Cyclic life establishment of the compressor disc of a gas turbine engine

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ABSTRACT

Gas turbine engines for fighter class aircrafts are having unique characteristics. It is not uncommon that gas turbine engine with deep throttles both during acceleration and deceleration not only consumes life but also deficiencies are manifested out in the form of defects. In this project, the dictum is to carry out a cyclic life analysis of a compressor disc which is already flying along with the fighter class engine fitted on a fighter class aircraft. Since both the geometrical drawing and the boundary conditions of the discs are known, a model is therefore prepared using the modeling software (CATIA). The discretized model is then subjected to analysis using ANSYS software and the maximum stress values and its locations were identified for all the RPM conditions of the engine. Alternating stresses are acquired from Goodman diagram and cyclic life is obtained from S-N diagram. Modal analysis is carried out for bending and umbrella mode and the values are superposed on Campbell diagram, to arrive at the resonances with the engine order. Efforts are also made to simulate defect in the component and the modes are captured to identify the resonance characteristics of the component with defects.

1.INTRODUCTION

Fighter class aircraft gas turbine engines are supposed to have very high reliability and no failure of components are permitted despite the fact that the engines undergo deep throttling quite often. The design aspects thereforeshould be reasonably fool proof so that no component failure occurs at any point of time during its ear marked cyclic life. This has developed a curiosity to understand the design efficacy of compressor disc. The project is on a first stage low pressure compressor disc of a flying engine which has almost done its life flawlessly.

The disc was modeled and analyzed for various types of stresses that are encountered by the disc during its operation. The mean stress that is obtained for various RPM's of the engine is superposed on the Goodman and the S-N curve,which has given indirectly the alternating stress and the equivalent cyclic life.Since the design aspects are to be learnt the modal analysis for this component is also carried out in order to identify the mechanical vibrations and its resonance in case these frequencies intersect on the engine orders in the sensible RPM range of the engine. Since the tool used for analysis is quite a powerful one, defects are simulated by creating notches at the root radius of the flanks, notches on the spline roots, etc.

2. GAS TURBINE ENIGNE

Turbojet engine consist of an air inlet, [1] an aircompressor, a combustion chamber, turbine and a nozzle. The air is compressed by the compressor and the air is heated in the combustion chamber by burning the fuel, and expanded by the turbine unit. Further, through the exhaust nozzle, the air is further expanded and results into production of thrust.

3. COMPRESSOR DISC

A model that is generated for the first stage low pressure compressor disc has to be now analyzed for its stresses and strains as applicable to the component [3]. The disc is fabricated out of martensite stainless steel which can resist most of the centrifugal and bending loads emanating from the blades located at the dovetail slots of the circumference of the disc.

As it is known, the blades are also fabricated out of martensite stainless steel, enormous centrifugal loads is lightly to appear on the slots of the disc.The compressor disc is rotated by the torque that is transferredfrom the turbine module through a hollow shaft, which is also fabricated out of martensite stainless steel.

The torque transfer to the disc is transfer to the disc is through 26 splines cut on the shank of the disc. The disc being a heavy component, saddle mounting is resorted to avoid overhang which can allow the tip of the blades to touch the casing during its maneuver.

Therefore there is a bearing location close to the hub of the disc for resisting an abnormal motion during manoeuver of the aircraft.

4. ANALYSIS OF THE COMPRESSOR DISC

While modeling and solving for the disc stresses, it is necessary to simulate the boundary conditions as existing in the engine. As far as the splines are concerned, both working and non-working sides of the splines are arrested from deformation in the running side of the splines. Secondly the bearing location is also not permitted from deformation due to seating of the inner race of the ball bearing. In order to reduce the time taken for the analysis of vibratory stresses (Modal analysis) for umbrella mode and other crack simulation aspects[2], the rotational symmetry concept is utilized by which only five dovetail slots along with disc diaphragm is used for analysis purpose. In such cases either sides of the portion is arrested from deformation. This is true when the disc if it were considered as a full disc, then these locations is automatically arrested.

