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**1.2.4.3 FE Calculations on the Dry Helium Rotating Seal**

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**Confidential**

# Stress Analyses of the Dry Helium Rotating Seal

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## Summary

By means of Finite Element calculations the stresses and strains in a dry rotating seal design have been determined. Two designs have been considered, i.e. a stationary graphite seal with a rotating steel counterpart and a rotating SiC seal with a stationary steel counterpart.

The seal is loaded by operating pressures, pressures in the seal gap, centrifugal forces, and thermal expansion. The seal gap pressures have been derived from a CFD study by NRG [1].

For the case of a stationary graphite seal the axial displacements of the seal face surface lie between 120 and 320 • m. The stresses in the seal ring vary from 102 MPa at one edge to 54 MPa in the rest of the ring.

For the case of a rotating SiC seal, the difference between thermal expansion of the seal and the steel supports leads to high stresses in the seal ring. By leaving a gap between the seal and the support high thermal stresses are prevented. Also deformation of the seal surface is prevented and therefore no significant changes in the seal surface are expected for this case. A maximum stress of 71 MPa is calculated in the seal ring with a radial gap between the seal and the support.

A rotating SiC seal with a radial gap between the seal and the seal support is preferable, because both the deformations of the seal surface and the seal stresses are limited.

# 1 Introduction

In the GT-MHR design for a High Temperature Reactor turbine and generator have separate housings. Seals around the main shaft should prevent that contaminated helium gas from the turbine housing flows into the generator housing. Two designs have been considered for this seal, i.e. a stationary graphite seal with a rotating steel counterpart and a rotating SiC seal with a stationary steel counterpart.

To determine whether the designs are acceptable and which design is preferable, Finite Element models have been set up to determine seal stresses and seal face deformations.

# 2 Problem Definition

On the seal face small chambers are manufactured in which pressure is built up due to the relative movement of the seal counterpart. The pressures on the seal face have been calculated previously in a separate NRG study [1].

Additionally the seal is loaded by operating pressures, pressures in the seal gap, centrifugal forces, and thermal expansion.

Considering the small dimensions of the pressure chamber on the seal face (depth 5-7 • m), the deformations of the seal face due to the forces acting on the seal may jeopardise the acceptable functioning of the seal. Additionally, whether the stresses in the seal remain within limits is unknown. Therefore Finite Element calculations have been undertaken to determine deformations, stresses, and strains in the considered dry rotating seal designs.

# 3 Graphite Seal Assumptions

## 3.1 Geometry

The geometry was based on design specifications set by the HTR-E WorkPackage 4 group [2]. This section considers the case in which the seal is manufactured of graphite and remains stationary in the rotor housing. The ring has an inner and outer diameter of 0.251 and 0.346 m respectively. As the seal is still in the design phase a number of less important dimensions have been arbitrarily chosen.

On the seal face 40 identical pressure chambers are manufactured. Each pressure chamber is accompanied by a small opening, which allows helium to flow into the pressure chamber. This opening is included in the model. The pressure chamber, however, is neglected because it is very small and has no influence on the structural behaviour of the seal.

### 3.2 Finite Element Model

Analyses have been performed with the general purpose Finite Element code MARC, version 2003. Due to symmetry, only a 9° segment (1/40 of the ring) needs to be modelled. The applied mesh for the stationary graphite seal with the seal support is shown in figure 1.

MARC element type 7, a linear cubic solid element, was used for the calculations. The model of the stationary graphite seal consists of 13802 elements and 15740 nodes. The seal is supported by a steel structure. The steel supports have been approximated in the model and between seal and support contact has been defined.

### 3.3 Material Properties

Material properties implemented are summarised in table 1.

table 1 Material properties implemented for graphite and steel

| | E-modulus [Pa] | Poisson ratio [-] | Density [kg/m3] | Thermal Expansion [1/K] |
|---|---|---|---|---|
| Steel | $2.0 \cdot 10^{11}$ | 0.3 | 8000 | $1.7 \cdot 10^{-5}$ |
| Graphite | $3.5 \cdot 10^{10}$ | 0.3 | 1750 | $6.4 \cdot 10^{-6}$ |

### 3.4 Initial Conditions

For the stationary graphite seal a stress-free initial state at 150°C has been assumed, which is the situation during manufacturing. Stresses will be considered at 20°C.

### 3.5 Pressures

The pressure boundary conditions on the seal face have been applied according to a CFD study on the seal previously performed by NRG [1]. The pressures applied on the seal face are illustrated in figure 4.

The seal separates the high-pressure in the turbine housing from the low-pressure at the outside. Therefore the outside of the ring is exposed to a pressure of 20 bar, while the inside of the ring is exposed to a pressure of 1 bar. These pressures have been applied to surfaces where no contact with the support structure exists.

Pressures have been implemented through the FORCEM user subroutine.

### 3.6 Temperatures

The temperature of the seal and the support will be approximately 150°C during assembly. The temperature increase or decrease and the effects of the differences in thermal expansion on stresses and strains have been included in the model.

# 4 Sic Seal Assumptions

## 4.1 Geometry

The geometry was based on design specifications set by the HTR-E WorkPackage 4 group [2]. This section considers the case of a SiC seal, which is mounted on the rotor, and is therefore rotating with the rotor angular velocity. The ring has an inner and outer diameter of 0.251 and 0.346 m respectively. As the seal is still in the design phase a number of less important dimensions have been arbitrarily chosen.

