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1. INTRODUCTION

This document describes the 3D finite element model of the HTR turbine disc + blade assembly and the calculations performed to demonstrate the conclusions reached in the previous documents [1] and [2].

This document complements the analysis addressed in Ref [1] in which it is concluded that under the stress level reached, the expected lifetime for the turbine is less than attainable. To reduce the stress level, it is suggested in the document [1] that the disc be cooled.

Ref [2] studies and analyses how to reduce the maximum temperature reached. It also addresses the calculation of cooling conditions and the final temperature of the cooled rotor.

Based on the conditions reached in Ref [2], the range of temperatures reached in the disc + blade assembly is analysed, as well as study the range of stresses obtained for this configuration due to the different loads on the assembly.

In particular, the analysis focuses on whether the distribution of stresses on the disc is affected, especially around the cooling holes, because of the change in turbine geometry due to the cooling introduced.

2. 3D FINITE ELEMENT MODEL

The IGES model employed for the establishment of the 3D model is presented in Ref. [1].

The finite element software used is PATRAN for preprocessing, and SAMCEF for the analysis and postprocessing. The elements featured in the model are 8-node and 6-node volume type elements, as well as 4-node shell elements.

The 3D model of the disc + blade represents a section which includes a complete blade and the corresponding disc section; in other words, the disc section has a circumference of 2.62774º obtained by dividing 360º by the number of blades (137) in the first stage.

The 4 mm diameter cooling hole in this disc section has been modelled exactly in the case studied in Ref [2].

In Ref [2] it is concluded that 77 x 3 mm diameter holes are sufficient to attain 60,000 hours of operation. However, to facilitate creation of the 3D model, 137 holes have been introduced on the basis that only one blade has been modelled with its corresponding disc section, and in the first-stage turbine set –the purpose of this study- there are 137 blades.

To build the model with only 77 holes, it would be necessary to model two blades and their corresponding disc section which would have resulted in a model too large to handle.

Figures 2-1 and 2-2 illustrate the three-dimensional model of the blade and disc, respectively.

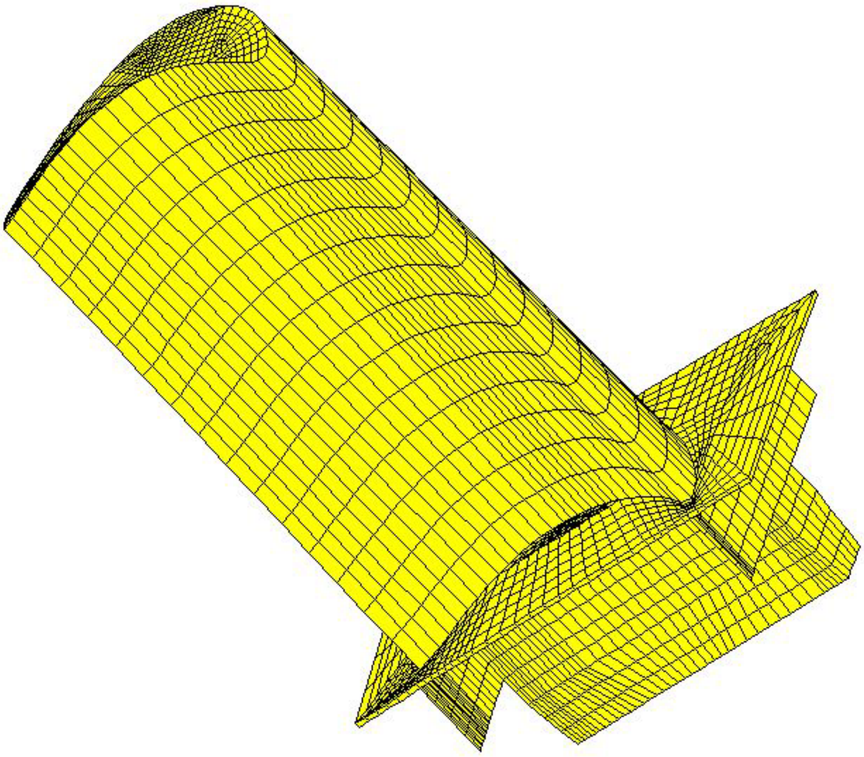

Figure 2-1 – 3D FEM – Blade

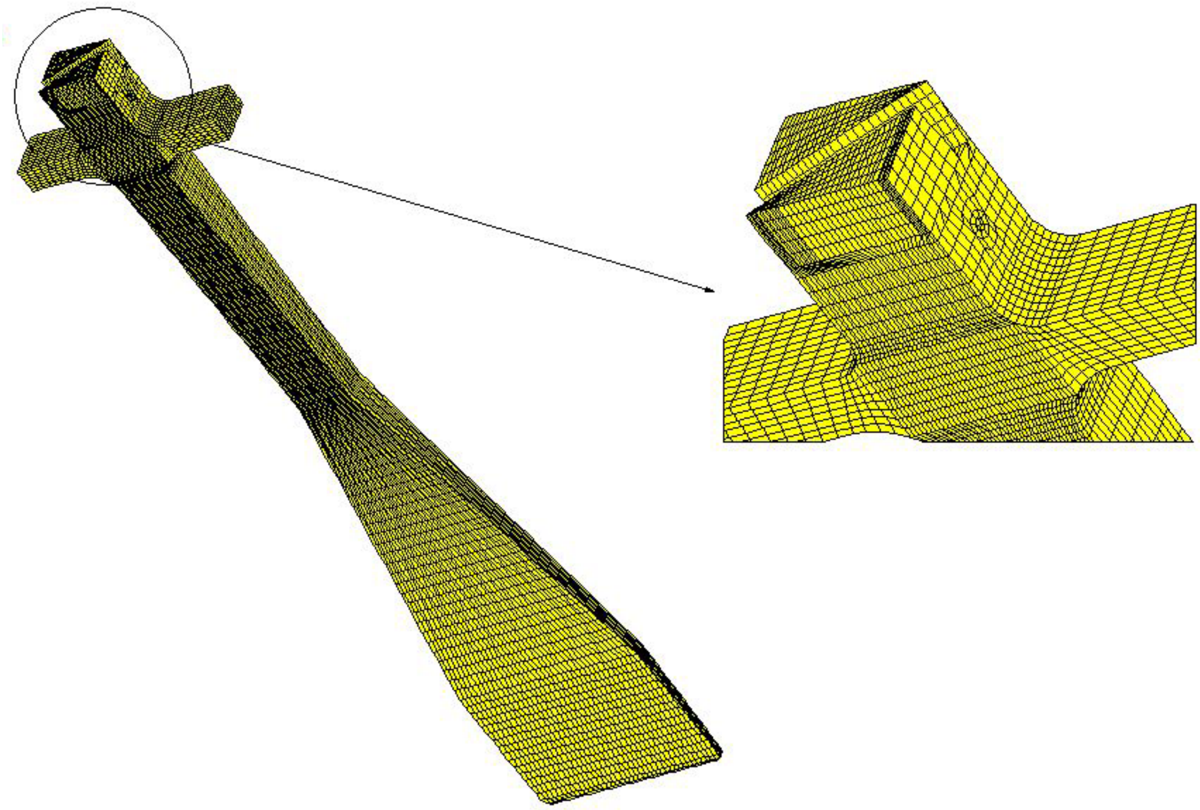

Figure 2-2 - 3D FEM of the Disc

The materials used in the analysis are Inconel 718 for the blade and Udimet 720 for the disc, the physical properties of which (taken from http://www.specialmetals.com and http://thermotech.co.uk) are as follows:

	Young's Modulus (GPa)	**Poisson Coefficient**	**Thermal Expansion Coefficient**	**Density (kg/m^3)**	**Thermal Conductivity (W/mºC)**
Inconel 718	146.5	0.29	$9.2\ 10^{-6}$	8190	22.5
Udimet 720	166.5	0.29	$8.2\ 10^{-6}$	8080	20.0

Since only one sector of the disc was modelled, suitable cyclic symmetry was introduced into the 3D model so that it could be correctly fixed in space. The model was built so that the displacements of the two cutting planes of the disc section were equal; in other words, that the two planes moved in the same way.