Loading the disc is the next effort to simulate the blades located in its slots. As it is known, when the rpm is increasing from the start of the engine, due to its own weight, the blades levitate in its slot and match with the imaginary lines of both disc slot and the blade flange on either sides. Normally it happens from the start of the engine reaching around 400 rpm beyond which the orientation is fixed with minor movements of the order of microns. Although transition elements have to be utilized as an interface between disc and the blade due to some constraints this could not be executed. Alternatively the flanges of thediscare fully arrestedfrom its deformation towards the blade side. In order to load these flanges, the weight of the first stage compressor rotor blade is distributed over the internal flanges of the dovetail slots. The same loading is repeated on all the slots as if the disc is rotated along with the blade in a realistic situation.

The model along with boundary condition, loading condition, material property like young's modulus, density, Poisson ratio are used for the analysis of the disc. Since more or less the assembly and the boundary conditions are akin to actual cases, the stresses that are obtained also need to be near actuals.

Initially the disc is rotated for 100% rpm (10,000 mechanical rpm) and the stresses are found to be nearly equal to 20 kg/mm^2 (200 Mpa) at the relief radius of the dovetail slots. This exercise is repeated for all other rpm conditions where the engine produces sensible thrust.The stresses that are calculated are pertaining to Von-mises stresses, so that the maximum stress encountered indirectly gives the cyclic life values.Since the material property is known for martensite stainless steel namely yield point and ultimate tensile stress, Good man diagram is evolved and the hypotenuse is the line joining $\sigma_y and \sigma_{ut}$.The mean value is obtained in terms of von mises stresses, is plotted and intersected to get the alternating stresses which is on the Y axis. Further from the atlas of martensite stainless steel material characteristics is available in terms of alternating stresses Vs number of equivalent LCF cycles.It could be therefore possible to identify the equivalent cyclic life in terms of zero-max-zero. The disc is then subjected to vibratory modes for various rpm conditions and the modal values for bending mode are extracted. As a follow-on exercise, since the disc encounter umbrella modes in the vibrational analysis attempt is made to extract its sub harmonics.

Campbell diagram is drawn for rpm of the engine Vs frequency of the component and therefore this diagram is utilized for extracting resonance conditions of the frequencies and the engine orders. Few points are found located, which show distress signals at the engine cross over frequencies in respect of the component design.

First stage compressor is fabricated out of MartensiteStainless steel in order to have a durable cyclic life. As far as the material is concerned both yield point and ultimate tensile stress point are known for the material obtained from literature having minus three sigma values.

The disc is modeled through CATIA and analyzed by ANSYS for its maximum steady state stress. It is identified that the maximum Von Mises stresses are found appearing at transition radius of the dovetail slot due to the blade centrifugally pulling out the flanks of the slot and therefore very high stresses are created in the recess.As far as disc is concerned the failure mode is low cycle fatigue and therefore the alternating stresses have to be more with the mean stresses at a lower order.

4.1 GOODMAN DIAGRAM

Knowing the yield stress and ultimate tensile stress, Goodman diagram can be drawn which bifurcates the serviceability and unservicebility of the component under consideration[4]. The steady-state stress for 100% rpm of the engine is found to be around 21.0 kg/mm^{2}and are shown in Fig.6.1&6.2. The same exercise is repeated for all other RPM conditions and the steady state values for various rpm conditions are placed in Table 4.1.

Table 4.1 Mean stress and Alternating Stress value for various RPM(Revolution Per Minute)

S.NO	RPM	MAXIMUM STRESS kg/mm^2	ALTERNATING STRESS kg/mm^2
1.	10000	21.0	68
2.	9500	17.5	69
3.	9000	15.7	70
4.	8500	13.5	75

4.2 S-N CURVE

Knowing the mean and alternating stresses for the disc at various RPM conditions, the curve meant for establishing alternating stress Vs number of zero-max-zero cycle is obtained from the Atlas of the materials [4]. As for as Martensite stainless steel is concerned the S-N curve could be obtained from the open literature and the same is used for extracting the cyclic life of the component. Thus it is possible to attain the cyclic life Vs the RPM of the engine is placed at Table 4.2.

