APISAT 2019

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ENGINEERS AUSTRALIA

The Optimal Design of Turbine Rotor For A Micro Gas Turbine

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- 1、Introduction
- 2、Optimization background
- 3、Optimal design
- 4、Test Verification
- 5、Conclusions

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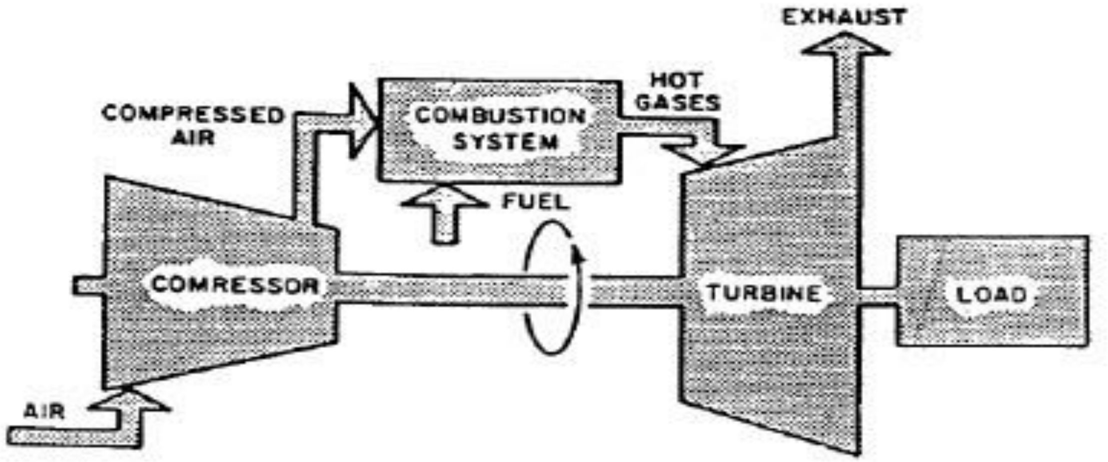
Introduction

- **Micro gas turbines**： Newly developed small thermal engines over these years.

- **Power**:Ranges from tens to hundreds kilowatts



Introduction

- Advantages：

1、small volume;

2、light weight;

3、wide range of suitable fuel；

4、 can be installed close to the users （which can significantly improve the reliability of the electric power supply）

- Widely used

Introduction

- Difficulties in rotor designing

1、increased turbine inlet temperature -the requirements of improve turbine efficiency

2、high speed（centrifugal loading）-spinning of the component results in internal stresses

3、thermal gradients -temperature variation across the metal causes different levels of deformation that lead to stressed

4、aerodynamics/pressure loading – pressure variations between the disk backface and hub, and between pressure side and suction side of blade , lead to stresses

5、interface loads – any other external loads as a result of inpeller attachment,etc

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Optimization Backgoround

- A new micro gas turbine designed by ENNP
- The power of this micro gas turbine is less than 500kw
- Centrifugal compressor、single-cylinder combustion chamber and two-stage axial flow turbine are used
- The structure of this two-stage axial flow turbine was blisk. The material of disks are nickel base superal loys

Optimization Backgoround

- During the performance test of this micro gas turbine, the low pressure turbine blisk suddenly broken