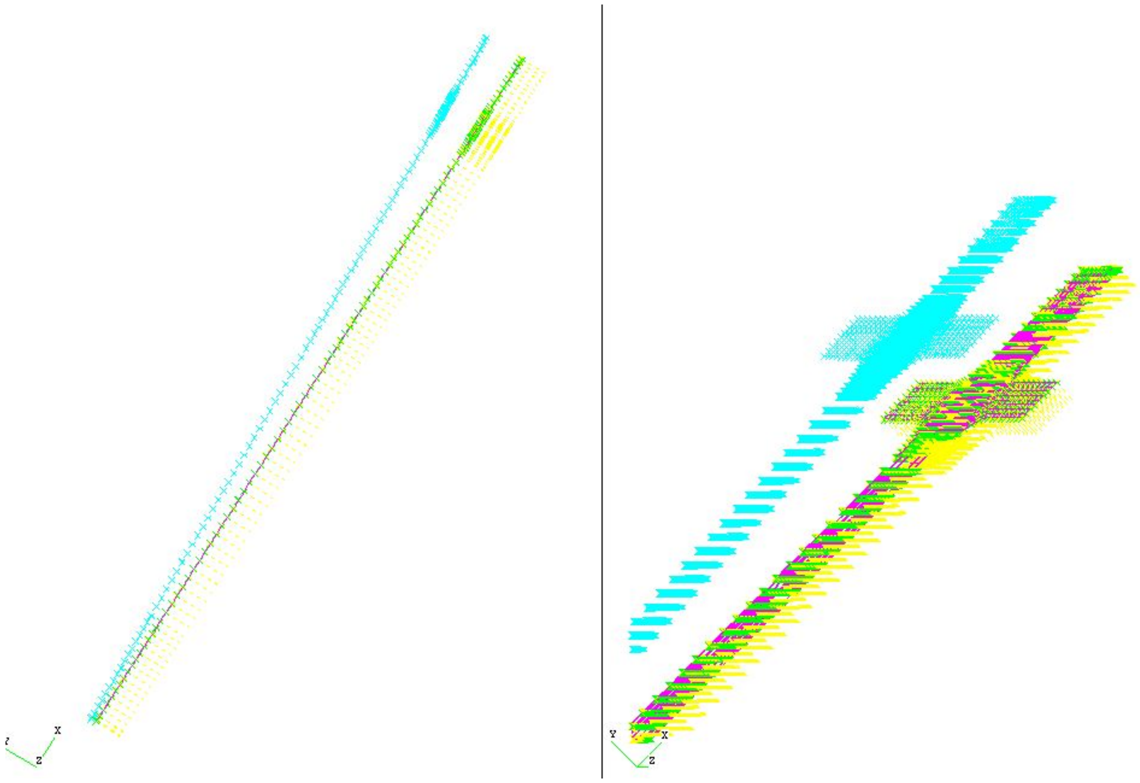

Figure 2-3 - Cyclic Symmetry Condition

3. THERMAL ANALYSIS

The following are the boundary conditions introduced into the model:

- **Helium temperature 815ºC**. A fixed temperature equal to that of the helium was introduced into the nodes of the blade walls in contact with the fluid

- **Cooling temperature 515ºC**. The temperature of the nodes of the inside of the hole was fixed to that of the cooling fluid. The temperature value was taken from the study carried out in Ref [2]

  The option of fixing the temperatures of the nodes instead of applying convection to the elements washed by the two currents (the main current and the cooling current) was selected. This direct application of the temperature is more conservative than the use of convection because, with the application of the latter, the nodes of the blade in contact with the main current would not reach the temperature of the helium (815ºC).

  The boundary conditions specified above are illustrated in Figure 3-1 below.

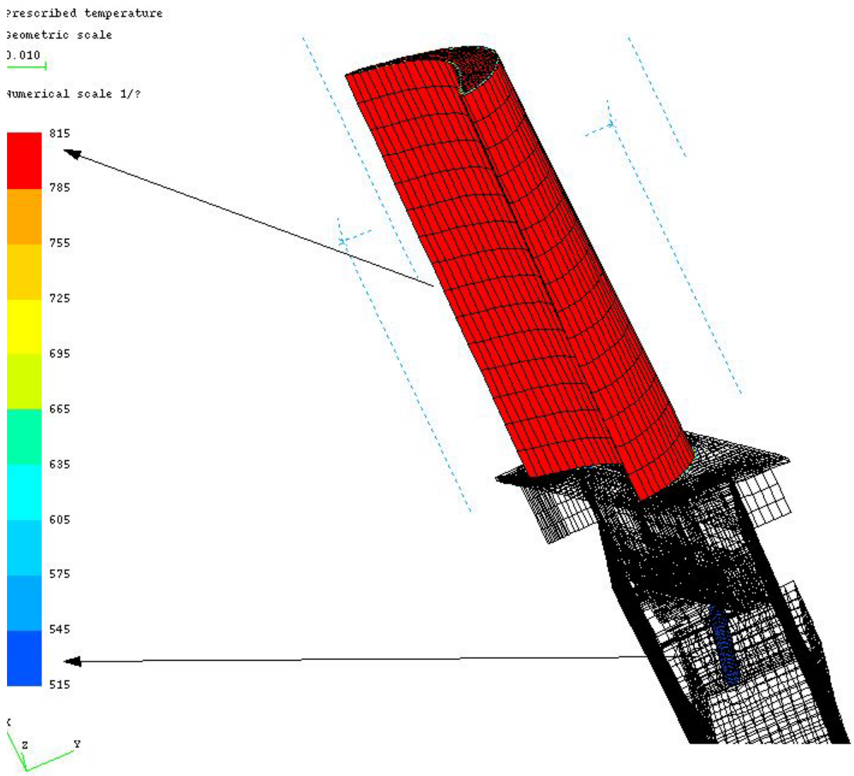

Figure 3-1 - Temperatures Imposed on the Thermal Analysis

- **Contact of the foot of the blade with the disc**. To model metal/metal contact, it was decided to simulate a 200 W/m^2 ºC conductance resistance identical to that used for the axisymmetric case in Ref [1], introduced between the nodes of the foot of the blade and the nodes of the disc that coincide with those of the foot of the blade

The figures included in Appendix A illustrate the temperature distribution obtained in the disc + blade assembly. This range of temperatures will be that used as a limit condition for the thermal-mechanical analysis which is addressed in the chapter on mechanical analysis.

It can be seen from Figure A1 in Attachment A that the maximum temperature reached is 566ºC in the area of the disc in contact with the foot of the blade, and in the bottom area of the disc, where the greatest stresses act due to rotation, the temperature reached is ~520ºC.

The cooling considered in the above thermal analysis allows achievement of the objective of reducing the disc temperature to values allowing stresses equal to or less than those obtained in Ref [1] and enabling the turbine to reach the required ~60,000 hours of operation in normal conditions.

The next chapter examines whether the geometric changes made to the disc affect the stresses reached due to the different loads which affect the assembly.

4. MECHANICAL ANALYSIS

The following is an analysis of the disc + blade 3D model to obtain the range of stresses due to loads that affect the model (rotation, pressure and temperature).

The aim is to verify that the required turbine life of 60,000 hours can be attained with the stresses reached in the disc at operating temperature, as well as to verify that the stresses induced around the hole by cooling do not produce a concentration of high stresses.

The blade is joined to the disc by securing the blade and disc node displacements (all directions) at the height of the foot of the blade, as illustrated in Figure 4-1.

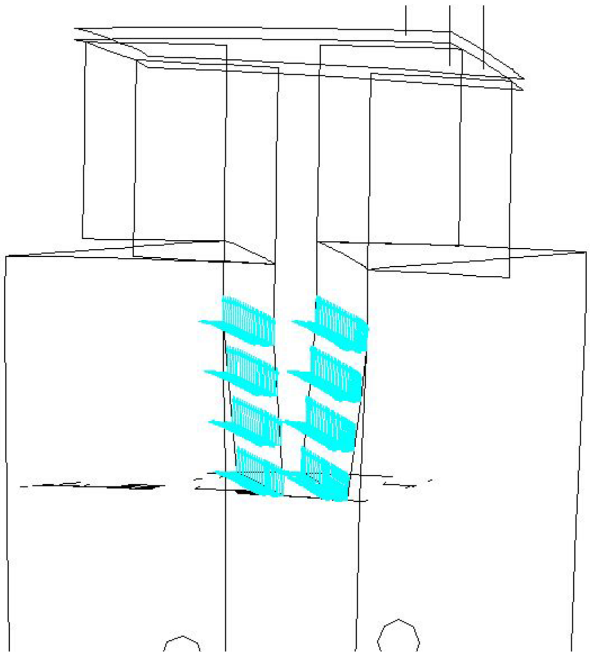

Figure 4-1 - Union of the Blade/Disc Displacements at the Height of the Blade Foot

4.1 LOADS DUE TO ROTATION

In this analysis the model is subjected to a rotation speed of 3000 rpm, which is the rated rotation speed of the turbine; ie, 314.16 rad/s.

The results are given in Figures B-1 to B-3 of Attachment B. It may be seen that, as in the case of the axisymmetric model addressed in Ref [1], the greatest stresses are found at the bottom of the disc (insider radius) and they are of the order of 240 MPa which coincide with the stresses found in the axisymmetric analysis Ref [1].

Stresses of ~170 MPa due to rotation appear around the cooling hole.

4.2 LOADS DUE TO PRESSURE

In this case the helium pressure produces loads on the blade surface, the distribution of which is obtained in accordance with the provisions of Ref [3] and illustrated in Figure 4-2.

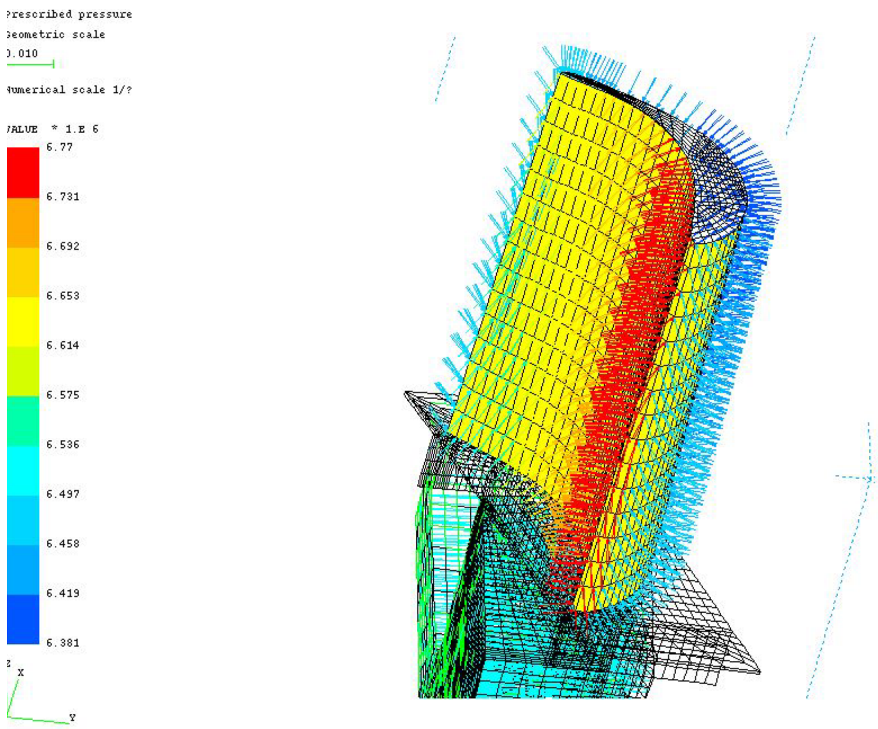

Figure 4.2 - Pressure Loads on the Blade Surface

The stresses obtained on the blade and the disc due to the pressure are illustrated in Figures B-4 to B-6 of Appendix B.