Table 4.2. Establishment of cyclic life of the disc

S.NO	RPM	MAXIMUM STRESS kg/mm^2	ALTERNATING STRESS kg/mm^2	CYCLIC LIFE OF THE DISC(ZERO-MAX-ZERO)
1.	10000	21.0	68	1.6×10^6
2.	9500	17.5	69	6×10^5
3.	9000	15.7	70	2×10^5
4.	8500	13.5	75	1.9×10^5

4.3 CAMPBELL DIAGRAM

It is customary that various structural components are to be checked for its resonance conditions both for its harmonics and sub harmonics.

The reason behind carrying out such an exercise at the initial stages of the engine design is thatcostlier modification may have to be carried out if failure is noticed during the operation of the engine. Not only the cost of modification would be more, this also brings in large delay in the execution of the program.

4.3.1 BENDING MODE

Each component is checked for its structural frequencies depending upon the type. For example as far as the disc is concerned the modal frequencies are emanating from the bending modes whose frequencies lie from 703.04 to 2633.7 Hz.

It could be seen that sub harmonics approximately five modals are extracted and the same is placed on the Table 4.3.

Table 4.3 Bending mode frequencies for various RPM

S.NO	RPM	BENDING MODE FREQUENCIES (Hz)
1.	10000	703.04 763.57 769.69 2044.3 2633.7
2.	9500	701.10 762.04 768.17 2044.10 2635.00
3.	9000	699.26 760.58 766.72 2043.80 2633.60
4.	8500	697.52 759.20 765.35 2043.60 2633.60

4.3.2 UMBRELLA MODE

The disc is also subjected to Umbrella mode because the hub is fitted to the shaft and the blades are located on the periphery of the disc. The Umbrella modes along with its sub harmonics are identified in the analysis and those values are placed in Table 4.4.

Table 4.4 Umbrella Mode Frequencies for Various RPM

S.NO	RPM	UMBRELLA MODE FREQUENCIES (Hz)
1.	10000	5605.40 8141.10 9972.10 10980.00 12125.00
2.	9500	5605.00 8140.40 9971.80 10979.00 12125.00
3.	9000	5604.60 8139.80 9971.50 10978.00 12125.00
4.	8500	5604.20 8139.30 9971.20 10977.00 12124.00

4.3.4 CAMPBELL DIAGRAM FOR CROSSOVER FREQUENCY

Campbell diagram is the one conventionally being drawn for all the structural components of the engine. On the x- axis of the Campbell diagram is marked sensible RPM of the engine and on the y-axis is the frequencies of the component namely, low pressure compressor disc. The frequencies are marked on y-axis encompasses bending mode and its harmonics, umbrella mode and its harmonics etc. As far as the Campbell diagram is concerned, primarily various engine orders are marked originating from the origin of the graph. It is to be noted that lower engine orders are close to x-axis while the higher engine orders are located well away from the x-axis. Superposed are the harmonics of the bending mode and umbrella mode.Within the operating regiments the nodes are extracted where the engine orders are crossing the bending mode harmonics and umbrella mode harmonics. It is important that lower order harmonics of bending mode and umbrella modes are to be addressed for any modification to overcome the intersections. At the intersection points of the modes and the engine orders are nothing but the frequencies of mechanical resonances. This means to say that if it all any vibration occurs on the engine, the mechanical vibrations are attributed to the frequencies of either Bending or Umbrella modals of the disc.

Table 4.5 Modal Analysis for Compressor Disc (Bending Mode)

S.NO	MODE	RPM	ENGINE ORDER	CROSSOVER FREQUENCIES FOR BENDING MODE (Hz)
1.	1	10400	4	704
2.	1	8200	5	703.9
3.	2	8700	5	760
4.	3	9200	5	766
5.	3	7600	6	765

It is also possible to identify the vibration due to aerodynamics of the flow on the air washed component. Normally this vibration is normal to the intersection point of the frequencies of mechanical components and the engine orders. The aerodynamics vibration although cannot be isolated from the overall vibration arises due to members of component or its fraction.