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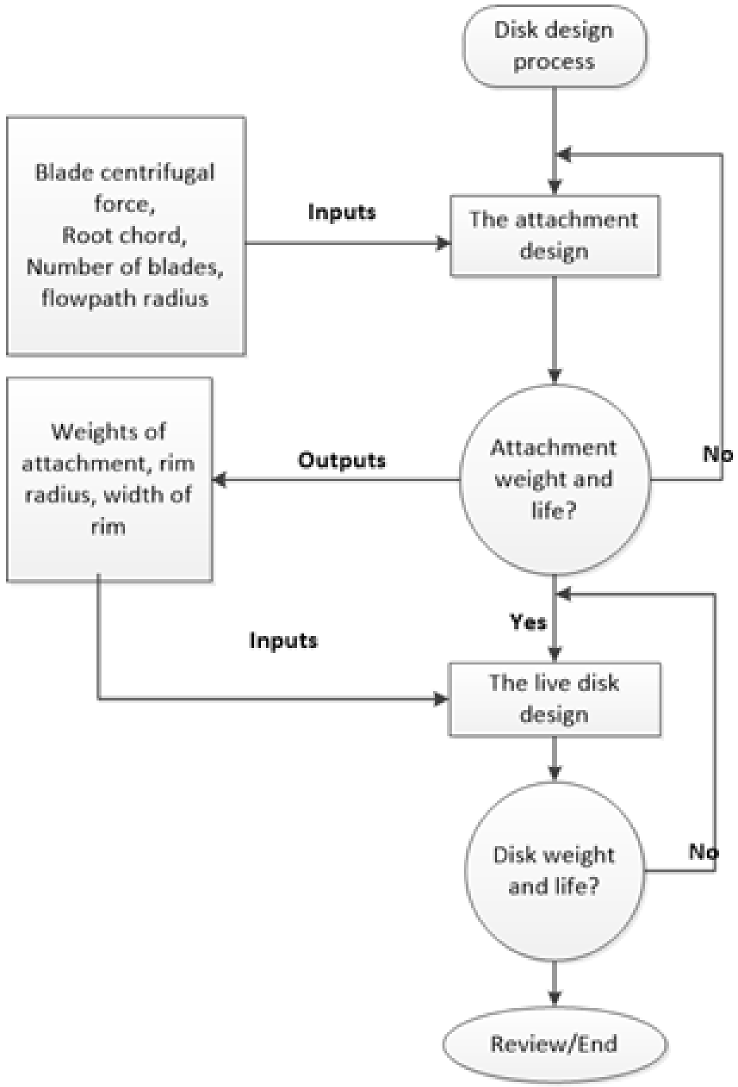
Optimal design-Objects

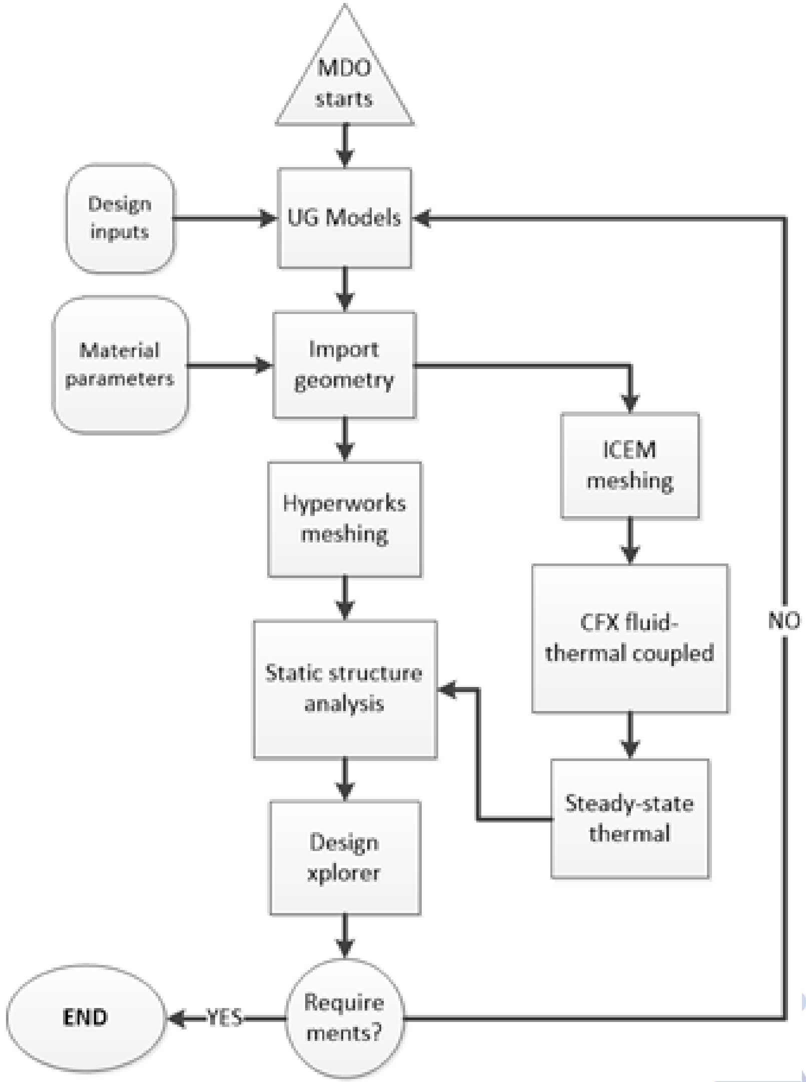
- The two-stage turbine blisk in this micro gas turbine with 36 blades in each stage were researched