On the seal face 40 identical pressure chambers are manufactured. Each pressure chamber is accompanied by a small opening, which allows helium to flow into the pressure chamber. This opening is included in the model. The pressure chamber, however, is neglected because it is very small and has no influence on the structural behaviour of the seal.

## 4.2 Finite Element Model

Analyses have been performed with the general-purpose Finite Element code MARC, version 2003. Due to symmetry, only a 9° segment (1/40 of the ring) needs to be modelled. The applied meshes for the rotating SiC seal with support, with and without a gap between seal and support are shown in figures 2 and 3 respectively.

MARC element type 7, a linear cubic solid element, was used for the calculations. The model of the rotating SiC seal consists of 3454 elements and 4246 nodes. The seal is supported by the steel rotor shaft. The steel supports have been approximated in the model and between seal and rotor shaft support contact has been defined.

## 4.3 Material Properties

Material properties implemented are summarised in table 2.

table 2 Material properties implemented for steel and SiC

| | E-modulus [Pa] | Poisson ratio [-] | Density [kg/m3] | Thermal Expansion [1/K] |
|---|---|---|---|---|
| Steel | $2.0 \cdot 10^{11}$ | 0.3 | 8000 | $1.7 \cdot 10^{-5}$ |
| SiC | $3.8 \cdot 10^{11}$ | 0.3 | 8000 | $3.9 \cdot 10^{-6}$ |

## 4.4 Initial Conditions

A stress-free initial state has been assumed at 20°C. Stresses will be considered under normal operating conditions, i.e. at 150°C.

### 4.5 Pressures

The pressure boundary conditions on the seal face have been applied according to a CFD study on the seal previously performed by NRG [1]. The pressures applied to the seal face are illustrated in figure 4.

The seal separates the high-pressure in the turbine housing from the low-pressure at the outside. Therefore the outside of the ring is exposed to a pressure of 20 bar, while the inside of the ring is exposed to 1 bar. These pressures have been applied to surfaces where no contact with the support structure exists.

Pressures have been implemented through the FORCEM user subroutine.

### 4.6 Centrifugal Forces

In this case the seal is rotating and centrifugal forces will lead to internal stresses and strains in the seal. A rotational velocity • of 3000 rpm has been assumed, leading to a centrifugal force for material with a density • of:

$$F_{cent} = \frac{m \cdot v^2}{R} = \frac{\rho \cdot V \cdot (R \cdot \omega)^2}{R} = \rho \cdot V \cdot \omega^2 \cdot R$$

The centrifugal force has been implemented as a body force via subroutine FORCEM, where for each element, depending on the element location (*R*) and the element volume (*V*), the centrifugal force on that element is calculated.

### 4.7 Temperatures

The temperature of the seal and support will be approximately 150°C. The temperature increases or decreases and the effects of the differences in thermal expansion on stresses and strains have been included in the model.

## 5 Results

The deformed shape and the stress distribution for the stationary graphite seal with rotating steel counterpart are shown in figure 5 and figure 6 respectively. The boundary conditions applied lead to an axial displacement of the seal face varying from 120 to 320 • m. Axial displacements of the stationary graphite seal are illustrated in figure 7.

For the rotating SiC seal, differences in thermal expansion between SiC and steel lead to high local stresses. The deformed shape and the equivalent Von Mises stresses are shown in figure 8 and figure 9 respectively. The axial displacement of the rotating SiC seal is shown in figure 10.

A gap between the inner seal surface and the steel support structure was shown to prevent the high local stresses in the rotating SiC seal. Due to this gap, maximum stresses in the seal are reduced to approximately 71 MPa. For this case the deformed shape and the equivalent Von Mises stresses are shown in figure 11 and figure 12 respectively. The axial displacement of the seal face varies between 80 and 81 • m. The axial displacement of the rotating SiC seal with a radial gap between seal and support is shown in figure 13.

The results are summarised in table 3.

table 3 Results summary

| **Case description** | **Maximum stress [MPa]** | | **Seal face displacement [• m]** |
|---|---|---|---|
| | **Edge** | **Overall** | |
| Stationary graphite seal | 102 | 54 | 120-320 |
| Rotating SiC seal, no gap | 1165 | 563 | 78-93 |
| Rotating SiC seal, with gap | 71 | 71 | 80-81 |

# 6 Conclusions

By means of Finite Element calculations the stresses and strains in a dry rotating seal design have been determined. Two designs have been considered, i.e. a stationary graphite seal with a rotating steel counterpart and a rotating SiC seal with a stationary steel counterpart.

The seal is loaded by operating pressures, pressures in the seal gap, centrifugal forces, and thermal expansion. The seal gap pressures have been derived from a CFD study by NRG [1].

The results are summarised in table 4.

table 4 Concluding results

| **Case description** | **Maximum stress [MPa]** | | **Axial displacement [• m]** |
|---|---|---|---|
| | **Edge** | **Overall** | |
| Stationary graphite seal | 102 | 54 | 120-320 |
| Rotating SiC seal, no gap | 1165 | 563 | 78-93 |
| Rotating SiC seal, with gap | 71 | 71 | 80-81 |

For the case of a stationary graphite seal, the axial displacements of the seal face surface lie between 120 and 320 • m. This is unacceptable considering the pressure chamber depth of 5 to 7 • m. The stresses in the seal ring vary from 102 MPa at one edge to 54 MPa in the rest of the ring.