Thus the Campbell diagram which is drawn for the first stage compressor disc and the Modal analysis for compressor disc are tabulated and placed in the Table 4.5& 4.6

Table 4.6 Modal Analysis for Compressor Disc (Umbrella Mode)

S.NO	MODE	RPM	ENGINE ORDER	CROSSOVER FREQUENCIES FOR UMBRELLA MODE (Hz)
1.	1	10900	31	5605.8
2.	1	10800	32	5605.7
3.	1	10600	33	5605.6
4.	1	10300	34	5605.5
5.	1	9900	35	5605.4
6.	1	9500	36	5605.3
7.	1	9400	37	5605.2
8.	1	9200	38	5605.1
9.	1	9000	39	5604.9
10.	1	8700	40	5604.8
11.	1	8500	41	5604.7
12.	1	8300	42	5604.6
13.	1	8000	43	5604.5
14.	1	7900	44	5604.4
15.	1	7700	45	5604.3

5. SIMULATION OF DEFECT

Based on the experience of crack formations and repair during operation and overhaul of the blade, a simulation exercise is contemplated using the powerful software namely ANSYS.

Therefore a notch is created at the leading edge flank root radius with an included angle of 30 degrees and to a depth of 5mm. As usual the material property, loading condition and boundary conditions are imposed on the model to evaluate the stress at various locations of the disc in addition to the apex of the crack. In the initial attempts although stress are shown at the root radius of the flank, absence of high stress at the apex of the crack is found to be surprising. An attempt to modify the aspect ratio of elements (fine most) has yielded the result as expected from the theory of fatigue and fracture. The stress at apex at various rpm conditions are calculated and found to be higher the rpm of disc, higher the Von-mises stress at the crack are shown in Fig. 6.6. The mean stress for the crack is shown in the Table 5.1. This is a good finding and proven by the fact of Paris Law dealing with crack, fracture toughness and number of cycles.

Table 5.1 cumulative damage- maximum stress

S.NO	RPM	CRACK MAXIMUM STRESS (kg/mm^2)
1.	10000	434.73
2.	9500	419.5
3.	9000	406.5
4.	8500	279.54

It is not adequate to identify a mean stress alone because it is necessary to identify the alternating stress and the residual life. As customary Goodman diagram is taken for analysing the alternating stress but this diagram could not give the desired result because this diagram is limited to elastic property of the material and to a limited extend, the plastic properties. But the stresses found to be at the apex of the notch comparatively is very high and could not get accommodated within the Goodman diagram [6]. So the attempt to identify the balance life is aborted.

	3	7600	6	765

Barring aside the above, an attempt is made to extract the frequency namely the bending modes and its harmonics for identifying the crossover frequencies of the disc with the engine orders. It is observed that the disc with the crack is having lesser bending frequency Table 5.2 – 5.5 and its harmonics than the serviceable disc.

Table 5.2 Bending Mode Frequencies for Various RPM

S.NO	RPM	BENDING MODE FREQUENCIES (Hz)

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1.	10000	509.04 763.57 769.69 2044.3 2633.7
2.	9500	508.1 762.04 768.17 2044.1 2633.7
3.	9000	507.8 760.58 766.72 2043.8 2633.6
4.	8500	507.5 759.2 765.35 2043.6 2633.6

Table 5.3 Umbrella Mode Frequencies for Various RPM

S.NO	RPM	UMBRELLA MODE FREQUENCIES (Hz)
1.	10000	5031.2 8208.6 10108 11135 12152
2.	9500	5030.6 8208.2 10107 11133 12151
3.	9000	5028.6 8208 10105 11131.6 12150
4.	8500	5028.1 8207.6 10103 11130 12148

Table 5.4 Modal Analysis for Compressor Disc with crack (Bending Mode)

S.NO	MODE	RPM	ENGINE ORDER	CROSSOVER FREQUENCIES FOR BENDING MODE (Hz)
1.	4	8400	15	2043.2

Table 5.5 Modal Analysis for Compressor Disc with crack (umbrella Mode)