- The two-dimension meshed model were chosen during the optimal design of this gas turbine disk



Optimal design-Design process

- Based on the CAE softwares(ANSYS and HYPERMESH)
- Blade aerodynamic forces was provided by CFD guys.
- The flow channel and aerodynamic would not be optimized in this paper



Optimal design-Thermal analysis

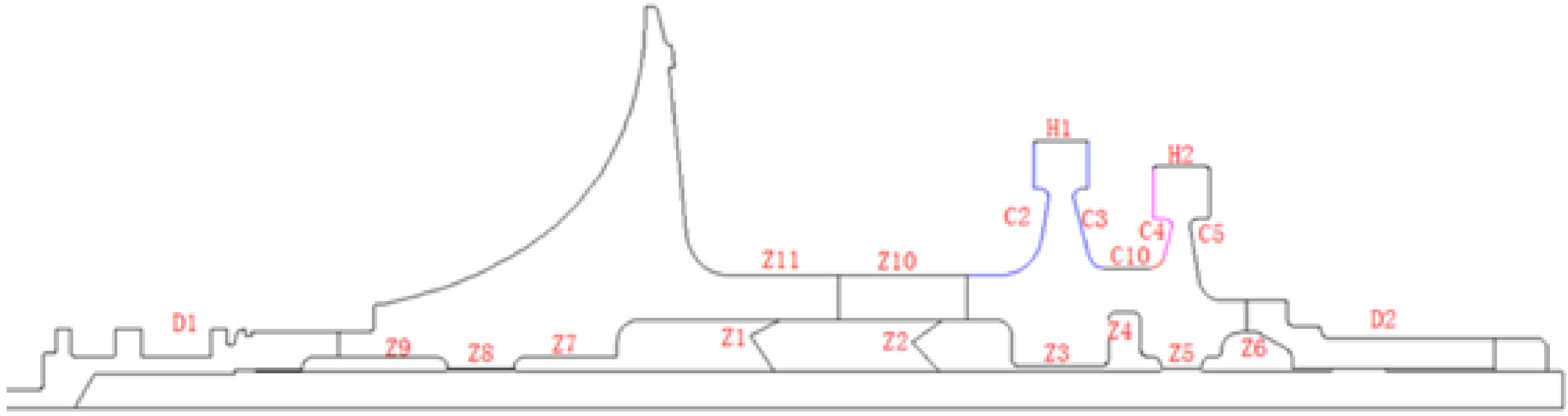
- The heat exchange boundary partition was shown in Fig.5.

1、C2 ~ C5 and C10 were calculated by surface heat transfer criterion of rotating cavity with radial air flow.

2、Z1~Z11 were calculated by free cavity rotating disk indicates heat transfer criterion.

3、H1 and H2 were calculated by heat transfer criterion of gas side edge plate.

4、D1 and D2 were treated as of natural convective boundary conditions.