Maximum stresses on the blade are reached for this load. More specifically, there is a stress peak upon contact between the foot of the blade and the disc plates. In this area a stress concentration of 480 MPa is observed.

This stress is localised in a very specific area of the foot of the blade -the place where the plate and foot are joined- because of the bending of the blade due to the unequal pressure on its faces. On the rest of the blade, the stress values vary from 200 MPa to 250 MPa.

4.3 LOADS DUE TO TEMPERATURE

For this calculation the range of temperatures obtained in the thermal analysis has been used, the distribution of which is illustrated in the following figure:

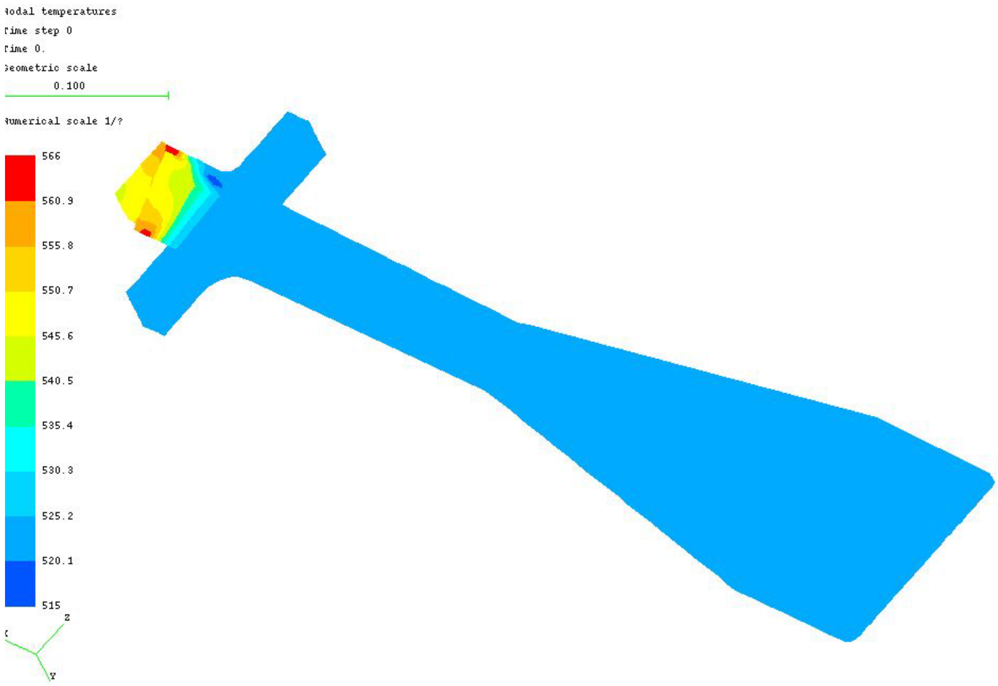

Figure 4-3 - Distribution of Loads due to Temperature

Since the main objective of these analyses is to determine how the introduction into the model of the cooling holes affects the geometry of the disc, the thermal-mechanical analysis has been performed exclusively on the disc, without taking into account the blade.

Since the blade has no restriction of any kind, apart from the joint to the disc which is discussed later, it can expand freely which means it will be practically free from temperature-induced stress.

The results obtained in this case are given in Figures B-7 and B-8 of Appendix B, where it may be seen that the stresses obtained at the height of the hole are of the order of 15 MPa.

When the foot of the blade reaches a higher temperature than the disc area where it is housed, it must be ensured that there is a gap between the foot of the blade and the disc which is wide enough for the blade to expand without entering into contact with the disc and giving rise to peak stress due to compression:

- Tmax at the foot of the blade = 720ºC
- Tmin in the disc at the height of the foot of the blade = 540ºC
- Maximum width of the foot of the blade = 6.5 mm
- $(\Delta\rho/\rho)$blade = 720 x 9.2 ∗ 10-6 = 0.0066 → $\Delta\rho$ = 0.0066 x 6.5 = 0.0429 mm
- $(\Delta\rho/\rho)$disc = 540 x 8.2 ∗ 10-6 = 0.0044 → $\Delta\rho$ = 0.0044 x 6.5 = 0.0286 mm

Therefore, to ensure that they do not enter into contact due to thermal expansion, the gap has to be wider than **0.01 mm**.

4.4 RESULTS

Following the study of the partial results of the stresses on the disc and the blade due to different loads, the resulting distribution of stresses will be analysed to determine the combination of loads.

Figures B-9 to B-12 of Appendix B illustrate the distribution of stresses for the blade and the disc. As can be observed, the maximum stress on the disc (255 MPa) is located at the bottom of the disc. Around the cooling hole there are maximum stresses of 143 MPa.

The greatest stress on the blade, of the order of 560 MPa, is located in the same place as that due to pressure; in other words, in the area of the joint between the foot of the blade and the disc. Maximum stresses on the rest of the blade reach values of between 220 MPa and 280 MPa.

With this stress level (255 MPa on the internal radius of the disc), according to Ref [1] the temperature of the disc in this area must be less than 776ºC in order to ensure

60,000 hours' operation. Therefore, since a lower temperature is reached in this area by cooling, the expected useful life can be attained.

5. CONCLUSIONS

The stress level of the assembly with the new geometry and subjected to the rotation, pressure and temperature loads, is not markedly greater than that reached in the initial configuration. In the case of the disc, therefore, maximum stresses due to rotation are maintained within the lower radius, and in the case of the blade stresses due to pressure are maintained at the height of the joint between the disc plates and the foot of the blade, the latter being a peak stress.

It is also demonstrated that the introduction of cooling holes into the disc does not produce an overly high concentration of stresses; maximum values of 143 MPa are reached in this area.

With regard to the blade, a local stress of 560 MPa is obtained in the area of the joint between the foot and the disc plates, while the maximum stresses on the rest of the blade are significantly less.

In view of the results, it is demonstrated that the change in geometry of the disc neither produces a greater level of stresses nor creates a concentration of stresses around the cooling holes. This configuration can therefore be maintained and with it the disc temperature can be reduced to values which enable the turbine to reach a useful life of ~60,000 hours.

6. REFERENCES

[1] High Temperature Reactor Components and Systems (HTR-E): *Preliminary Thermal — Structural Analysis of the HTR-E Turbine — Issue 1. July 2003.* Empresarios Agrupados Internacional, S.A., Document No. 092-110-E-M-00004

[2] Preliminary Analysis of HTR Power Turbine Rotor Cooling

[3] E-mail from VKI to EEAA (22 January 2004): *Pressure and Temperature Distribution for Helium Turbine*