S.NO	MODE	RPM	ENGINE ORDER	CROSSOVER FREQUENCIES FOR UMBRELLA MODE (Hz)
1.	1	9700	31	5030.8
2.	1	9400	32	5030.4

3.	1	8900	34	5028.2
4.	1	8400	36	5027.9
5.	1	8200	37	5027.7
6.	1	8000	38	5027.2

6. RESULTS AND DISCUSSION

In this project work an attempt is made to understand the design aspects of the first stage compressor disc which would give data for an abinitio design. Initially therefore the disc that is available in the laboratory is used for identifying various dimensions which would help to draft the model.

The model is drafted in CATIA without losing any critical dimension of the disc. In order to set the material property for the purpose of analysis the library of material is used for the purpose to extract young's modulus-2 X 10^5Mpa, Poissonratio-0.322, yield strength-780Mpa, ultimate strength-1230Mpa.

As per the boundary conditions are concern, since the torque transfer to the disc is through the splines all the deformations are arrested for the spline located at the rear side of the shaft . Since the disc is saddle mounted, the bearing location is identified and the deformations in all the direction is arrested.

As far as the loading is concerned the disc is rotating at 10000rpm (for max rating) and the blades that are located on the 21 slots are given the distribution of the weight of the blade uniformly on both the flanks Fig. 6.1. Based on the material property of the disc, it is possible to know the yield stress and ultimate tensile stress and it has paved the way to generate the Goodman diagram which can be used for identifying the alternating stress.

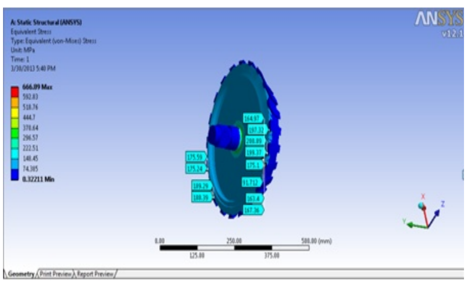

Fig. 6.1 Static Structure Analysis for 10000 rpm

Similarly from the open literature, it is possible to extract S-N curve which can be used concurrently to identify the cyclic life of the component knowing the alternate stress. The obtained value for various rpm condition of the disc in respect of mean stress, alternating stress and cyclic life of the component is placed at Fig. 6.2.

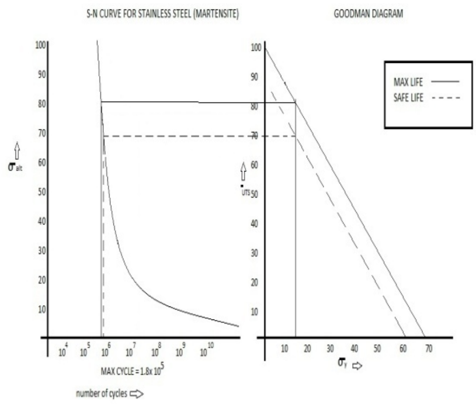

Fig. 6.2 S-N curve and Goodman diagram for 10000 rpm

It could be seen from the table that as the mean stress is increasing, the alternating stress decreases because as per the design intent the mean stress and alternating stress are complimentary to each other, so the table satisfy the design technical requirements.

The von-mises stresses that are obtained for various rpm conditions namely 10000, 9500, 9000 and 8500.and the values so obtained are placed in Table.4.1& 4.2. It is not adequate to identify only the mean stress but it is necessary to capture the alternating stress also. Its pertinent to note that the alternating stresses are also evaluate in the same location as that of mean stress which is located at the leading edge of the slot recess, for various rpm conditions.Modal analysis for various rpm conditions are also carried out.

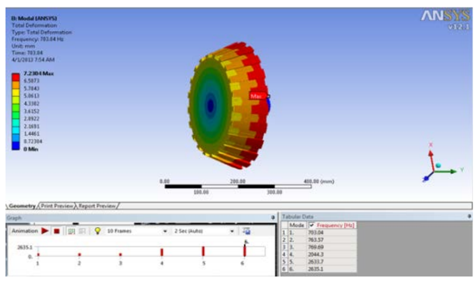

Fig. 6.3 Bending mode frequency for 10000 rpm

In modal analysis, the model is considered due to cyclic symmetry option used the 5 slot disc derives values of 21slot disc.