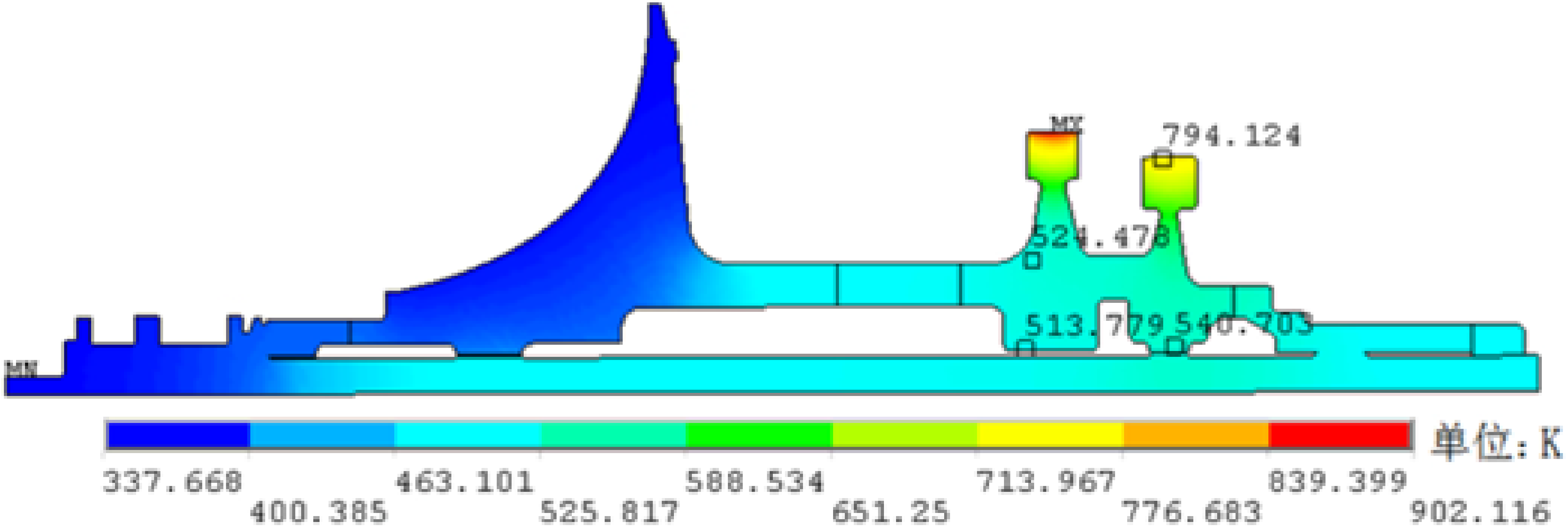
Optimal design-Thermal analysis

- The results of heat transfer boundary

Partition number	Heat transfer coefficients ($W/K \cdot m^2$)	Heat transfer temperature (K)	Partition number	Heat transfer coefficients ($W/K \cdot m^2$)	Heat transfer temperature (K)
C2	582	507	Z4	52	520
C3	588	537	Z5	456	535
C4	415	537	Z6	103	530
C5	386	567	Z7	47	500
C10	685	517	Z8	128	505
H1	87	515	Z9	47	505
H2	62	535	Z10	585	495
Z1	9	490	Z11	372	495
Z2	9	490	D1	20	288
Z3	866	505	D2	20	288