For the case of a rotating SiC seal, the difference between thermal expansion of the seal and the steel supports leads to high stresses locally in the seal ring. By leaving a gap between the seal and the support high stresses are prevented. Additionally, a gap prevents deformation of the seal surface and therefore no significant changes in the seal surface are expected for this case. A maximum stress of 71 MPa is calculated in the seal ring with a radial gap between the seal and the support.

A rotating SiC seal with a radial gap between the seal and the seal support is preferable, because both the deformations of the seal surface and the seal stresses are limited.

## References


[1] N.B. Siccama, “CFD Calculations on the Dry Helium Rotating Seal”, NRG-report 20793/03.54363/C (2003).

[2] P. Parent, “HTRE-WP4, Helium Rotating Seal, Dimensional Data Proposition for NRG Calculations”, Framatome report 03 CER 038 C (2003).

# Figures

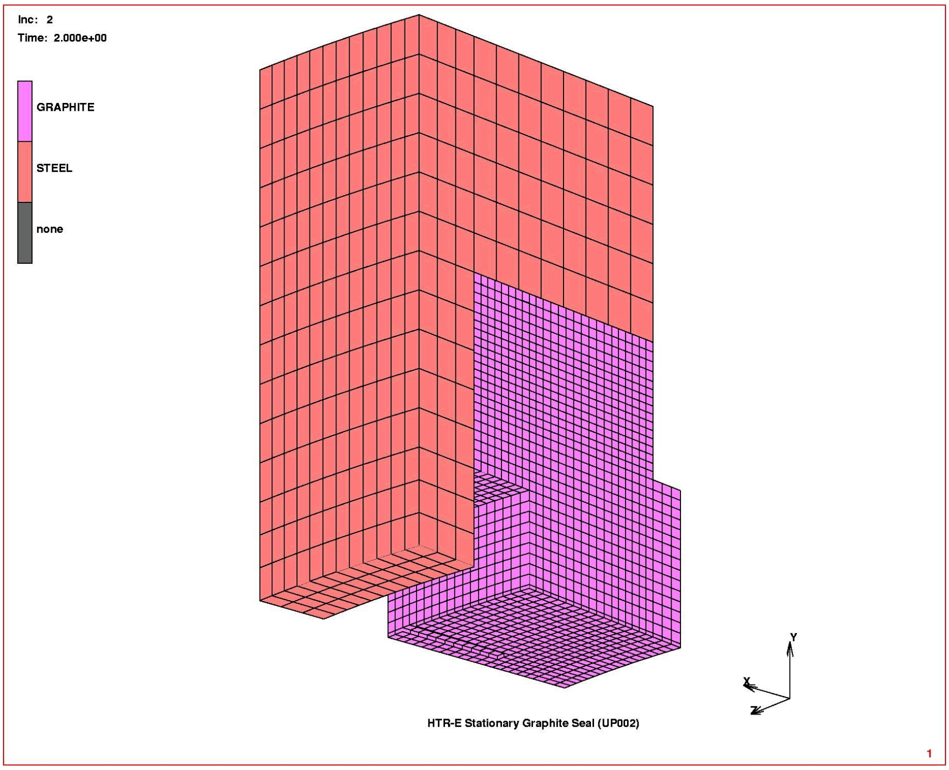


figure 1 Mesh of stationary graphite seal model

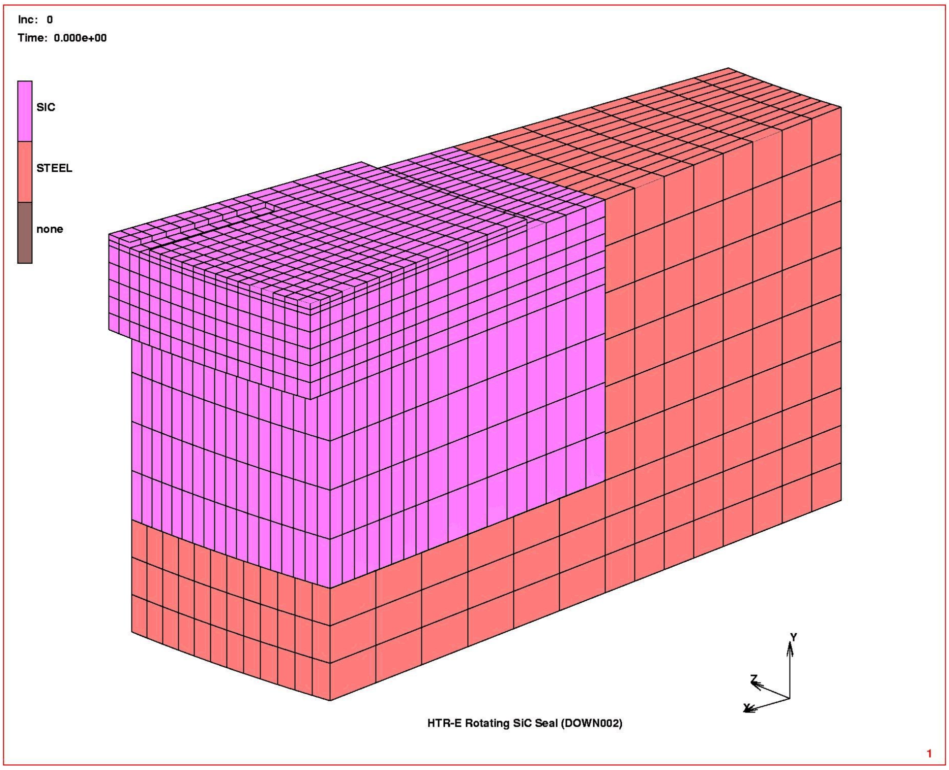


figure 2 Mesh of the rotating SiC seal model

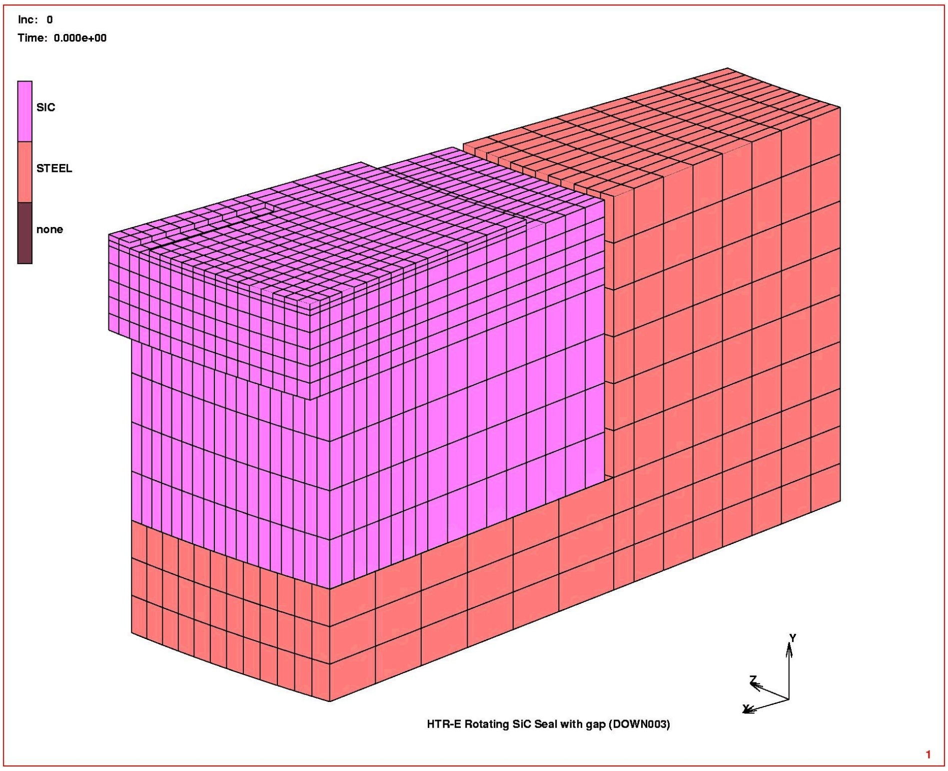


figure 3 Mesh of the rotating SiC seal model, with assumed gap between seal and support

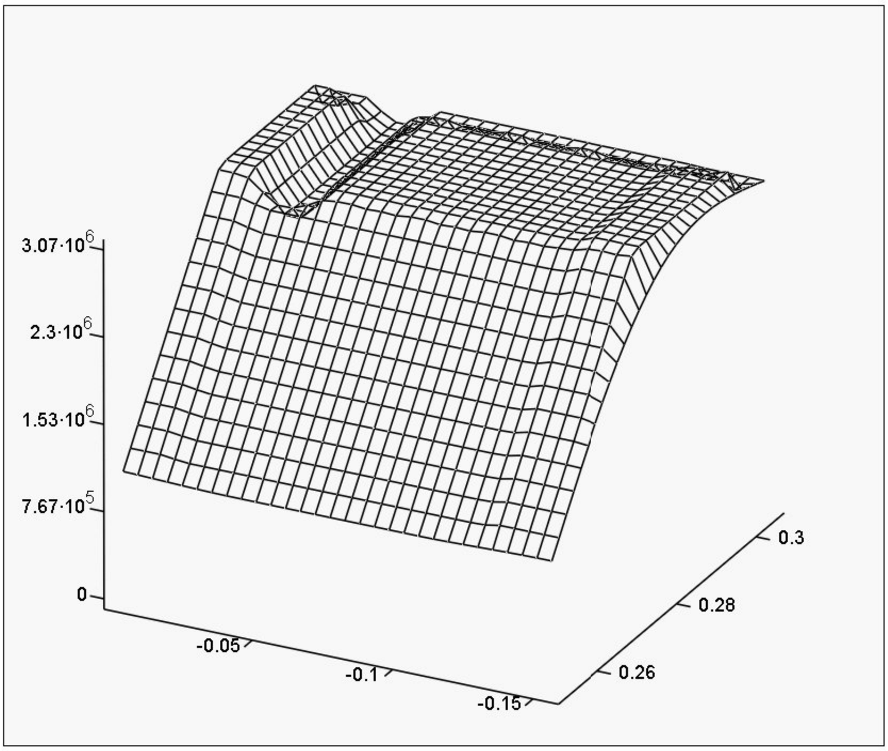


figure 4 Pressure distribution [Pa] implemented on the seal face based on [1] (R-• co-ordinates)

