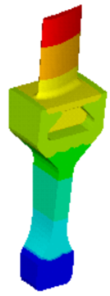

Fig 8: Pressure on the Sector Model

Fig 9: Temperature on the Sector Model

Pre Stress Modal Analysis

Pre-stress Modal Analysis is performed on the model to evaluate the natural frequency of the model under the applied loading conditions. General Modal analysis doesn't include the softening due to the applied loads. Hence to include it pre stress analysis is performed. Macro developed to perform the pre stressed modal analysis.

6. RESULTS AND DISCUSSIONS

The mode shapes bending and torsion are considered in present work as shown in Figure 6.1 and 6.2. External excitations are mainly due to the disturbances, upstream and downstream rotor and stator counts. In this analysis the engine orders are designated as IE, 2E, 3E, 22E, 44E and 46E. IE, 2E and 3E as the basic disturbances due to difference in the pressure, 22E as half of the blade count, 44E as downstream stator counts and 46E as upstream stator counts. The external excitations frequency for Eth engine order at speed N is presented in table 1

Table 1: External Frequency at various speeds

Sr No	Speed in RPM	Frequencies in Hz					
		1E	2E	3E	22E	44E	46E
1	0	0	0	0	0	0	0
2	1000	16.67	33.33	50	366.67	733.33	766.67
3	3000	50	100	150	1100	2200	2300
4	5000	83.33	166.67	250	1833.33	3666.67	3833.33
5	9000	150	300	450	3300	6600	6900

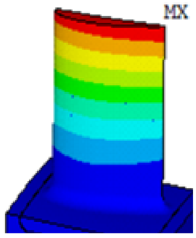

Fig 10: Bending Mode Shape

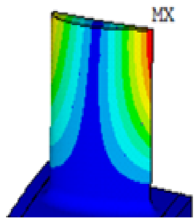

Fig 11: Torsion Mode Shape

The natural frequency for bending and torsional mode shapes with nodal diameter 0, 1, 2 and 3 at various speeds for rotor blade disk and blisk are presented in Table 2 and 3

Table 2: Natural Frequencies for rotor blade disk

Sr No	Speed in RPM	Nodal diameter 0		Nodal diameter 1		Nodal diameter 2		Nodal diameter 3	
		Bending Frequency in Hz	Torsion Frequency in Hz	Bending Frequency in Hz	Torsion Frequency in Hz	Bending Frequency in Hz	Torsion Frequency in Hz	Bending Frequency in Hz	Torsion Frequency in Hz
1	1000	1565.33	5617.27	1579.38	5621.57	1584.54	5635.84	1590.18	5680.91
2	3000	1567.12	5617.76	1581.41	5623.38	1586.35	5637.25	1592.62	5681.82
3	5000	1579	5621	1594.85	5625.65	1599.64	5639.38	1605.85	5685.31
4	9000	1605	5629	1620.38	5632.47	1626.21	5647.48	1631.36	5694.47

Table 3: Natural Frequencies for blisk

Sr No	Speed in RPM	Nodal diameter 0		Nodal diameter 1		Nodal diameter 2		Nodal diameter 3	
		Bending Frequency in Hz	Torsion Frequency in Hz	Bending Frequency in Hz	Torsion Frequency in Hz	Bending Frequency in Hz	Torsion Frequency in Hz	Bending Frequency in Hz	Torsion Frequency in Hz
1	1000	1644.72	5733.29	1660.36	5734.87	1666.59	5738.99	1680.93	5758.95
2	3000	1644.45	5733.23	1660.1	5734.24	1666.35	5738.92	1680.71	5758.86
3	5000	1642.74	5732.71	1658.32	5734.28	1664.61	5738.48	1679.38	5758.41
4	9000	1639.5	5731.85	1655.16	5733.32	1661.46	5737.51	1676.35	5757.55

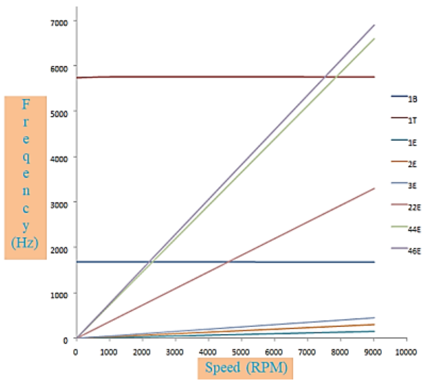

Fig 12: Campbell diagram

The interference margins are calculated for bending and torsion mode shape at various speeds with different engine orders. The interference margins for rotor blade disk and blisk were presented in Table 4 and 5