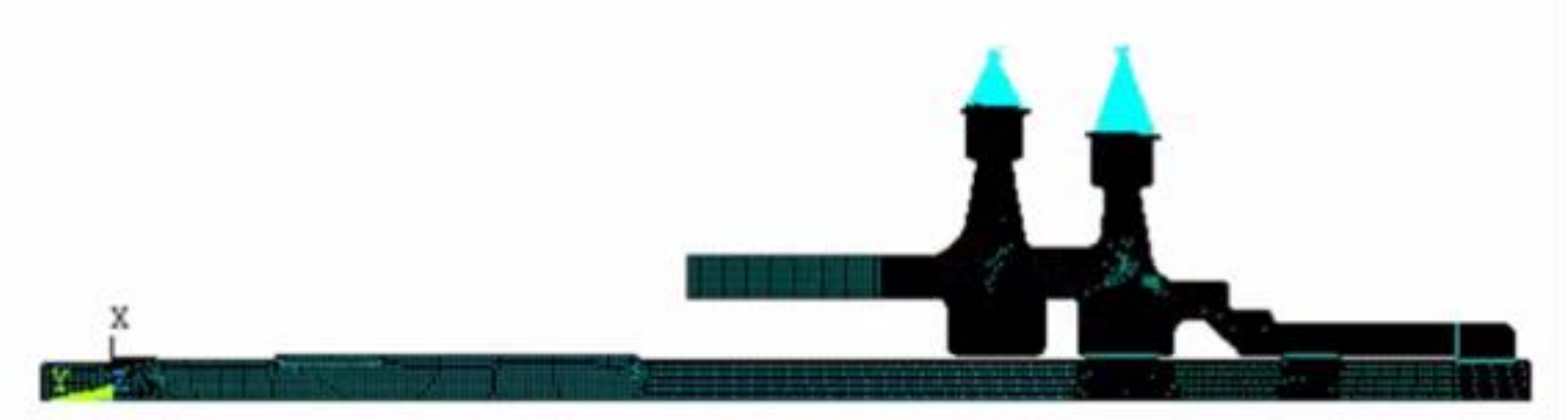
Optimal design-Thermal analysis

• The distribution of temperature



Optimal design-Thermal-structure analysis

- Most of the model was meshed by axisymmetric element. For non-axisymmetric structures, such as holes and roots, the plane stress element with thickness were used.
- Boundary conditions contained axial displacement constraints in the tooth surface which was connected with the compressor and the contact between different componments.