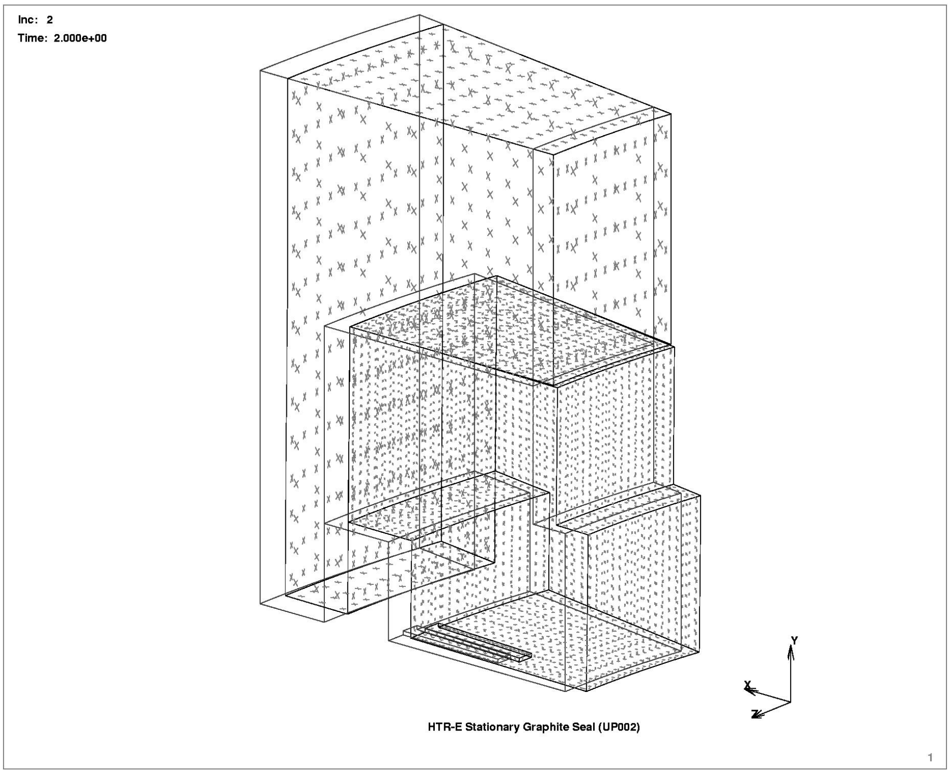


figure 5 Deformed shape of the stationary graphite seal model

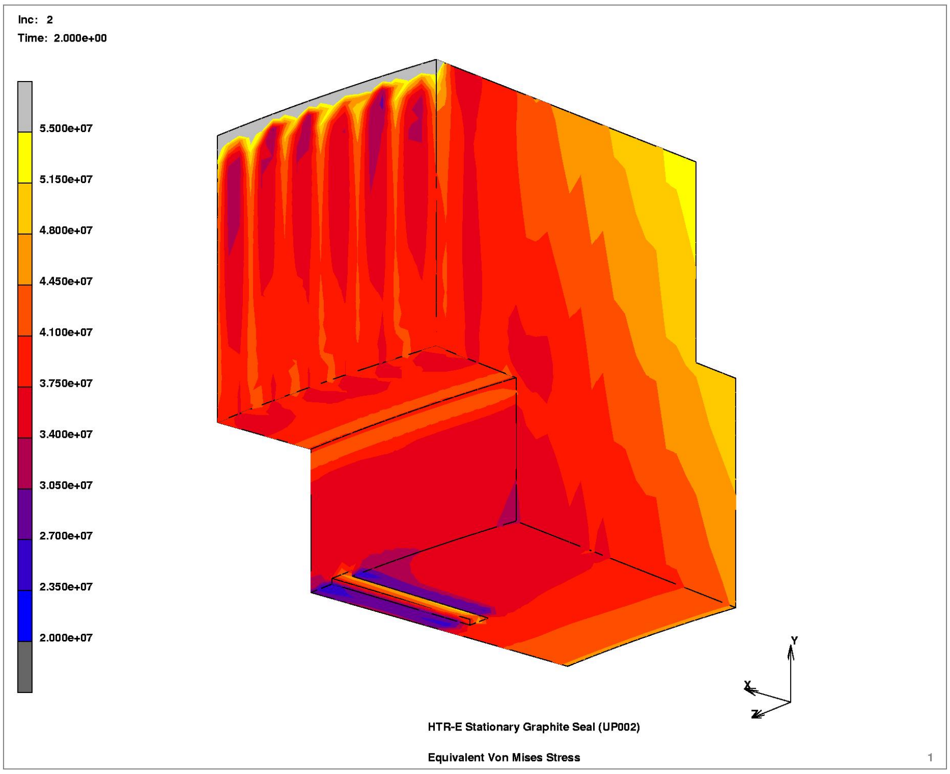


figure 6 Equivalent Von Mises stresses in Pa in the stationary graphite seal

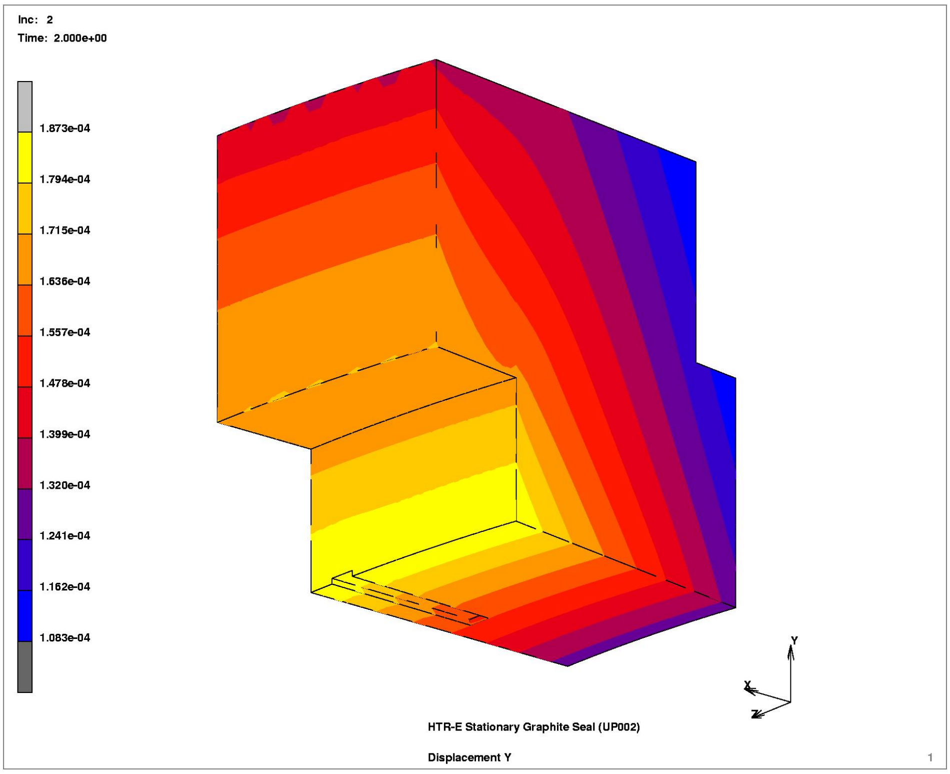


