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Structural Analysis of Rotating Parts of an Ultra-micro Gas Turbine

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ABSTRACT

Ultra-micro gas turbine (UMGT) generator of button size is expected to be a next generation power source for various applications, such as aerospace systems, robots, etc. (Epstein, 1997) However, because of its tiny size, some structural problems must be thoroughly investigated before full-fledged development work. The problems will be rooted in such features as (1) A very short shaft connects high-temperature turbine and low-temperature impeller, (2) Two-dimensional shape, (3) High-revolution speed of 2,400 k rpm, (4) Silicone as the structural material. The results revealed that the impeller and turbine rotors deform into like shallow dishes caused by high temperature gradients and centrifugal forces.

NOMENCLATURE

Angular velocity ω [rad/s]
Coefficient of linear expansion $\alpha\ [/K]$
Density $\rho\ [kg/m^3]$
Modulus of elasticity $E\ [Pa]$
Poisson's ratio ν
Rotating speed $N\ [rpm]$
Thermal Conductivity $\lambda\ [W/mK]$

INTRODUCTION

This document describes a part of the results obtained in a UMGT feasibility study project funded by the New Energy and Industrial Technology Development Organization. (Yoshiki, 2002) This project tried to explore the technical challenges and solutions that would be necessary in developing button-size gas turbine electric generators. Besides a number of research topics in thermo-fluid dynamics, structural problems were recognized to be one of the challenges to be overcome.

The challenges are rooted in such features of the UMGT as (1) A very short shaft connects high-temperature turbine and low-temperature impeller, (2) Two-dimensional shape, (3) High-revolution speed of 2,400 k rpm, (4) Silicone as the structural material.

This study was focused on static thermal and stress analyses using finite element methods. The thermal boundary conditions used were based on the analysis in the same project. (Nagasaki, 2002)

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METHOD OF SIMULATION

Finite element analyses were conducted using NASTRAN software. Three types static analyses were done using steady state conditions.

(1) Temperature distribution calculation on impeller, shaft and turbine by steady-state thermal conduction analysis.
(2) Thermal stress, strain and deformation calculation by static load analysis using the temperature conditions obtained by Nagasaki (2002).
(3) Static stress, strain and deformation calculation caused by centrifugal load

MODELS, BOUNDARY CONDITIONS AND MATERIAL PROPERTIES

Finite element model

Models for this analysis were created from two dimensional cross-section CAD data, provided by UMGT research project in the Gas Turbine Society of Japan. (Fig.1) All the elements were made of three-dimensional solid elements. Because this analysis is for preliminary design study, only major rotating parts were modeled. In other words, such parts as bearings were omitted.

Boundary conditions

In setting the boundary conditions, one point of the bearing portion of the shaft was constrained in both axial and radial directions. The surface temperature of impeller and turbine was set to the values provided from the thermo-fluid dynamic analysis done in this UMGT project. The maximum temperature was 1300 K at the turbine inlet, and the minimum temperature was 300 K at the impeller inlet respectively.

Material properties

Plating metal layer by layer may have higher potential for stronger structure and smoother surface over silicone etching. However, silicone was chosen as the material that constructs the UMGT for its mature manufacturing process that can be available today for the complex figures of turbine blades. Material properties of the silicone were set to such values as follows.

ρ : 2.328 kg/dm^3 λ : 0.15 W/mK α : 3.7x10^{-6}/K.
E : 150x10^9 Pa ν : 0.17.

RESULTS OF THE SIMULATION ON THE ORIGINAL DESIGN

Temperature distribution

a) Large gradients in temperature contour were found. (Fig.2) The small distance between the turbine and the impeller causes these large gradients, while the absolute temperature values remain at the same value as those in large gas turbines.

b) The temperature at the bearing portion is somewhere in between the high and low temperature zone. This means the bearing temperature is subject to large variation.

Thermal stress and deformation

a) Heavy stress concentration is observed at the junction of the shaft-turbine and the shaft-impeller. (Fig.3) This should be due to the sudden change of rigidity and thermal energy flux.

b) The periphery of the turbine moves toward the impeller. (Fig4) The periphery of the impeller moves apart from the turbine. These dish-like deformations are due to the temperature difference between the both sides of the turbine disk or the impeller. The maximum displacement is 10 μm ranges.

