Compressor disc structural optimization for gas turbine engine

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Abstract. Gas turbine engine compressor increases the pressure and temperature of the air stream through it before entering the combustion chamber. Due to the extreme in rotational speed and high temperature, the gas turbine engine compressor is subjected to a complex force system that causes different failures mode in the compressor disc. Therefore, the design and calculation of compressor discs always consist of very high requirements with the problem of reducing the volume and increasing the durability of the disc structure. This paper presents a computational method that optimizes the design of a compressor disc geometry that meets performance and durability parameters but has the lowest mass based on finite element method. The calculation program is implemented by software Abaqus 6.13, algorithms and analysis results are processed through the Python 3 programming language.

1. Introduction
Compressor is classified as Class-I critical component of Gas turbine engine (GTE) and therefore the compressor disc is one of the most important component of the engine.

Figure 1. Cross section of engine compressor.

Considering the compressor structure of some similar size turbojet and turbofan engines worldwide today such as Microturbo TRI-60 [1], there are many similarities in term of crossectional geometry of the compressor disc. The disc cross-sections all have the largest outer rim to enough arrange the blade platform, the large inner bore to support concentrated stress and the narrow web body. This study explains the reasons for choosing the basic disc shape of GTE used in the axial compressor.

The purpose of this study is to provide an analysis tool to design and calculate the structure of the compressor disc to meet the engine operating condition and durability but with the smallest volume. With this programme the user can understand the design requirements for compressor disc in a GTE, such as material limitations, requirements for dimensions and weight limits.
In order to verify the accuracy and the reliability of the program, the calculation was performed with the 2nd stage-compressor of a turbojet engine, which is designed with a 4-stage compressor with a rotating speed of 24000 Rpm at the maximum thrust of 400 kgf. The compressor discs made of Aluminium 7075 belongs to the plasticity material family. The ultimate stress value of Aluminium 7075 in the calculation is 650 Mpa. Eq.

2. Theory

The compressor disc of the GTE are subjected to its own rotating force, gas load, thermal load and thrust of the engine as well as flight instrument maneuverability (accelerating, circling, landing…). The important design loads are body force, blade load and thermal load [2]. Due to the high rotating speed, the compressor disc is experiencing high stresses and vibrations. Hence, to prevent the failure of the disc, the stress field should be determined.

The methodology and structural formulation in this paper are based on the following assumptions:

- The disc material is considered homogeneous, isotropic and follow Hook’s law [3].
- The variation of the temperature gradient in the axial direction can be neglected.
- Tangential stress on the disc caused by gas load can be neglected.

![Figure 2. An element on the disc.](image)

![Figure 3. Displacement of an element.](image)

Considering an element on the disc, the equilibrium equation can be express as:

\[
\sum F_i = \frac{dF_t}{dr} dr - F_r dt + \rho r^2 \omega^2 dr dt = 0
\]

(1)

where: \( F_t \) and \( F_r \) denote force in tangential and radial direction, \( t \) and \( r \) are tangential and radial distance to the centroid of any differential element, \( \rho \) is disc’s mass density, \( \omega \) is rotational velocity.

The displacement of an element on the disc is shown in Figure 3, the relationship between straint and displacement can be written as:

\[
\varepsilon_t = \frac{du}{dr}
\]

(2)

\[
\varepsilon_r = \frac{(r + u) dt - r dt}{r dt} = \frac{u}{r}
\]

(3)

Strain of an element when considering thermal load:
\[ \varepsilon_{\text{total}} = \varepsilon_e + \varepsilon_{\text{thermal}} \]  
\[ \varepsilon_{\text{thermal}} = \alpha T \]  

The behavior of the disc material was assumed follow Hook’s law, so (4) can be written as:

\[ \varepsilon_e = \frac{1}{E}(\sigma_e - \nu \sigma_r) + \alpha T \]  
\[ \varepsilon_r = \frac{1}{E}(\sigma_r - \nu \sigma_e) + \alpha T \]  

where \( \varepsilon \) and \( \sigma \) are strain and stress, \( E \) is modulus of elasticity, \( T \) is temperature, \( u \) is displacement.

Considering disc at the steady state temperature distribution \([4]\), \( T(r) = R_T \log(r/R_b) \), \((R_r\) is outer rim radius, \( R_b \) is inner bore radius).