figure 7 Axial displacement in m of the stationary graphite seal

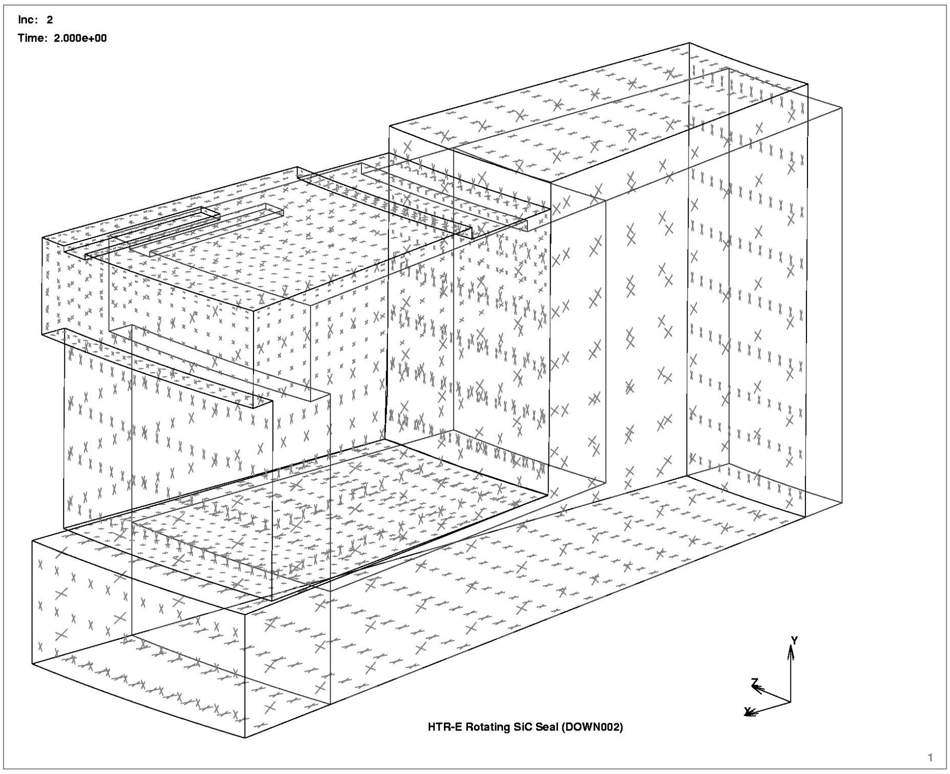


figure 8 Deformed shape of the rotating SiC seal model, no gap

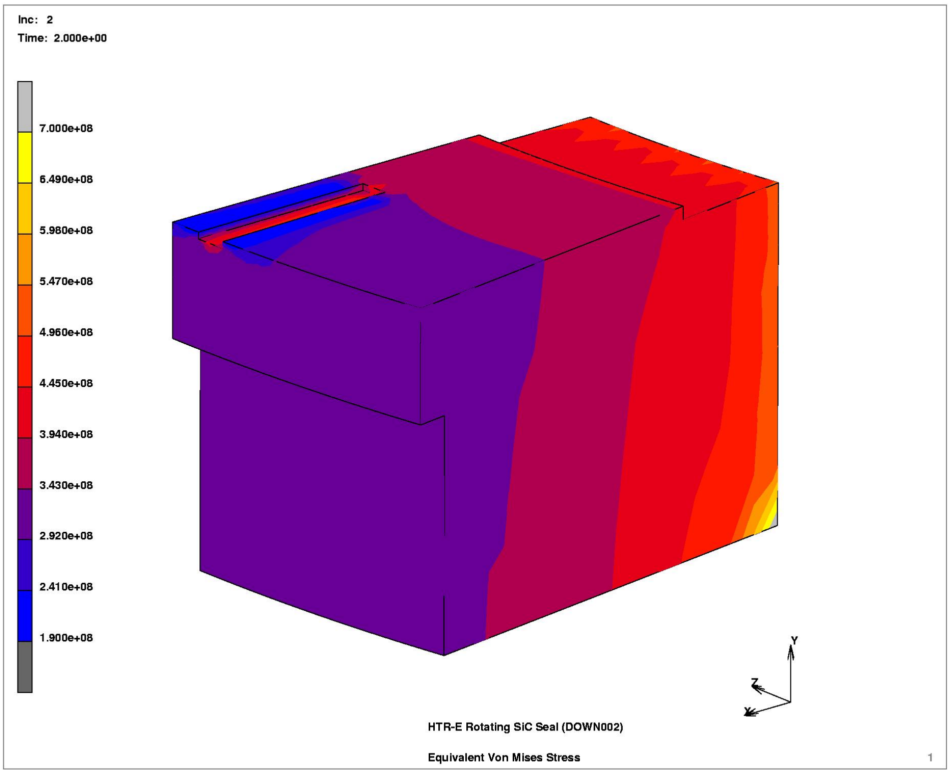