Table 4: Interference Margin for rotor blade disk

Sr No	Engine Order	Bending Interface Margin in Hz at Various Speeds in RPM				Torsion Interface Margin in Hz at Various Speeds in RPM			
		1000	3000	5000	9000	1000	3000	5000	9000
		Nodal diameter 0				Nodal diameter 0			
1	1st	1548.7	1517.1	1495.7	1455	5600.6	5567.8	5537.7	5479
2	2nd	1532	1467.1	1412.3	1305	5583.9	5517.8	5454.3	5329
3	3rd	1515.3	1417.1	1329	1155	5567.3	5467.8	5371	5179
4	22nd	1198.7	467.1	-254.3	-1695	5250.6	4517.8	3787.7	2329
5	44th	832	-632.9	-2088	-4995	4883.9	3417.8	1954.3	-971
6	46th	798.7	-732.9	-2254	-5295	4850.6	3317.8	1787.7	-1271
		Nodal diameter 1				Nodal diameter 1			
1	1st	1562.7	1531.4	1511.5	1470.4	5604.9	5573.4	5542.3	5482
2	2nd	1546	1481.4	1428.2	1320.4	5588.2	5523.4	5459	5332
3	3rd	1529.4	1431.4	1344.9	1170.4	5571.6	5473.4	5375.7	5182
4	22nd	1212.7	481.4	-238.5	-1680	5254.9	4523.4	3792.3	2332
5	44th	846	-618.6	-2072	-4980	4888.2	3423.4	1959	-967.5
6	46th	812.7	-718.6	-2239	-5280	4854.9	3323.4	1792.3	-1268
		Nodal diameter 2				Nodal diameter 2			
1	1st	1567.9	1536.4	1516.3	1476.2	5619.2	5587.3	5556.1	5497
2	2nd	1551.2	1486.4	1433	1326.2	5602.5	5537.3	5472.7	5347
3	3rd	1534.5	1436.4	1349.6	1176.2	5585.8	5487.3	5389.4	5197
4	22nd	1217.9	486.35	-233.7	-1674	5269.2	4537.3	3806.1	2347
5	44th	851.21	-613.7	-2067	-4974	4902.5	3437.3	1972.7	-952.5
6	46th	817.87	-713.7	-2234	-5274	4869.2	3337.3	1806.1	-1253
		Nodal diameter 3				Nodal diameter 3			
1	1st	1573.5	1542.6	1522.5	1481.4	5664.2	5631.8	5602	5544
2	2nd	1556.9	1492.6	1439.2	1331.4	5647.6	5581.8	5518.6	5394
3	3rd	1540.2	1442.6	1355.9	1181.4	5630.9	5531.8	5435.3	5244
4	22nd	1223.5	492.62	-227.5	-1669	5314.2	4581.8	3852	2394
5	44th	856.85	-607.4	-2061	-4969	4947.6	3481.8	2018.6	-905.5
6	46th	823.51	-707.4	-2227	-5269	4914.2	3381.8	1852	-1206

Table 5: Interference Margin for blisk

Sr No	Engine Order	Bending Interface Margin in				Torsion Interface Margin in Hz			
		1000	3000	5000	9000	1000	3000	5000	9000
		Nodal diameter 0				Nodal diameter 0			
1	1st	1628.1	1594.5	1559.4	1489.5	5716.6	5683.2	5649.4	5582
2	2nd	1611.4	1544.5	1476.1	1339.5	5700	5633.2	5566	5432
3	3rd	1594.7	1494.5	1392.7	1189.5	5683.3	5583.2	5482.7	5282
4	22nd	1278.1	544.45	-190.6	-1661	5366.6	4633.2	3899.4	2432
5	44th	911.39	-555.6	-2024	-4961	5000	3533.2	2066	-868.2
6	46th	878.05	-655.6	-2191	-5261	4966.6	3433.2	1899.4	-1168
		Nodal diameter 1				Nodal diameter 1			
1	1st	1643.7	1610.1	1575	1505.2	5718.2	5684.2	5651	5583
2	2nd	1627	1560.1	1491.7	1355.2	5701.5	5634.2	5567.6	5433
3	3rd	1610.4	1510.1	1408.3	1205.2	5684.9	5584.2	5484.3	5283
4	22nd	1293.7	560.1	-175	-1645	5368.2	4634.2	3901	2433
5	44th	927.03	-539.9	-2008	-4945	5001.5	3534.2	2067.6	-866.7
6	46th	893.69	-639.9	-2175	-5245	4968.2	3434.2	1901	-1167
		Nodal diameter 2				Nodal diameter 2			
1	1st	1649.9	1616.4	1581.3	1511.5	5722.3	5688.9	5655.2	5588
2	2nd	1633.3	1566.4	1497.9	1361.5	5705.7	5638.9	5571.8	5438
3	3rd	1616.6	1516.4	1414.6	1211.5	5689	5588.9	5488.5	5288
4	22nd	1299.9	566.35	-168.7	-1639	5372.3	4638.9	3905.2	2438
5	44th	933.26	-533.7	-2002	-4939	5005.7	3538.9	2071.8	-862.5
6	46th	899.92	-633.7	-2169	-5239	4972.3	3438.9	1905.2	-1162
		Nodal diameter 3				Nodal diameter 3			
1	1st	1664.3	1630.7	1596.1	1526.4	5742.3	5708.9	5675.1	5608
2	2nd	1647.6	1580.7	1512.7	1376.4	5725.6	5658.9	5591.7	5458
3	3rd	1630.9	1530.7	1429.4	1226.4	5709	5608.9	5508.4	5308
4	22nd	1314.3	580.71	-154	-1624	5392.3	4658.9	3925.1	2458
5	44th	947.6	-519.3	-1987	-4924	5025.6	3558.9	2091.7	-842.5
6	46th	914.26	-619.3	-2154	-5224	4992.3	3458.9	1925.1	-1142