APPENDIX A

TEMPERATURE DISTRIBUTION — FIGURES A-1 AND A-2

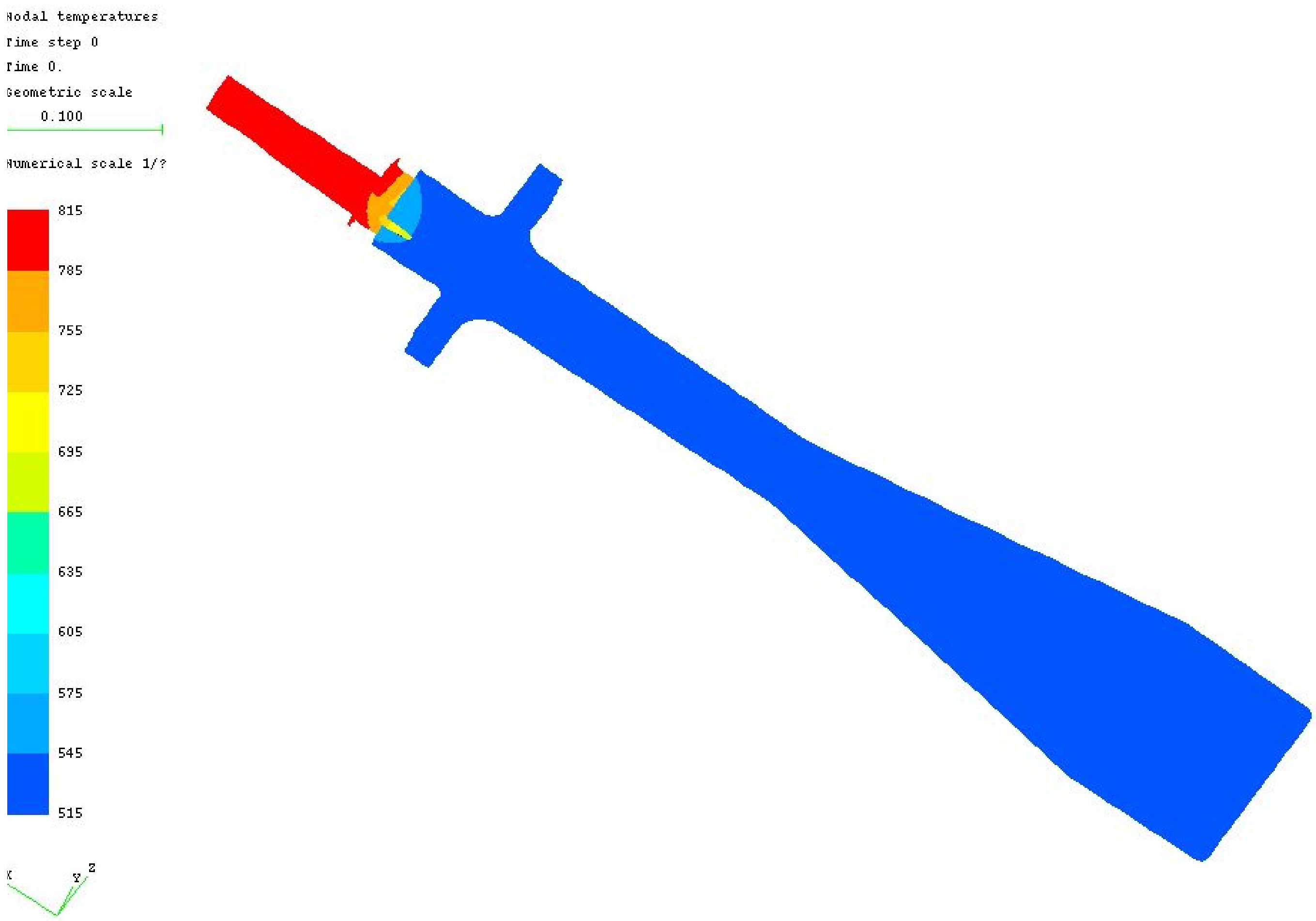

Figure A-1 Temperature Distribution in Blade + Disc Set

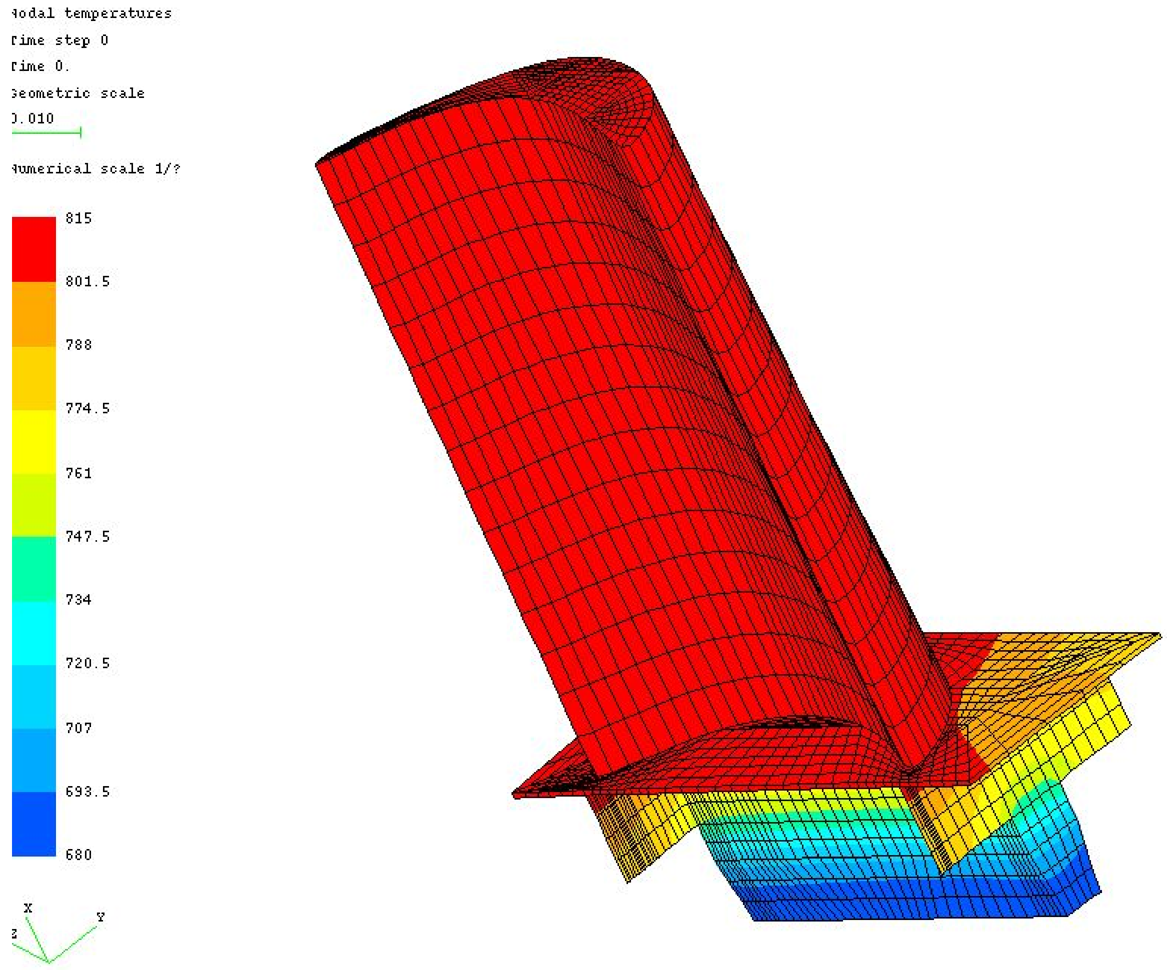

Figure A-2 Temperature Distribution on Blade

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APPENDIX B

STRESS DISTRIBUTION — FIGURES B-1 TO B-12

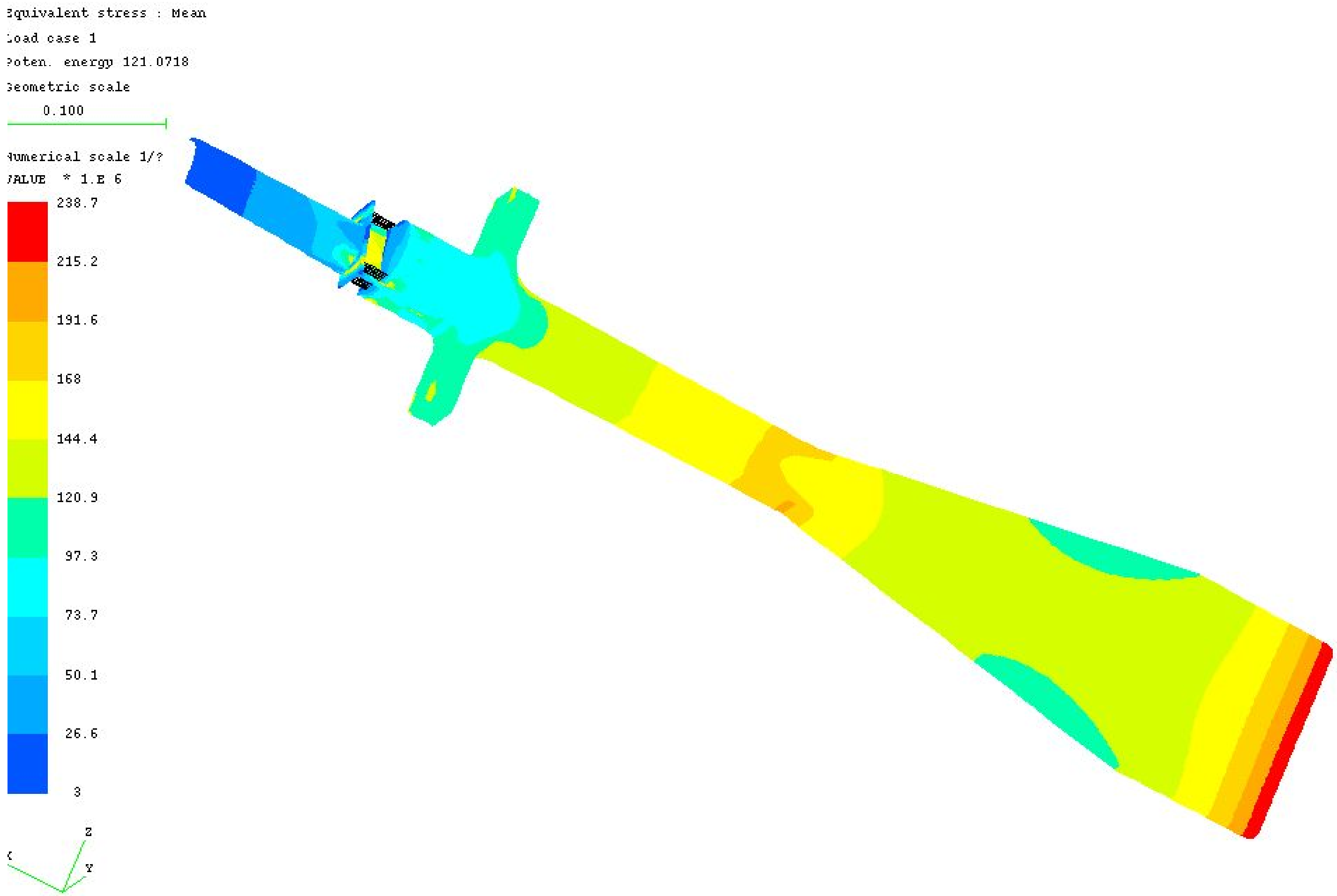