figure 9 Equivalent Von Mises stresses in Pa in the rotating SiC seal, no gap

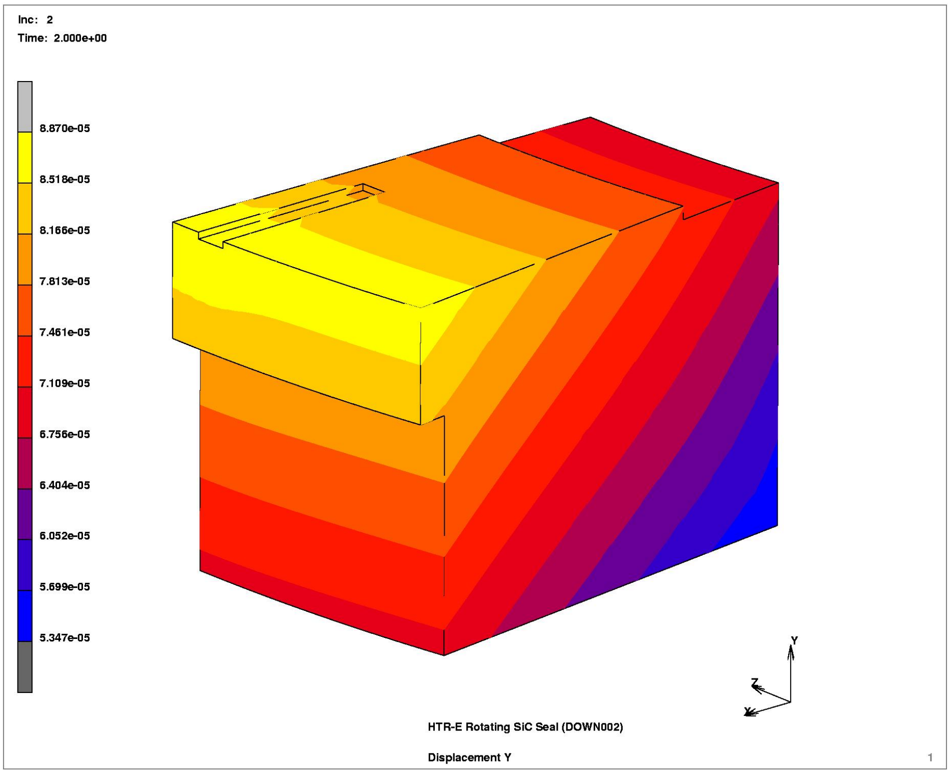


figure 10 Axial displacement in m of the rotating SiC seal, no gap

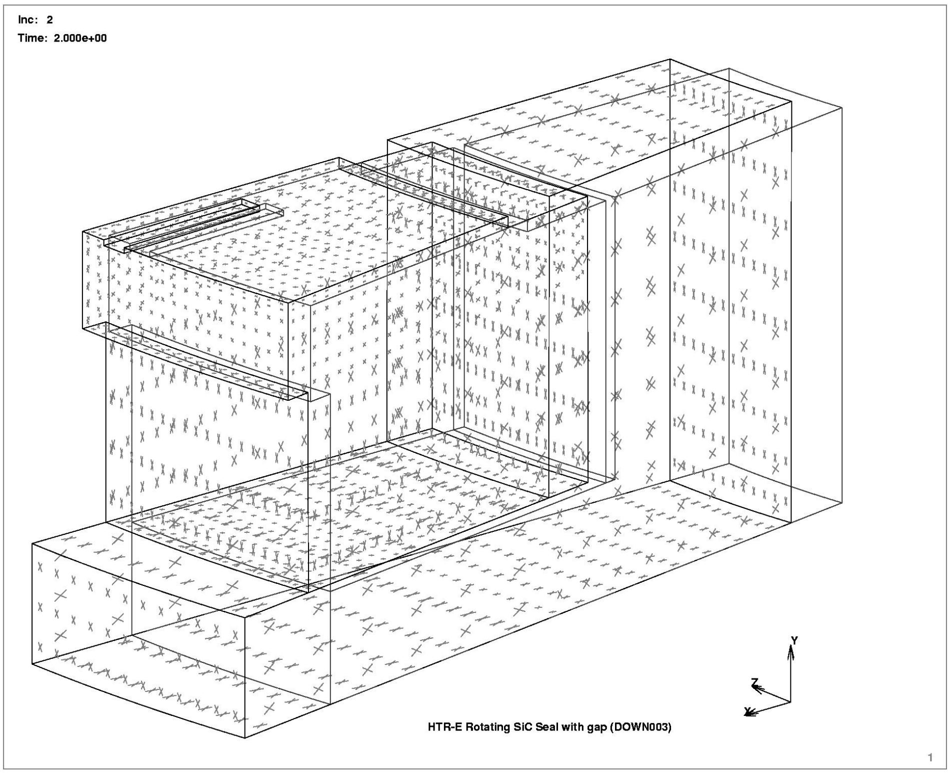


figure 11 Deformed shape of the rotating SiC seal model, with gap