Stress and deformation by centrifugal load

a) There are stress concentration areas on both turbine and impeller at the junctions of the discs and leading and/or trailing edges of the vanes.

b) The peripheries of the turbine and the impeller move closer to each other. The maximum displacements are 47 μm for the turbine and 38 μm for the impeller. (Fig.5)

DISCUSSION ON THE ORIGINAL DESIGN

From these calculated results, the structural design of the UMGT was found to contain challenging tasks. They are:

(1) Relaxation of the thermal stress concentration caused by steep temperature gradient is the major structural design challenge.

(2) Keeping the bearing temperature at desirable range by managing overall thermal management is another design challenge.

(3) The thermal and centrifugal loads add up dish like deformation of the turbine disc. Tip clearance management is necessary including housing design.

(4) Technology development for precise measurement and calculation of the temperature, load and deformation for the UMGT is crucial.

(5) The magnitudes of the deformation caused by the centrifugal and thermal loads are in the same order in this calculation. However, this is true only in a steady state conditions. As the rotating speed changes, the centrifugal loads changes. Thermal deformation may change slower than that caused by the centrifugal loads. This invites complex design considerations.

(6) Using centrifugal forces to counteract the thermal deformation should be included in the UMGT structural design.

INVESTIGATION OF ALTERNATIVE ROTOR DESIGNS

The alternative rotor designs

To solve the above problems, some design changes were investigated using finite element analysis.

The first alternative design is to build vanes as shell shaped structure with cavities. (Fig.6) This design aimed two effects. One is to reduce the centrifugal forces at the offset positions from disk center planes. The other is to increase thermal convection by increasing the surface areas.

The other alternative is to increase bending rigidity of the impeller and turbine disks by making them into sandwich structures. Adding another disks on the free edge of the vanes does this structural modification. (Fig.7)

The effects of the alternative designs

The temperature distribution and the thermal stress distribution of these two alternative designs remained in the same manner as those of the original design in a macroscopic sense. As a result, the deformations of the rotor peripheries in axial directions decreased just a little. (Table 1) However, the deformation caused by the centrifugal forces successfully reduced to a half in case of the cavity vanes and to one-third in case of the sandwich structure rotors.

HOUSING THERMAL DEFORMATION ANALYSIS

The UMGT housing design may have more freedom in its shape than that of the rotor design. However, the essential parts such as compressor and turbine housing should be in ax symmetrical forms that are compatible with the rotating parts. Therefore, the model is created based on the figures appeared in the MIT project. The rotating parts were eliminated in this housing deformation analysis. Heat flow through rotating parts may not be negligible, but the authors avoided too much complication.

The temperature distribution calculated illustrates that the temperature at the rotor bearing rises over 900 K. (Fig.8) This temperature is within 100 K range of the shaft temperature in the bearing portion. However, in selecting the type and the material for the bearing, this high temperature should be carefully considered.

Because the turbine housing bulkhead temperature rises up to 1300K while the inlet side bulkhead remains at 300K, the thermal expansion difference between the two is significant. The housing deforms in a dish like shape. (Fig.9) The maximum displacements at the peripheries are up to 27 μm for the compressor housing and 33 μm for the turbine housing respectively. (Table 2)

The thermal deformation directions of the housing are the same as the rotor deformation directions excluding the impeller deformation under centrifugal loads. Therefore, the deformation management problem is to reduce the displacement difference between the rotor and the housing, especially at impeller side, and not to bring all the deformations to zero. However, this is the true only in a steady state conditions. The transient conditions such as start and stop operations are not considered yet.

CONCLUSIONS

(1) Important aspects of the UMGT structural design were investigated using finite element modeling and static analyses.
(2) Large temperature gradient was found to be one of the design challenges that the UMGT project will face.
(3) The stress concentration at the junction between the shaft and the disks should be carefully considered.
(4) The temperature at the bearing may rise up to 900 K unless cooling functions are embedded in design. This temperature must be considered in bearing type and material selections.
(5) The impeller and the turbine disks deform in a dish like shape by both thermal and centrifugal loads.
(6) The magnitudes of the deformation are in the same order among the load types.
(7) The housing deforms by the temperature gradient in the same order as the rotor.
(8) Great care must be taken in the UMGT structural design considering the above-mentioned phenomena.