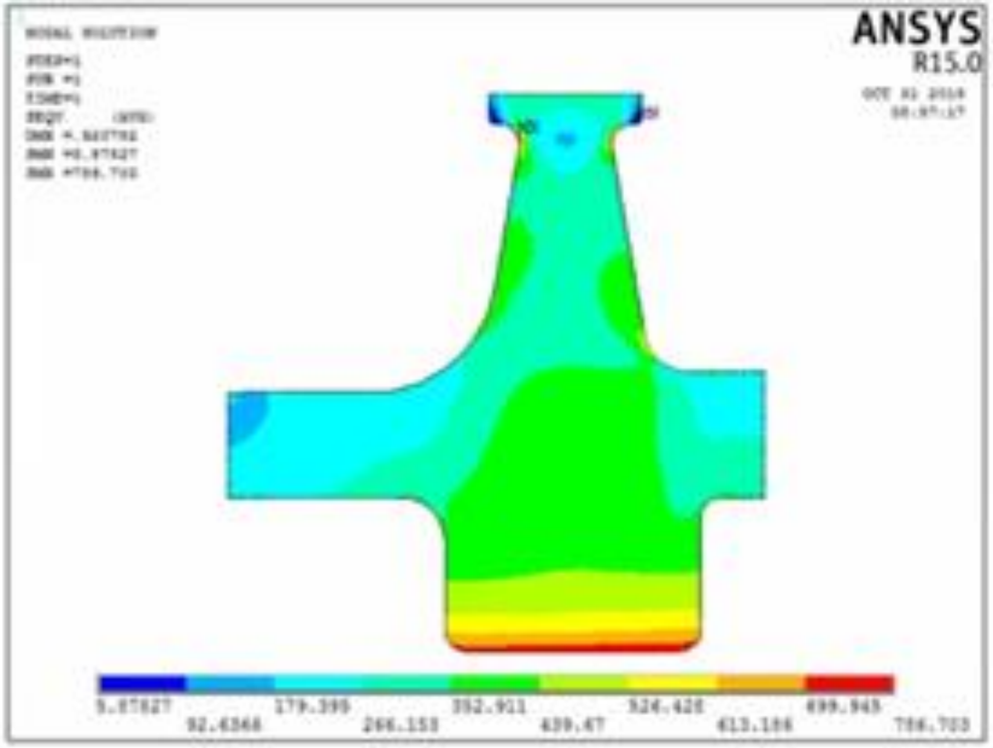


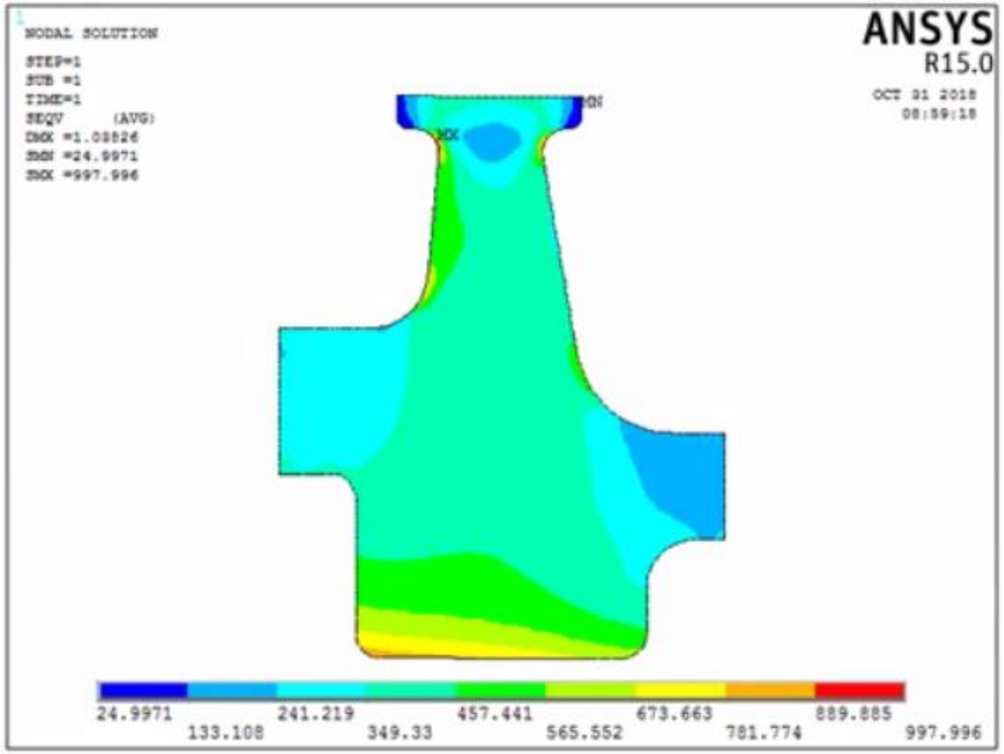
Optimal design-Thermal-structure analysis

- The mass point was used to simulate the blades. We established an MASS21 element in the center of the mass of the blades. The mass of this element was assigned as the mass of all the blades. The MASS21 element was connected to the disk by the MPC184 element.
- Aerodynamic force of blades was simulated by MASS21 element in the center of aerodynamic force.
- Centrifugal loads and cavity pressure also need to be considered in our analysis