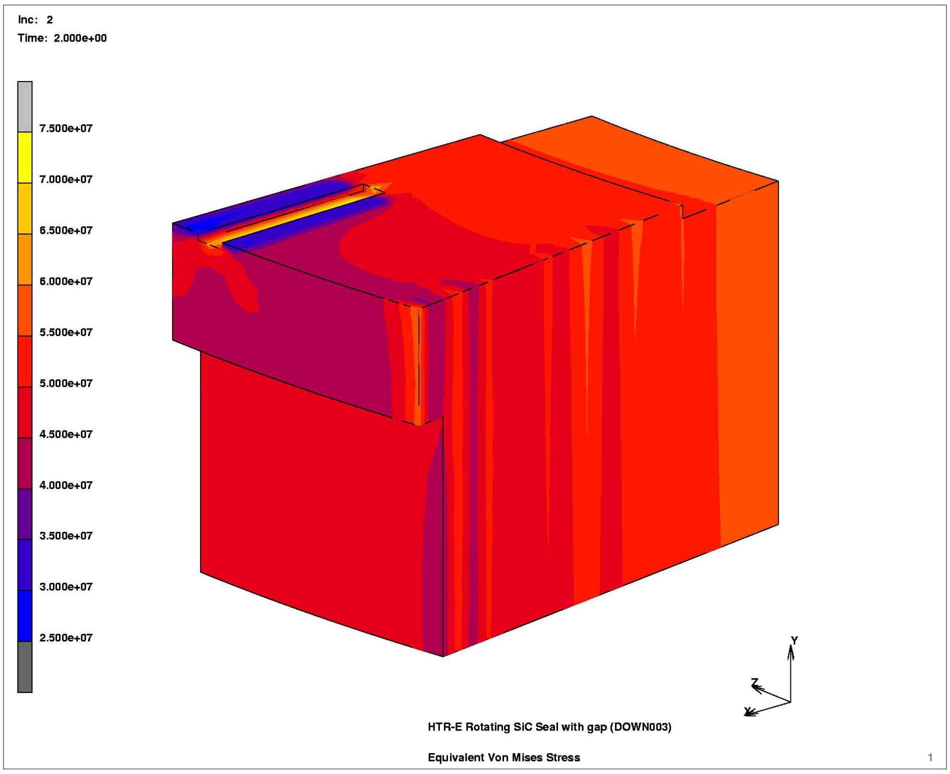


figure 12 Equivalent Von Mises stresses in Pa in the rotating SiC seal, with gap

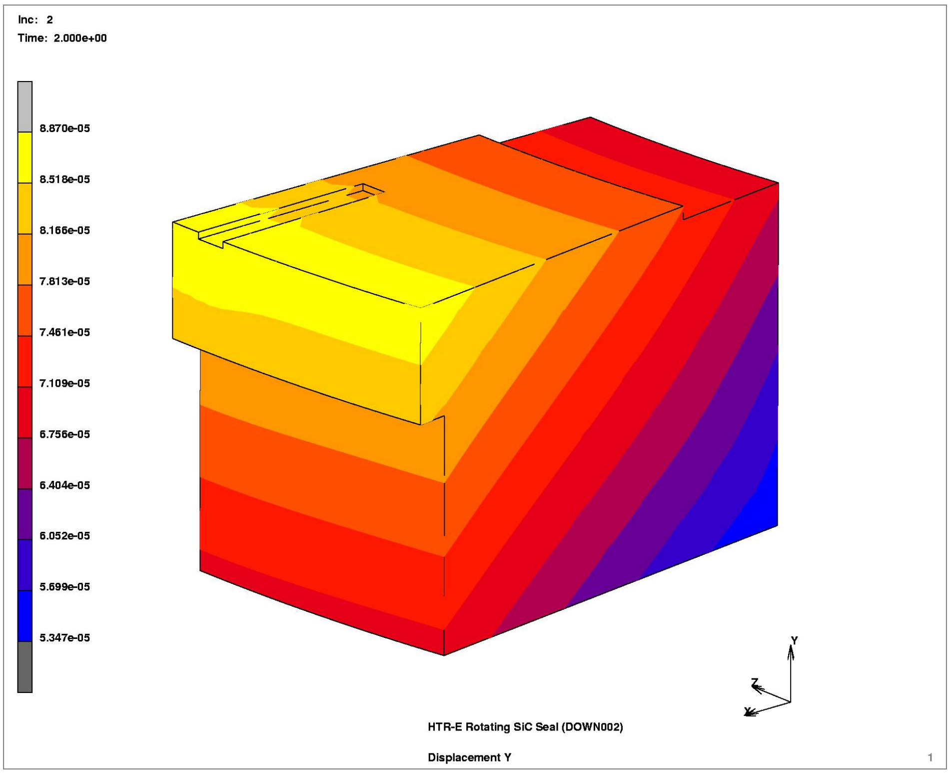


figure 13 Axial displacement in m of the rotating SiC seal, with gap