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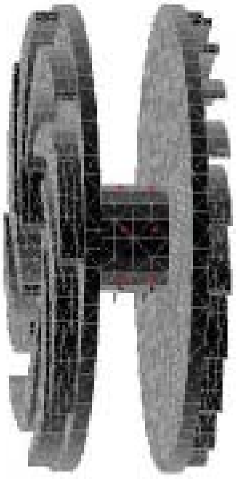

Fig. 1 The finite element model of the UMGT rotor

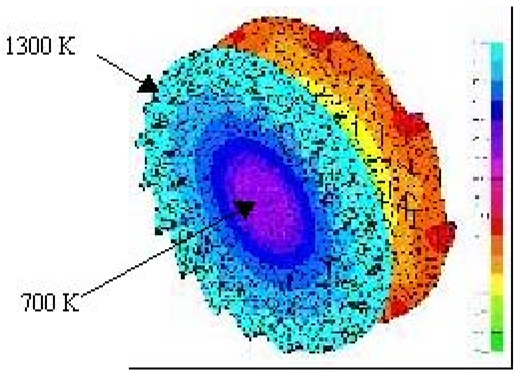

Fig.2 Temperature distribution over the turbine surface

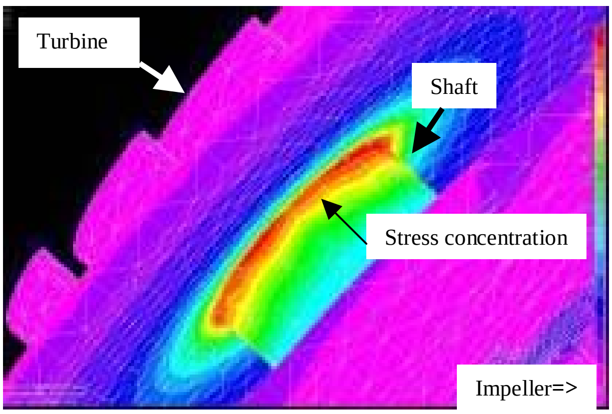

Fig. 3 Thermal stress concentration at the shaft and the disk connection.

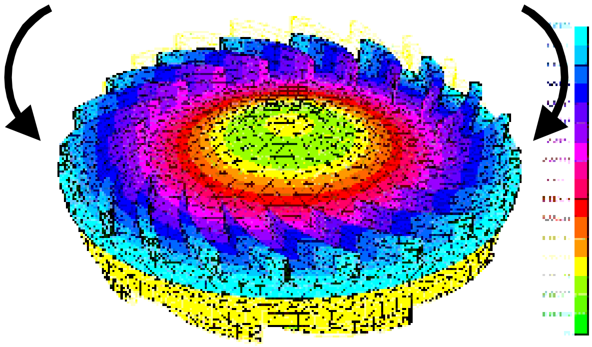

Fig.4 Thermal deformation of the turbine disk. The periphery moves toward the impeller and turns itself into a dish shape.

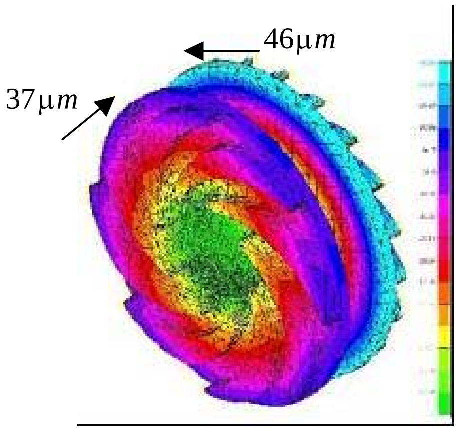

Fig.5 Deformation of the impeller (left) and the turbine (right) under the centrifugal forces.