Figure B-1 Distribution of Stresses on Blade + Disc due to Rotation

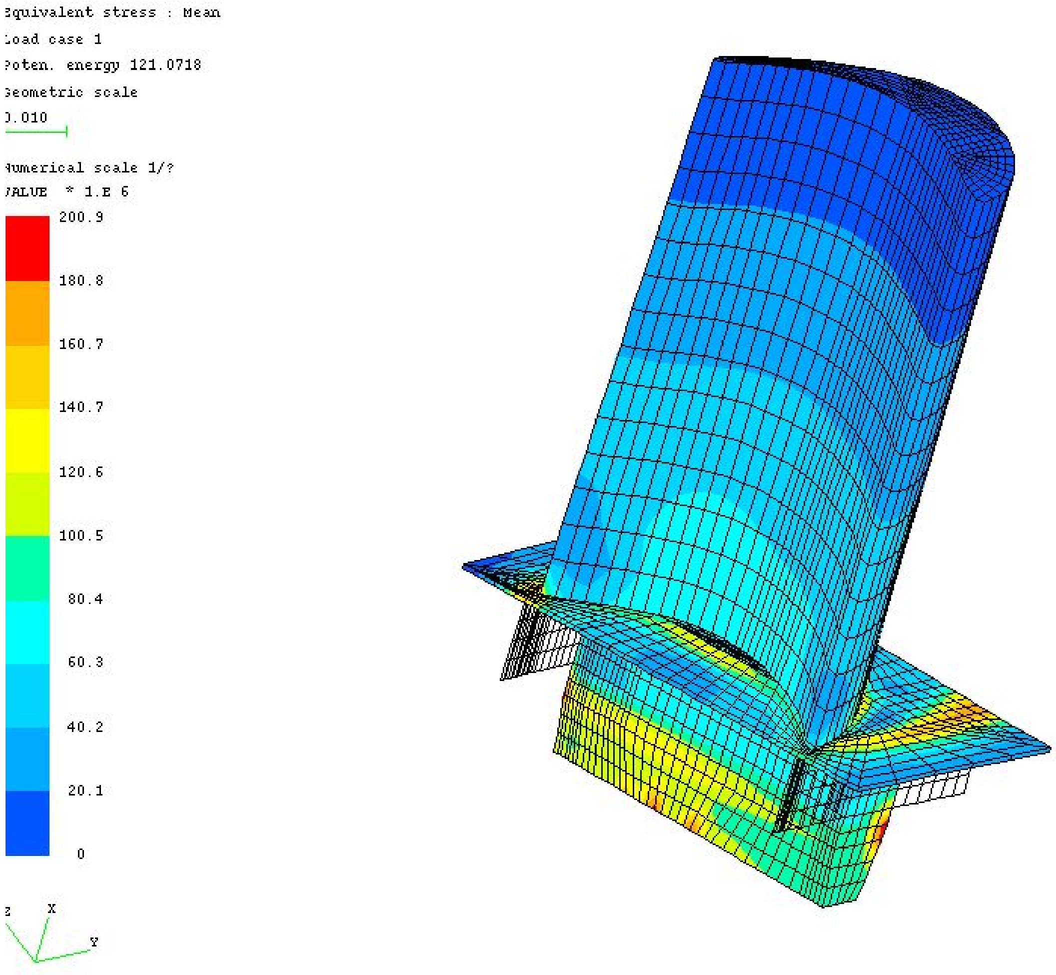

Figure B-2 Distribution of Stresses on Blade due to Rotation

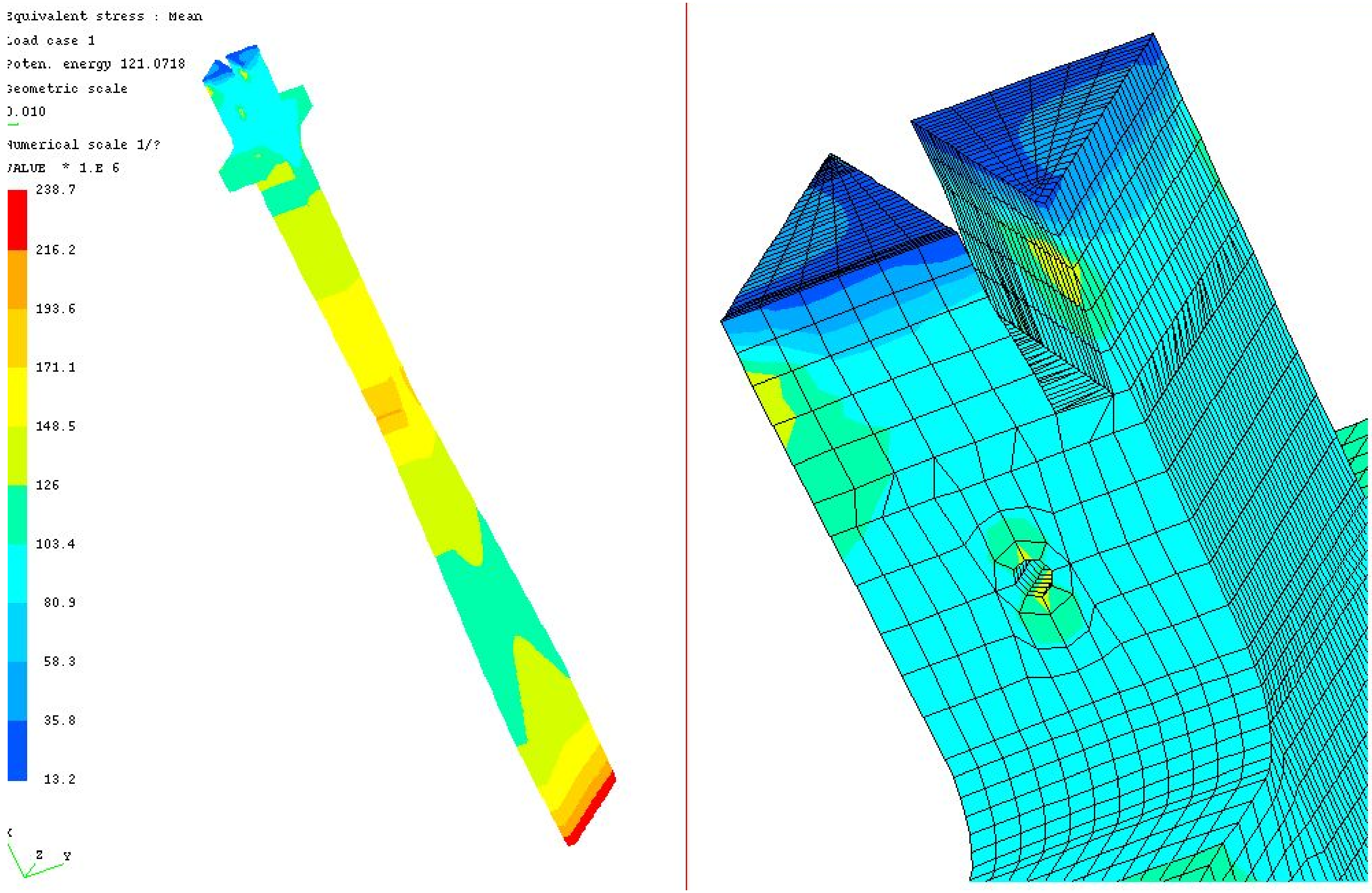

Figure B-3 Distribution of Stresses on Disc due to Rotation

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Figure B-4 Distribution of Stresses on Blade + Disc due to Pressure