Optimal design-Optimal results

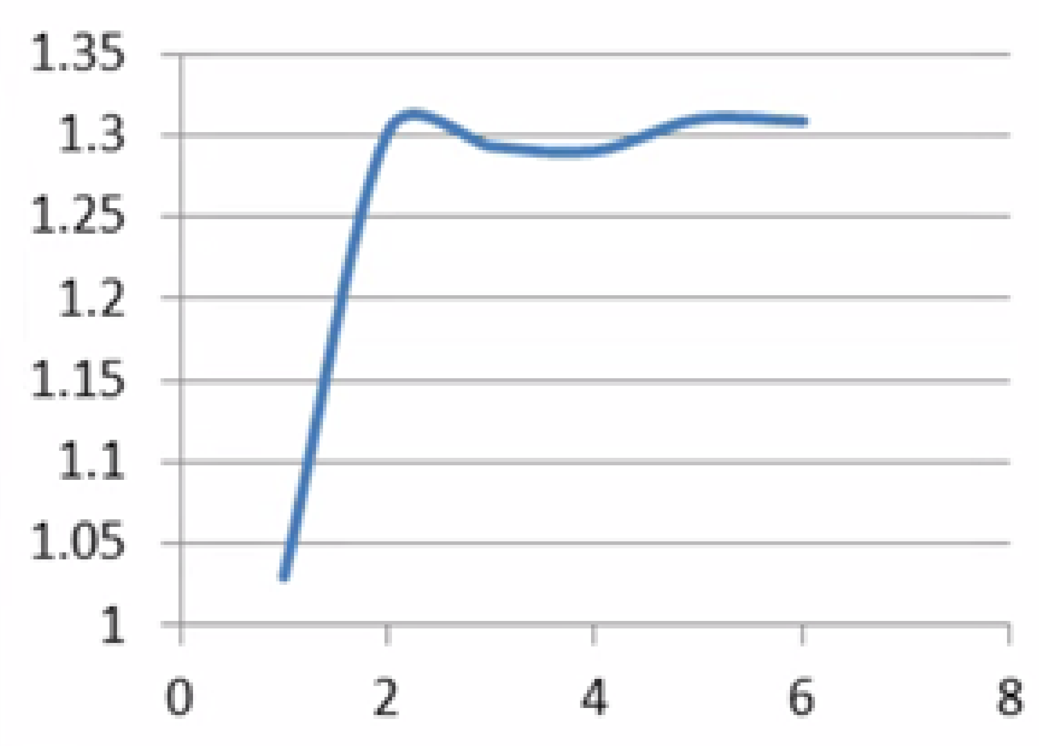
- About six wheels of optimization design
- The maximum stress of disk1 was about 787MPa.
- The maximum stress of disk2 was about 998MPa.

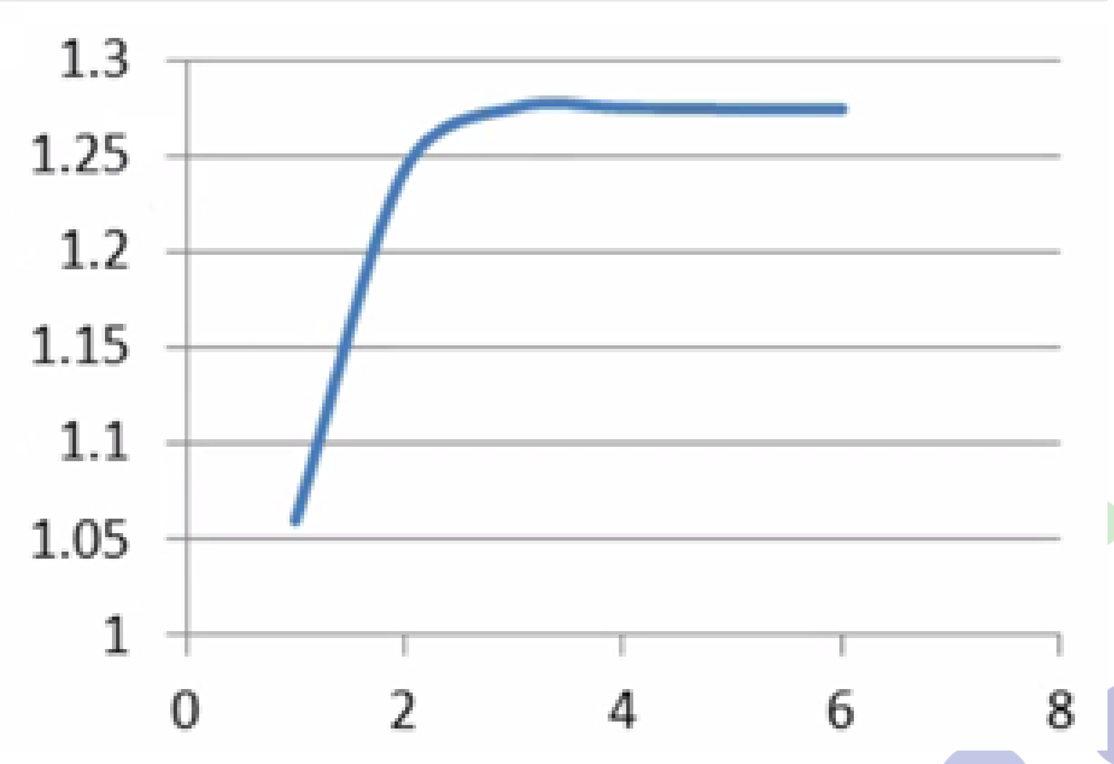




Optimal design-Optimal results

- The improvement of circumferential fracture margin of our two-stage turbine disk