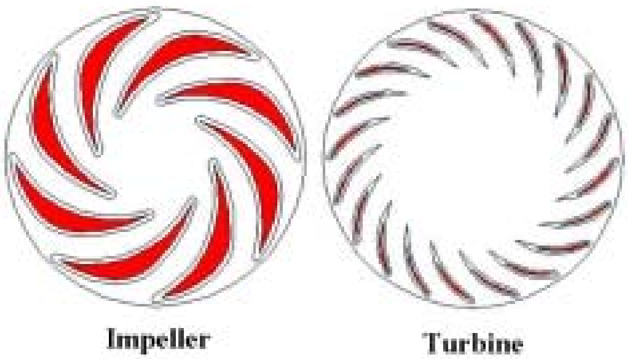

Fig.6 Alternative design A: Vanes with cavities. Vane shell thickness is 0.075mm.

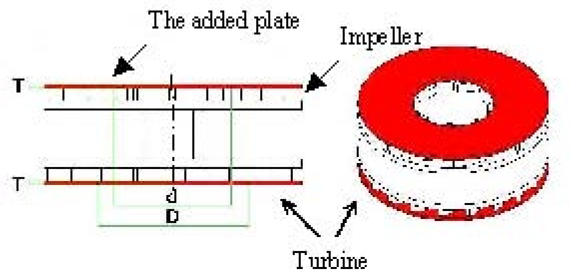

Fig.7 Alternative design b: Sandwich structure impeller and turbine with added plates.

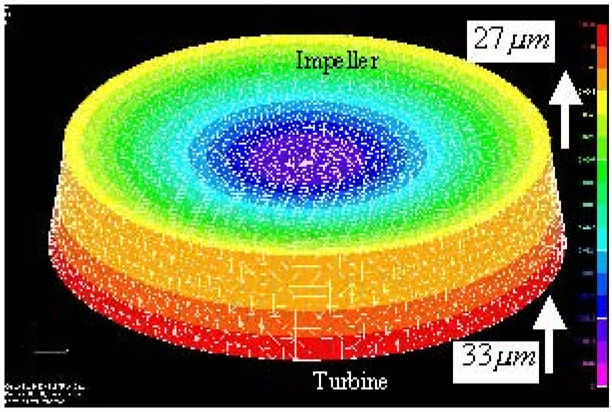

Fig.9 Deformation of the UMGT housing.

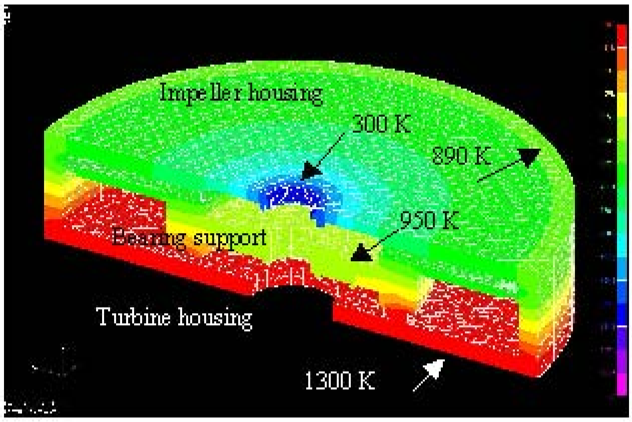

Fig.8 Temperature distribution over the UMGT housing.

Table 1 Effects of alternative rotor designs on the deformation suppression at the periphery of the disks.
The unit of the numeric data is μm. The displacement values are positive in the axial direction from the turbine to the impeller.

Design			Thermal load		Centrifugal load	
			Impeller	Turbine	Impeller	Turbine
Original			7.50	15.0	**-38.0**	47.0
A	Vanes with cavities		7.21	14.4	**-20.3**	23.2
	Sandwich structure	added plate thickness				
B1		0.075mm	6.32	12.6	**-13.8**	15.8
B2		0.10mm	6.26	12.5	**-12.7**	14.5
B3		0.15mm	6.16	12.3	**-10.5**	12.9

Table 2 Thermal deformation of the housing.
The unit and the direction are the same as those in Table 1.

	Thermal load		Centrifugal load	
	Impeller side	Turbine side	Impeller	Turbine
Housing deformation	27	33	*****	