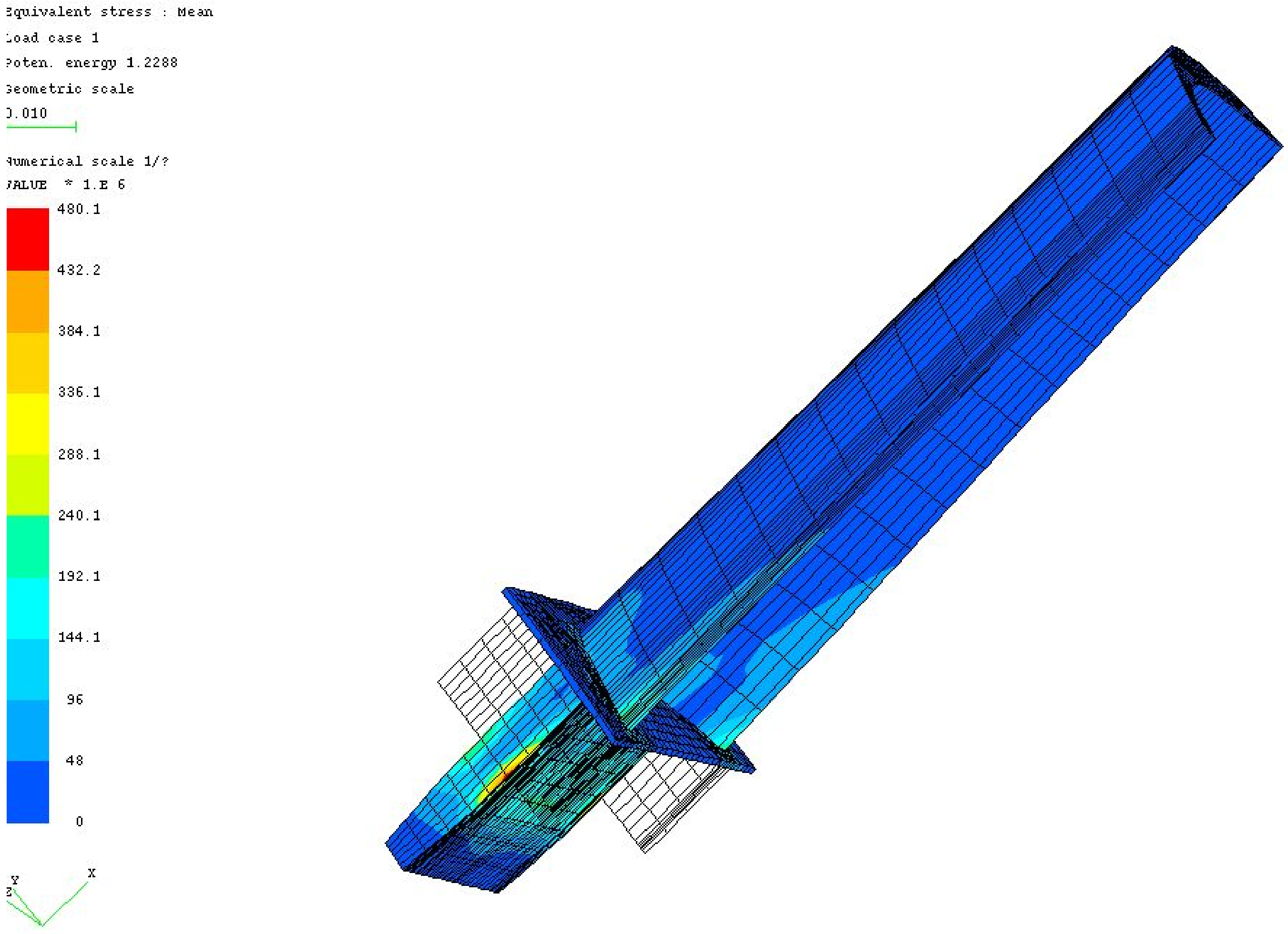

Figure B-5 Distribution of Stresses on Blade due to Pressure

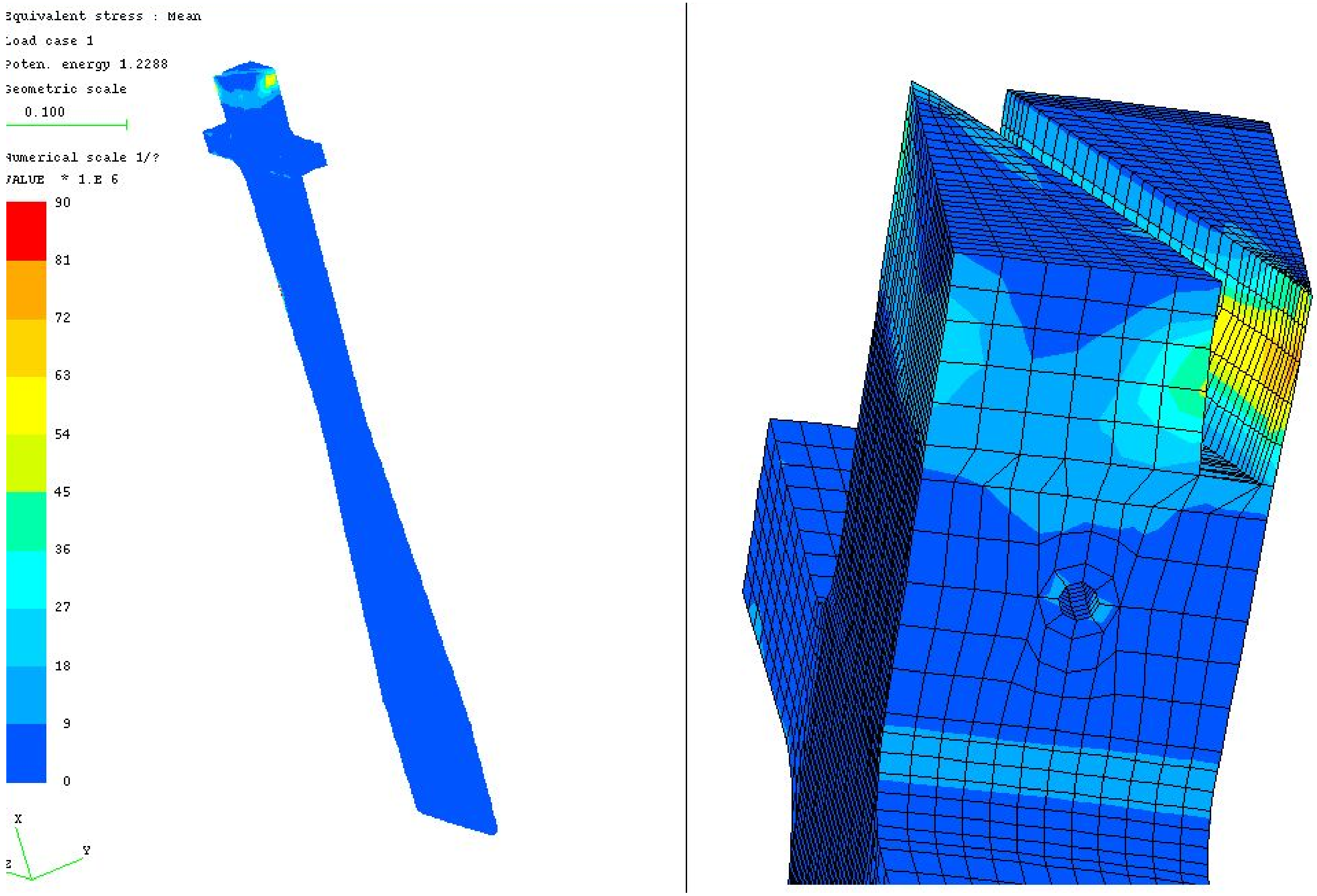

Figure B-6 Distribution of Stresses on Disc due to Pressure

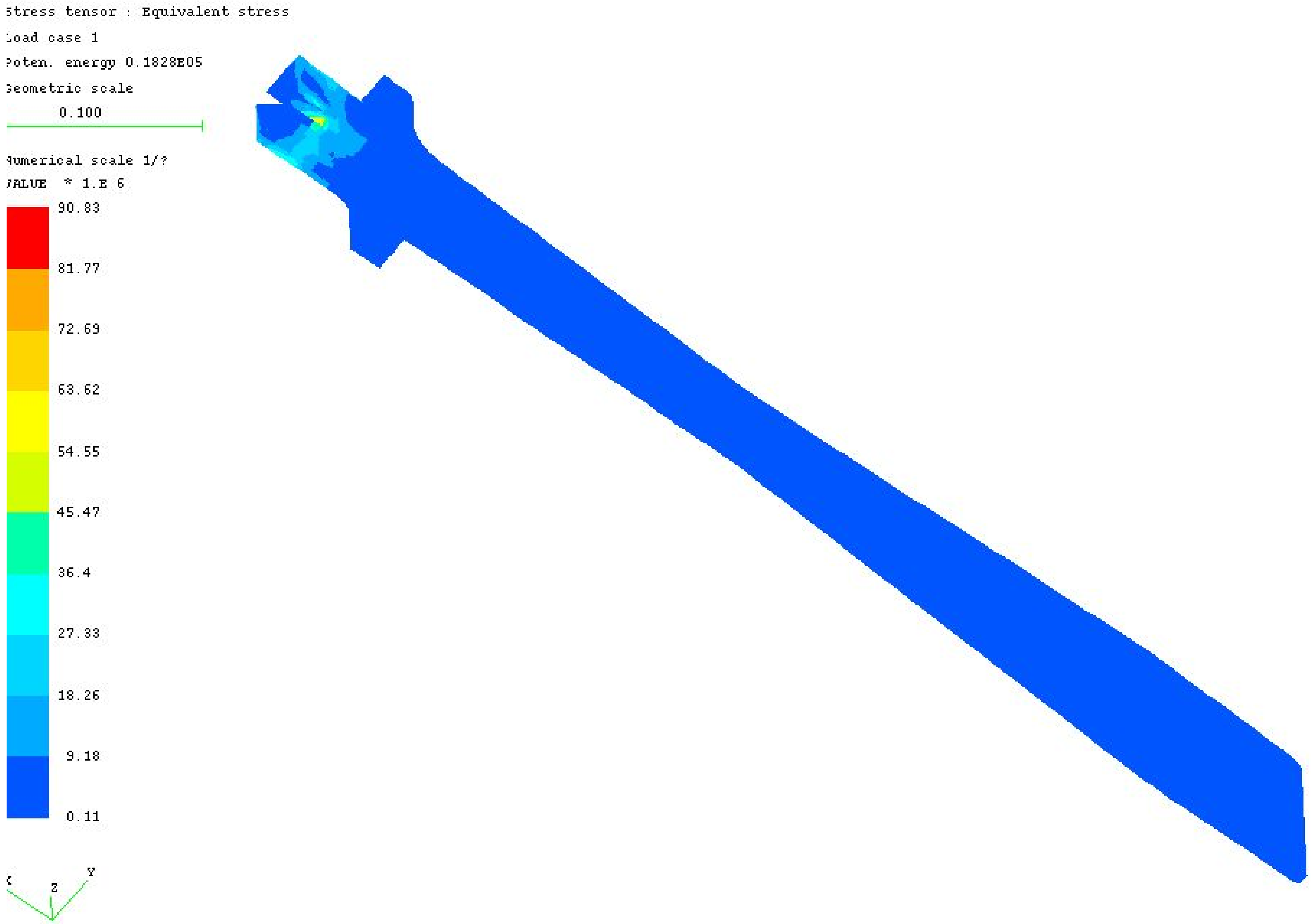

Figure B-7 Distribution of Stresses on Disc due to Temperature