Natural frequency and excitation frequency values are plotted on Campbell diagram as shown in Figure 12, The evolutions of the natural frequencies corresponding to modes are drawn in frequencies of the rotation speed of the shaft. The necessary condition for an engine order (EO) excitations to excite a blade disk is treat the EO frequency coincides with the natural frequency of the structure. This lead to detect the possible resonate point.

7. CONCLUSIONS

The design and modal analysis of the compressor rotor blade disk and Blisk are carried out. It is found that this rotor dynamic effect causes changes in natural frequencies, critical speeds and vibration response of the structure. Created rotor blade disk and Blisk in such that weight of the Blisk is reduced by 13.51% compared to the rotor blade disk. The natural frequencies for basic mode shapes obtained for various speeds are identified, the rotor blade disk frequency increase with increase the speed of the rotor and Blisk frequency decreases with increasing the rotor speed. Campbell diagram is constructed with natural frequency and excitation frequency and observed the failure points

- The least interference value of the rotor blade disk is identified as -227.48 at 22 EO and speed 5000 rpm for bending mode shape and -905.33 at 44 EO and speed 9000 rpm for torsion mode shape.
- The least interference value of the Blisk is identified as -153.95 at 22 EO and speed 5000 rpm for bending mode shape and -842.45 at 44 EO and speed 9000 rpm for torsion mode shape.

The Campbell diagram shows that the critical interface margin values are not close to zero at engine operating speeds. Natural frequency and excitation frequency are not intersecting at engine operating speeds, therefore no chance of resonance at operating speeds. Reduced weight Blisk is replacing the life cycle completed Rotor blade disk. By the introducing Blisk the weight of the component reduced to 13.51% of the rotor blade disk. This reduction of weight leads to improve the performance of the compressor and life of the component.

8. REFERENCES

1. Mr.S.Karthik, Mr.R.Anand, Mr.Adel Abbas, Mr.T.Kathiravan, Mr.K.Parthiban “Optimization And Vibration Analysis Of Compressor Blade With Different Blade Angle” Volume 5, 2017.
2. D.A.Kilpatrick, A.D.S.Carter And L.O'neill “Blade Vibratory Stresses In A Multi-Stage Axial-Flow Compressor” Volume 6 Issue Iv, April 2014.

3. Paweł Rygiel, Wojciech Obrocki, Jan Sieniawski "Numerical Vibration Analysis Of Turbine Engine Compressor Blades Depending On Geometry And Position Of The Damage" Vol. 41, 2017.
4. Merle C. Huppert, Howard F. Calvert and Andre J. Meyer "Experimental Investigation Of Rotating Stall And Blade Vibration In The Axial-Flow Compressor Of A Turbojet Engine" National Advisory Committee For Aeronautics, 2008
5. Dr.Ir.I. Lopez , W. Rooyakkers, Dr.Ir.R.Vijgen "Experimental Modal Analysis Of A Turbine Blade" Technische Universiteit Eindhoven Department Mechanical Engineering Dynamics And Control Group Eindhoven, 2004.
6. Lucjan Witek, "Experimental Modal Analysis Of The Turbine Blade" Rutmech, 2016.
7. Miroslaw Witos "On The Modal Analysis Of A Cracking Compressor Blade" Research Works Of Air Force Institute Of Technology, 2008.
8. Masataka M., Yasutomo K., Thom K., Katsuhisa F., Yoshiki K., and Seigo I, "Root and Groove Contact Analysis for Steam Turbine Blades, Japan Society of Mechanical Engineers International Journal", 1992, Solid Mechanics and Strength of Materials, v.35, pp.508 - 5 14, 2014.
9. S. S. Manson, A.J. Meyer, Jr, H. F. Calvert and M.P. Hanson "Factors Affecting Vibration Of Axial-Flow Compressor Blades" National Advisory Committee For Aeronautics, Cleveland, Ohio, 2014.