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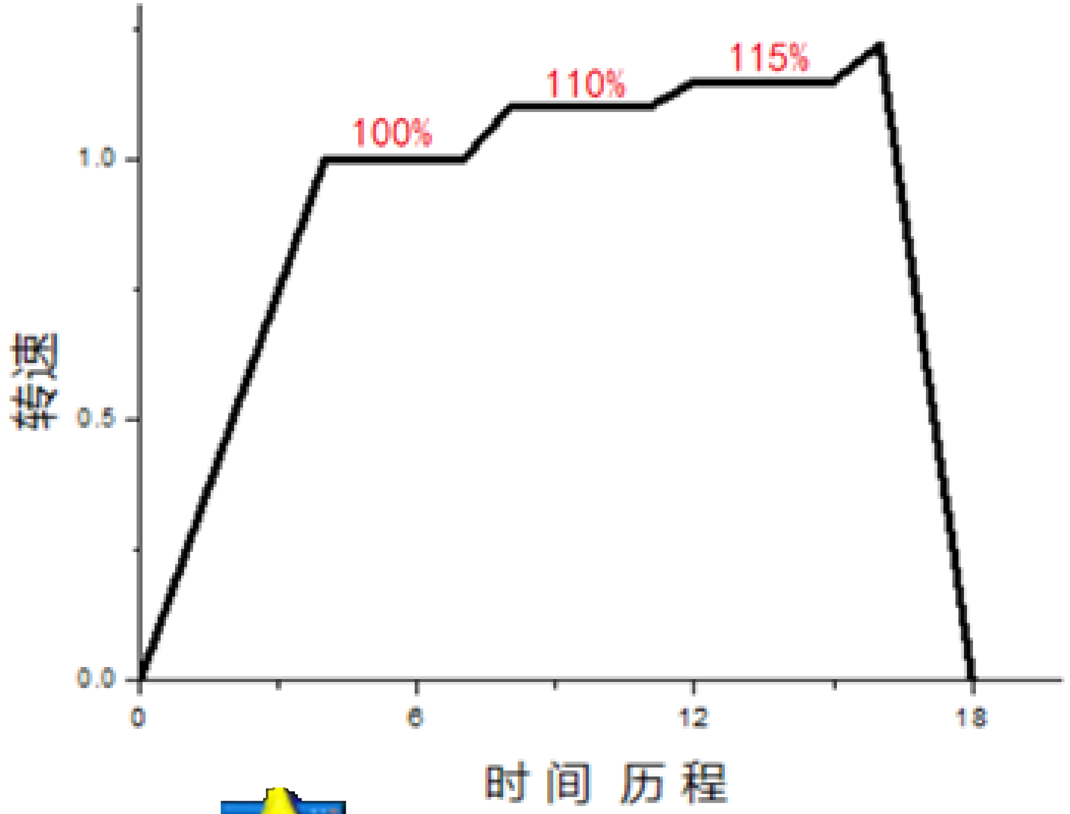
Test Verification

- An over-speed test for this turbine rotor were carried out

- The result of this test shows that the optimization design for this turbine rotor was very successful

- The deviation of ruptured speed between the test and theoretical analysis was only 1.35%



rupture test

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Conclusions

- Turbine disc works at high speed environment withstanding the tremendous centrifugal force and aerodynamic force caused by blades that involves complex aerodynamics, heat transfer, structural deformation, and other complex situations.
- Based on the accomplishment of data transfer between different disciplines and different meshed model, we finished multidisciplinary design optimization of our turbine disk.
- During the use of our micro gas turbine, the new turbine disks worked safely and reliably without damage.

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Thank you !

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