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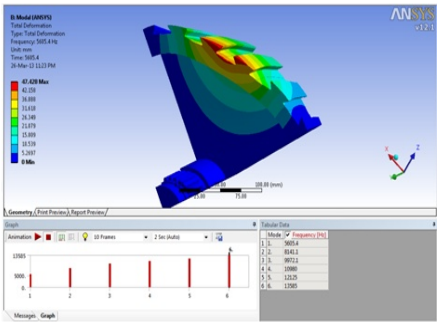

Fig. 6.4Umbrella mode frequency for 10000 rpm

Both frequencies are obtained for bending of the disc due to air loads and umbrella modes obtained from the forces acting because of the blade and the axialforces experienced by the compressor. The tabulated values for the bending frequency and umbrella mode frequency are placed in Table 4.3 - 4.6.

It could be seen from the values that the frequencies obtained for various bending mode and umbrella mode are more or less the same and these are independent of the rpm of the disc and the figures are placed in Fig. 6.3 & 6.4.Campbell diagram for various modes have been drawn to find out the cross over frequency.

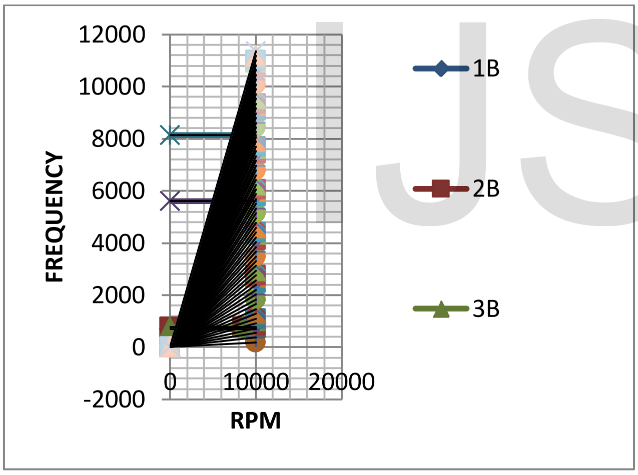

Fig. 6.5 Campbell diagram for low pressure compressor disc

Based on experience many types of cracks more identified at the root relief of the flanks and an attempt is made to extract the damages encountered by the disc and the residual life available after the crack initiation and propagation. It is necessary that a crack while it is generated, it absorbs low cycle fatigue energy and for the purpose of crack propagation since reduced stresses are required, this generally is accomplished in the HCF mode.Based on this understanding a crack is simulated at the leading edge of the slot with an included angle of 30 degree and for depth of 5mm. As usual analysis is carried out using ANSYS and the stress values are obtained at the apex of the crack and the values are taken from the Fig.6.6.

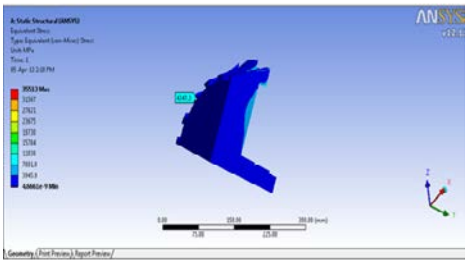

Fig. 6.6 Static structure analysis for failure component (10000 rpm)

Unless and otherwise the elements are very very fine, it is near impossible to get the stress values at the apex of the notch and that is the experience we got during our analysis.

Therefore as done earlier similar approach is carried out to arrive at the stress values at the apex of the notch and for varying rpm conditions, the results obtained is placed at Table 5.1. It is heartening to note that the von-mises stresses obtained at the apex of the notch has reduced as per the rpm of the disc.