From equation (6) and (7), we have the stress of the element:

\[ \sigma_r = E\varepsilon_r + \nu\sigma_e - E\alpha T \]  
\[ \sigma_e = E\varepsilon_e + \nu\sigma_r - E\alpha T \]  

Substituting equation (2) (3) (6) (7) into (8) and (9):

\[ \sigma_r = \frac{E}{1-\nu^2} \left( \frac{du}{dr} + \nu \frac{u}{r} - (1+\nu)\alpha T \right) \]  
\[ \sigma_e = \frac{E}{1-\nu^2} \left( \frac{du}{dr} + \nu \frac{u}{r} - (1+\nu)\alpha T \right) \]  

The disc thickness as a linear equation of the radial distance:

\[ t = ar + b \]  

Equilibrium equation (1) is rewritten:

\[ \frac{d\sigma_r}{dr} + \frac{2ar+b}{ar^2+br} \sigma_r - \frac{1}{r} \sigma_\theta + \rho \omega^2 = 0 \]  

Substituting equation (10) and (11) into Eq. (13):

\[ \frac{d^2u}{dr^2} + \left( \frac{2ar+b}{ar^2+br} \right) \frac{du}{dr} + \left[ \frac{v \frac{1}{r^2} + v \left( \frac{2ar+b}{ar^2+br} \right) }{r^2} \right] u = \alpha (1+v) \left[ \frac{dT}{dr} + \left( \frac{2ar+b}{ar^2+br} - 1 \right) T \right] - \frac{\rho \omega^2 (1-v^2)}{E} r \]  

Equation (14) represent the disc displacement field and boundary condition. The difference between equation (14) and the algorithm of this study is that Eq. (14) include the variation of disc thickness and temperature as function of disc radius. The boundary condition can be taken at the disc bore and rim.

3. Optimization process

Equation (14) along with the boundary conditions can be solved by a number of different methods, such as superposition, shooting and transformation [5]. However, these methods are only appropriate for a continuum with a one-time variation in the coefficients of [2]. Concerning the boundary and working conditions of compressor disc, the disc thickness varies continuously with the disc radius, so the use of the above methods is no longer appropriate [6] [7]. This study uses finite element methods in combination with a programming language based on Python script [8] to solve the equation (14).
The initial geometry of compressor disc is rectangular with length $bw$ and width $tw$. The value $bw$ is limited by the outside diameter of disc rim, the value $tw$ is limited by the the width required to arrange the blade hub [9].

The overall process of this study is shown in figure 5. The user must input the combination load and the input dimensions into the program.

3.1. Load combination
The compressor disc subjected to three main forces, namely body force, gas load and the temperature through each stage of compressor. The rotating speed of 24000 Rpm, which equivalent to 2513 rad/s. The temperature was calculated by the thermal conductivity of the disc material.

Blade pressures on the disc rim are a combination of radial pressure and tangential pressure, in which, the radial pressure component is much larger than that of the tangential one [10, 12].
The study uses disc geometry digitization by coordinate control. The blades have complex structure in three dimension. The layer of blade can not model on the 2D plane, so the blades are removed and replaced by the equivalence force components acting at the center of gravity of the blades [11]. The Figure 6 shown the constraining point of the blades in 2D plane.

![Figure 6. Constraining point of the blades.](image)

The stress component caused in the tangential direction was neglected in this study. The blade pressure in the radial direction is 18.1 Mpa at the maximum rotation speed.

![Figure 7. Pressure of the blades on disc rim.](image)

As expected, the stress is only concentrated around the bore area (red zone in figure 1) and gradually decreases towards the outer rim. At the outer rim, stress is approximately equal to zero corresponding with the (14). The disc web has relatively small stress, ensuring only the structure and continuity of stress gradients. However, the rectangular disc causes the disc volume to be too large and reduces the rotating disc strength. The compressor disc will be divided into three main areas before the optimization process.

![Figure 8. Stress gradient of initial disc.](image)
The pairs of dimensions are given with a limit of mass and material constraints. The input dimension includes: outer rim radius $R_6$, the rim height $t_3$ (required to arrange the blade hub). The dimension constraints in the program:

- $R_6 > R_5 > R_4 > R_3 > R_2 > R_1 = \text{Bore radius}$
- $R_2 - R_1 = L_b$ (Bore height)
- $R_3 - R_2 = L_s$ (Bore fillet)
- $R_5 - R_4 = L_w$ (Web height)
- $R_6 - R_5 = L_r$ (Rim height)

3.2. Boundary condition
The inner bore of disc is assumed to be fixed to the shaft. The outer rim disc is free from and maintained at uniform temperature gradient. Thus the boundary condition are given by:

- $r = R_1, u_r = 0, T = 0$
- $r = R_6, \sigma_r = 0, dT/dr = T_v$

3.3. Optimization process
The program works by loop changing pair of dimensions on the disc and getting the maximum stress result. The stress limit conditions is assigned to each loop. The program will return a set of dimension that meets the initial assigned stresses.

The stress is concentrated at bore area and rim fillet (red zone in figure 10). The program always pays attention to the correlation of these two stress areas.
Figure 11. Correlation of disc stress via bore radius.

The bore radius $R_1$ is inversely proportional to the mass and the bore stress and proportional to the rim fillet stress (figure 11).