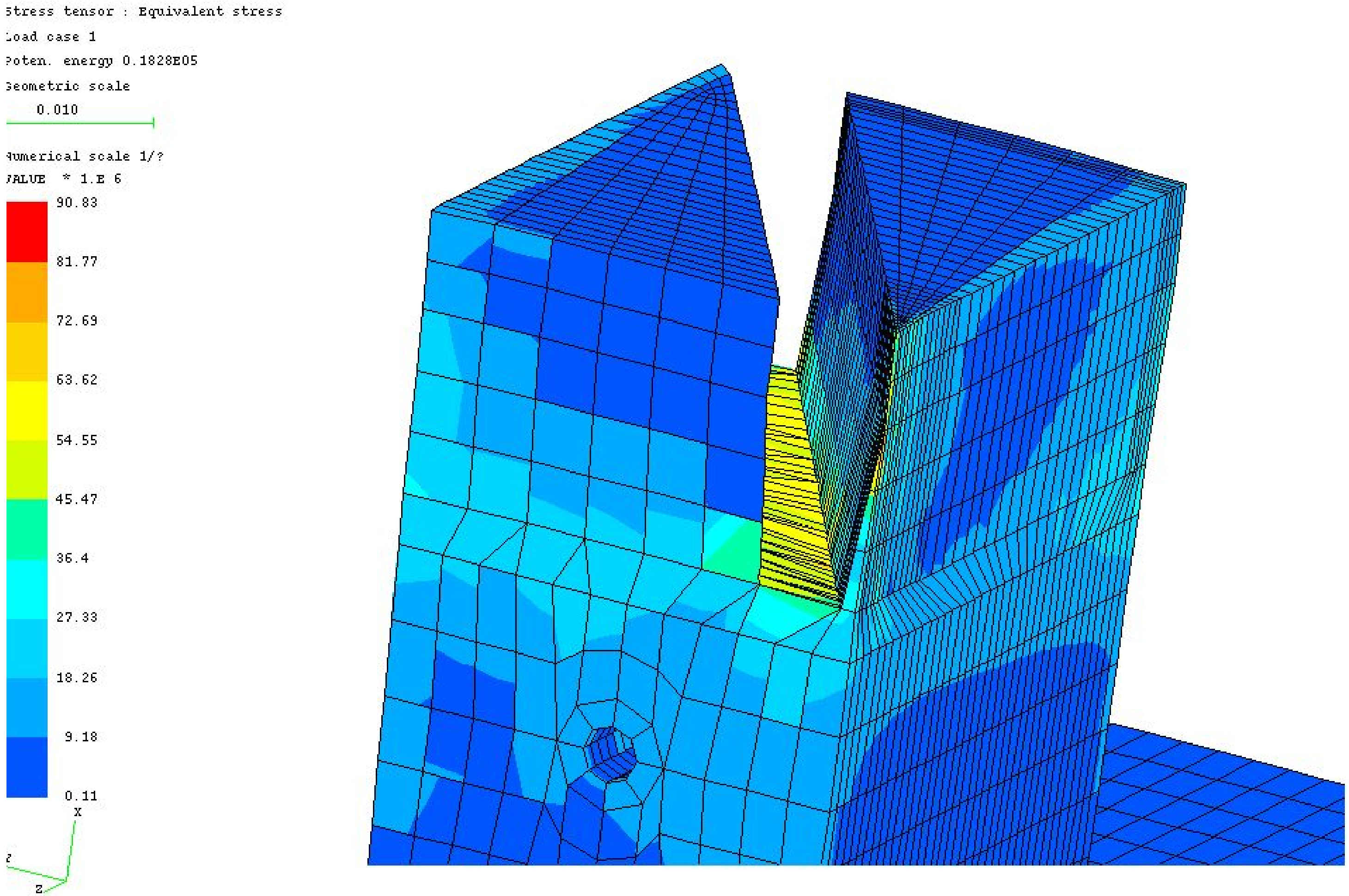

Figure B-8 Distribution of Stresses on Disc due to Temperature

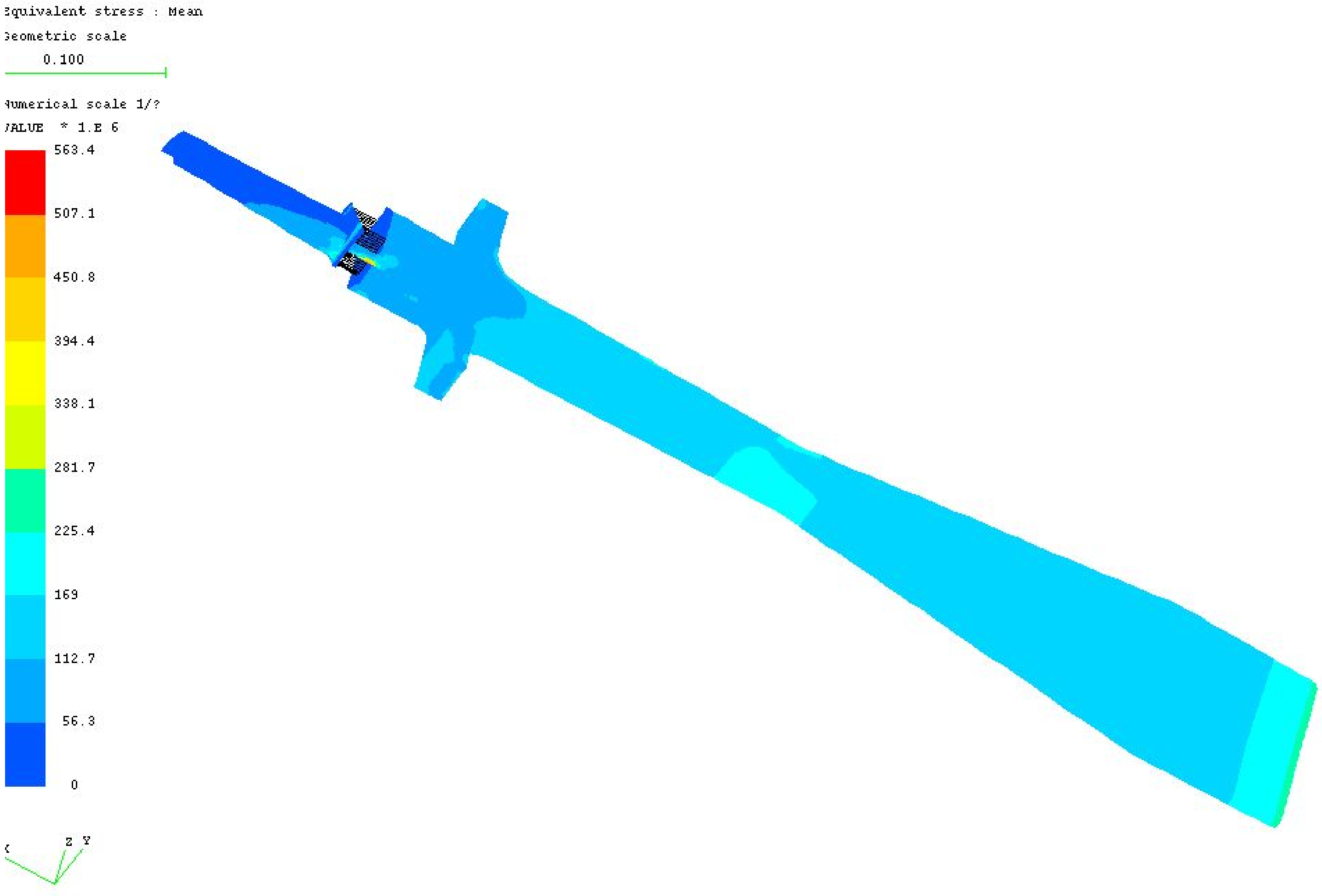

Figure B-9 Distribution of Stresses on Blade + Disc due to Load Combination

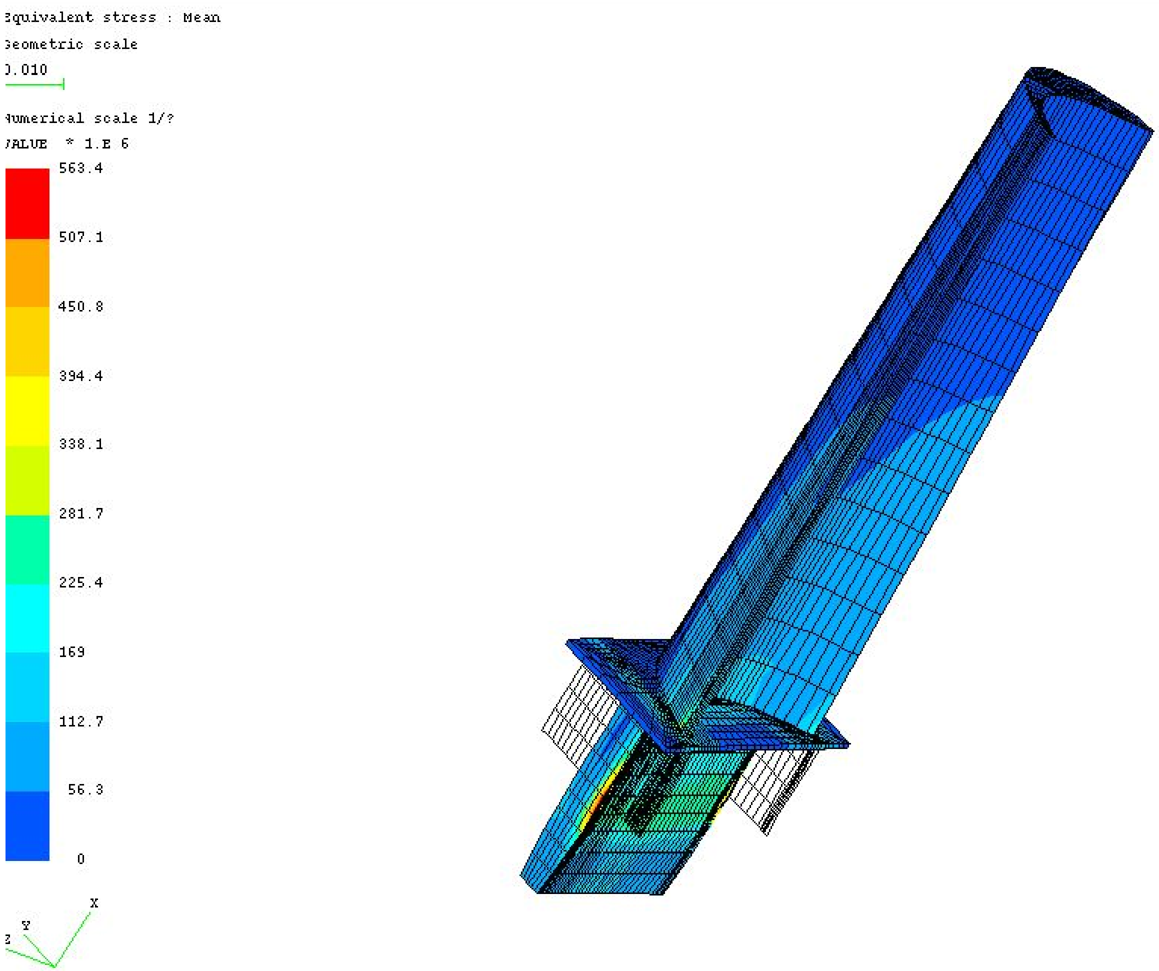

Figure B-10 Distribution of Stresses on Blade due to Load Combination