This is further proved using the Paris law and stress intensity factor. Besides the disc as usual is checked for vibratory stresses due to bending and umbrella mode frequencies to check the feasibilities of obtaining the engine crossover rpm shown in Fig. 6.7 & 6.8.

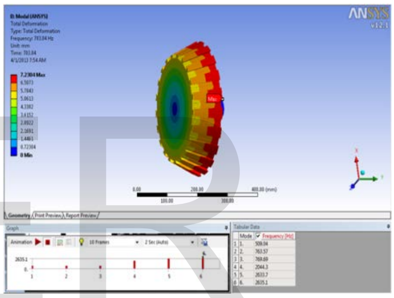

Fig. 6.7 Modal analysis for bending mode (10000 rpm)

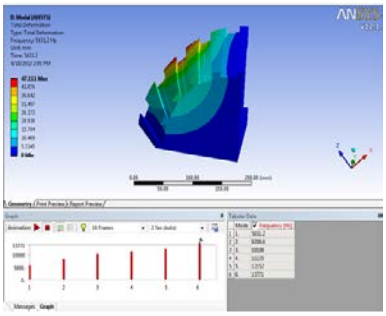

Fig. 6.8 Modal analysis for umbrella mode (10000 rpm)

Campbell diagram which is the one to bring out the rpm versus frequency of the engine and the component respectively brought out the engine crossovers for both serviceable disc and the disc with the flaw. Both these drawings are placed at Fig. 6.5 & 6.9. The frequency value obtained for varies rpm condition for both bending and umbrella modes are placed in Table 5.2 – 5.5.

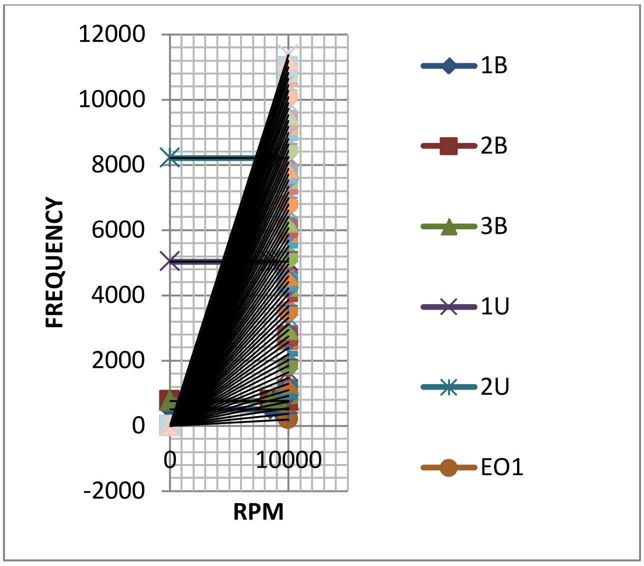

Fig. 6.9 Campbell diagram for low pressure compressor disc with notch

It could be seen from all the above that lot many lessons could be learnt from the above exercises both for steady and vibratory condition of the disc and reasonable inputs were obtained due to these exercises.

7. CONCLUSION

This project"cyclic life assessment of low pressure compressor disc as given large inputs regarding the design efficacies of a fighter class engine compressor disc"has given the strategy to be followed for assessing the disc in terms of its life through FEM. It is also given exposure towards modeling and analysis in CATIA and ANSYS respectively.

Among the all works done following are the significant outputs obtained while carrying out the project:

1. Analysis of the disc for its mean stress values for RPM conditions of the module. Location of mean stress values on the disc.
2. Evaluation of alternative stresses for the given mean stress value based on material property. Besides strain obtained due to alternating stresses and its location.
3. Assessment of modal frequencies for both bending and umbrella mode.
4. Identification of crossover frequencies of the bending modes against the engine orders.
5. Identification of crossover frequencies of the umbrella mode against the engine orders.
6. Assessment of steady state stresses at the apex of the notch at various RPM conditions.
7. Finally the cyclic life is evaluated for the compressor disc without notch for RPM conditions using S-N diagram.

All the above efforts of this analysis has given a strong foundation for designing low pressure compressor disc for fighter class aircraft, if an opportunity is given.

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