The users can easily evaluate the results and the correlation of each dimension to the stress on the disc through the graph of stress changes (figure 12) output from the program. The algorithm will remove non-conforming dimension values where the stress exceeds (red zone - figure 12) the tensile strength, and return the acceptable value (ideal region - figure 12).

After receiving the value from the symmetric model, the original dimension value set was defined within the range. Continue to make fine adjustments with the asymmetric model to get a more suitable dimension set.
4. Result and discussion

Figure 13. Cross-section of 2nd stage of compressor.

$Rf_2$ and $Rf_3$ values are taken from the compressor line of blade hub. The disc cross-section is no longer symmetrical.

| Value | (mm) | Value | (mm) | Value | (mm) |
|-------|------|-------|------|-------|------|
| $t_1$ | 3    | $R_1$ | 40   | $t_{51}$ | 17.6 |
| $t_2$ | 6    | $Lr_1$ | 69.5 | $t_{52}$ | 15   |
| $L_a$ | 10   | $Lr_2$ | 76.25 | $t_{61}$ | 5    |
| $L_s$ | 8    | $t_w$ | 6    | $t_{62}$ | 5    |
| $rf_1$ | 1    | $rf_2$ | 6    | $rf_3$ | 8    |

The final verification step of program was to adjust the small dimension on the disc as $rf_1$ to match the manufacturing process.

Table 2. Mass comparison of compressor disc.

| Version       | Mass   | Reduction ratio |
|---------------|--------|-----------------|
| Rectangular disc | 2.21 kg |                  |
| Input disc    | 1.62 kg | 15 %             |
| Last disc     | 1.28 kg | 41 %             |

Table 2 showcases the change of disc mass before and after the calculation program, the final mass value of the disc is 1.28 kg which was decreased by 40% from the initial disc and 20% from the input dimension set.

Figure 14. Stress field on the last disc.
The disc compressor is optimally reconstructed with the arrangement of blades on the rim face. Receiving the maximum stress on the compressor disc is 503 Mpa < 650 Mpa (ultimate strength), hence the disc compressor operates within the safe limits of the disc material.

5. Conclusion and scope
Finite element method can well solve the rotating disc problem with many variable parameters in the coefficient that the analytical solution cannot be obtained. The optimization process for weight reduction and better efficiency can also be carried out to rotating disc by finite element method.

A program was developed for structurally designing and calculating the compressor disc of GTE. The program of this study can be applied with the axial-compressor disc. It is necessary to strictly follow the input boundary conditions. The program was performed and validated with a compressor disc, the mass of disc is reduced by 20% compared with the input disc. However, this program does not apply to turbine discs as it depends on the different loads and its boundary conditions. Hence, the authors continue to perfect the program as the following scope:

- Take into account the fatigue of material in the optimization program.
- To get the best result, number of parameter can be increase.
- Developing the program for the turbine disc as its operating condition.

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References
[1] Lopez J P, Charles M 1992 Turboreacteur Simple et Performant a Compressor Axial Quadri-Étage et Turbine Axiale Pour la Propulsion de Missiles Subsoniques Turboréacteur TRI 60 - MICROTURBO
[2] Armand S C 1995 Structural Optimization Methodology for Rotating Disks of Aircraft Engines NASA Technical Memorandum
[3] Boresi P and Schmidt R J 2002 Advanced Mechanics of Material
[4] Rosyid A and Es-Saheb M 2014 Stress Analysis of Nonhomogeneous Rotating Disc with Arbitrarily Variable Thickness Using Finite Element Method Engineering and Technology
[5] Na T Y 1979 Computational Method in Engineering Boundary Value Problems," Academic Press, New York
[6] Sharma J N, S Dinkar and K Sheo 2011 Analysis of Stresses and Strains in a Rotating Homogeneous Thermoelastic Circular Disk by Using Finite Element Method Computational and Application
[7] 1994 Engineer Design Guide, Rotating Machinery NASA Lewis Research Center.
[8] Python Tutorial Python Scripting in ABAQUS
[9] 1965, Aerodynamics Design of Axial-Flow Compressors NASA Lewis Research Center - Washington D.C
[10] Hojjati M H, S Jafari, 2008 Theoretical and Numerical Analyses of Rotating Disc of Non-uniform Thickness and Density International Journal of Pressure Vessels and Piping 85 694-700
[11] E Onat, G W Klees 1979 A Method to Estimation Weight and Dimensions of Large and Small Gas Turbine Engines NASA, Vols. CR-159481
[12] Repetckii O, Ryzhikov I, Tien Quyet Nguyen 2016 Dynamics of gas turbine engines rotors taking into account non-linear effects, Vibroengineering PROCEDIA 8 361-5
[13] Ryzhikov I, Repetckii O and Tien Quyet Nguyen 2019 Software package for assessing parameter detuning influence on the durability of impellers of aircraft gas turbine engines, IOP Conference Series: Materials Science and Engineering 632(1) 012107