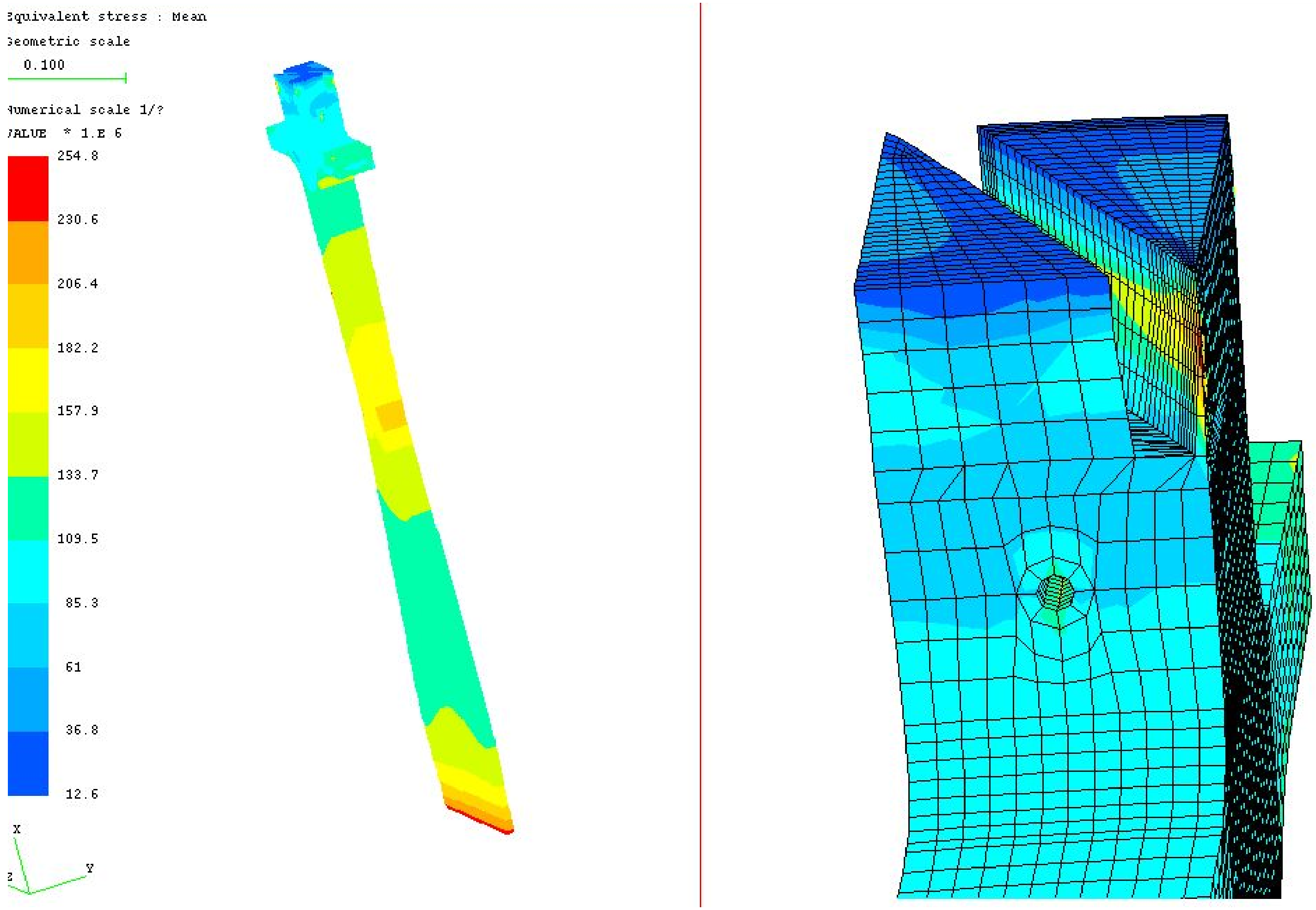

Figure B-11 Distribution of Stresses on Disc due to Load Combination

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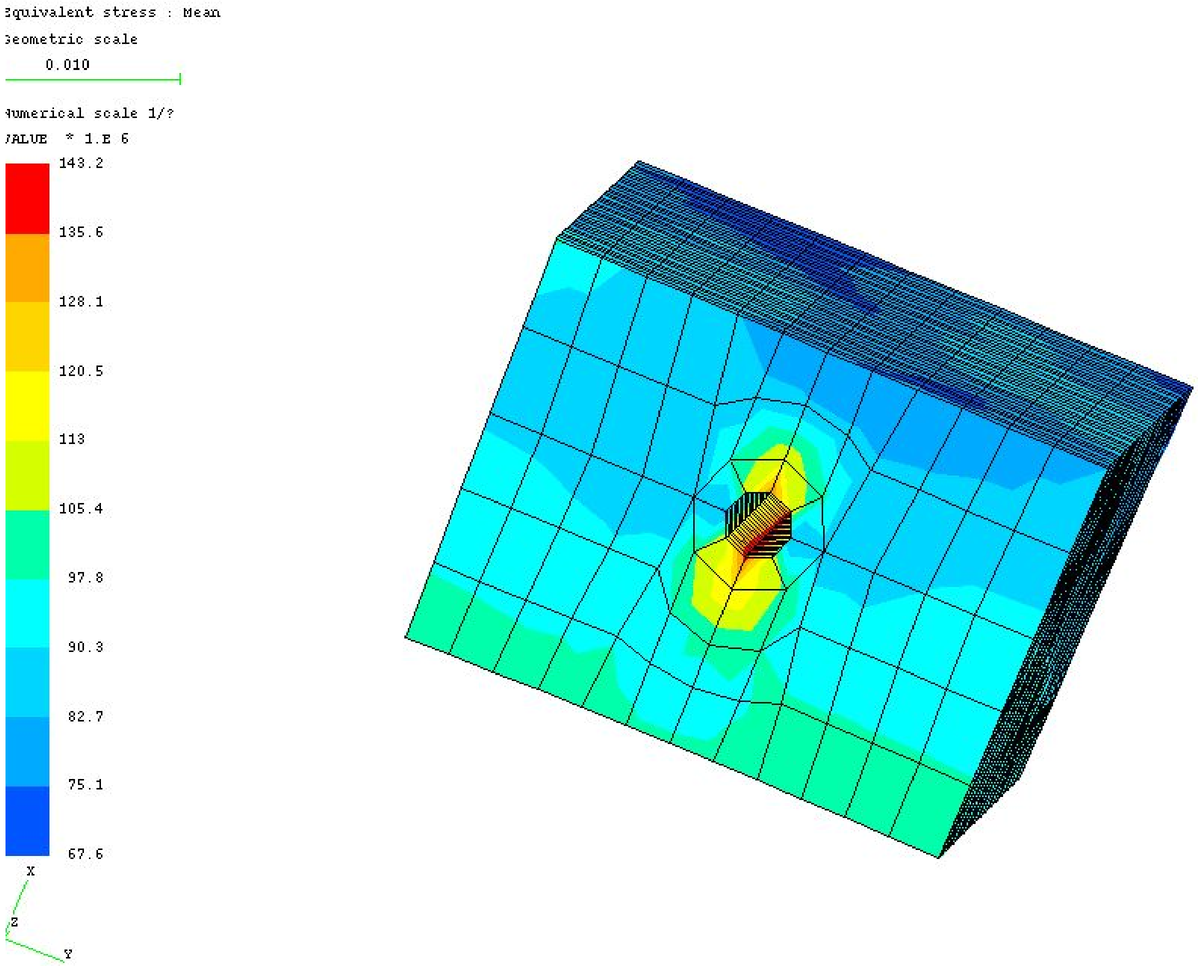

Figure B-12 Distribution of Stresses around Cooling Hole due to Load Combination