Direction de la recherche technologique
Département des technologies pour les énergies nouvelles
Service Matériaux et Procédés
Laboratoire Simulation et Mécanique des Matériaux

**Rapport technique DTEN/DL/2003/051**

# HTR-E - WP1
---
# Review of existing technologies from past HTR experiences and other industries

***Deliverable D3a***

*Auteur :* L. Briottet

| | |
|---|---|
| **Référence du contrat** | **E-OTP n°A-RCGCH-03-04 LOC7D8 680302** |
| **Référence PRODEM** | 01-00302 |
| **Nature du rapport** | Rapport intermédiaire : Délivrable D3a |
| **Confidentialité** | Les informations contenues dans le présent document sont la propriété du CEA. Il ne peut être reproduit ou transmis à des tiers sans l'autorisation expresse du CEA. |

| | **Rédacteur** | **Vérificateur** | **Approbateur** |
|---|---|---|---|
| **Nom** | Briottet | Burlet | Le Naour |
| **Fonction** | Ingénieur | Chef de Laboratoire | Chef de Service |
| **Signature** | | | P.I. |
| **Date** | 17/07/2003 | 17/07/03 | 18/07/03 |

**EUROPEAN COMMISSION**
**5th EURATOM FRAMEWORK PROGRAMME 1998-2002**
**KEY ACTION : NUCLEAR FISSION**

# HTR-E

## High-Temperature Reactor Components and Systems

**CONTRACT N°**
**FIKI-CT-2001-00177**

**WP1**

***Review of existing technologies from past HTR experiences and other industries***

***Deliverable n°3***

Laurent Briottet

CEA / Grenoble

France

Dissemination level : RE
Document N°: HTR-E 03/07 D-1-1-2-1
Status : Preliminary
Deliverable n°3

# Modifications

| Revision | Date | Modifications |
|---|---|---|
| A | July 2003 | First issue |
| | | |
| | | |
| | | |
| | | |
| | | |
| | | |
| | | |
| | | |
| | | |
| | | |
| | | |
| | | |
| | | |
| | | |
| | | |
| | | |
| | | |
| | | |
| | | |
| | | |

## LISTE DE DIFFUSION

Rapport complet à :

| | | |
|---|---|---|
| C.E.E. | Serveur SINTER | (electronic file) |
| DOB | M. SCHUSTER | 1 ex. |
| DTEN/DIR | M. DANROC / M. ROUZAUD | 1 ex. |
| | M. BACCINO | 1 ex. |
| DEN/DM2S/SEMT/LM2S | Mme NICOLAS | 1 ex. |
| DEN/DTP/SETEX | M. SAEZ | 1 ex. |
| DTEN/SERVICE | M. LE NAOUR | 1 ex. |
| SERVICE/LABORATOIRE | Mme BURLET | 1 ex. |
| | M. BRIOTTET | 1 ex. |
| | M. COUTURIER | 1 ex. |
| | Archivage | Original + 1 ex. |

Page de garde + résumé + Liste de diffusion à :

Chef du DTEN/SCSE
Chef du DTEN/SMP/LESA
Chef du DTEN/SMP/LPMP
Chef du DTEN/SMP/LPTS

Page de garde + résumé + Liste de diffusion + Bordereau d’envoi à :

Ingénieur Qualité
Bureau financier (Ventes)

# Abstract

Within the HTR-E project, the design of a helium turbomachine in the direct cycle is analysed in the work package 1. The objective of this document is to review the technological choices proposed in past experiences in HTR or other industries concerning gas turbines. This report is mainly devoted to three particular aspects : materials for turbine discs and blades, cooling systems options and deblading accidents. From this bibliographical analysis, some recommendations are drawn for a future High Temperature Reactor. It is shown that even with the latest developments in materials, it will be necessary to cool the discs. Moreover, for safety reasons in case of deblading, a horizontal shaft aligned with the reactor vessel is recommended for the turbomachine.

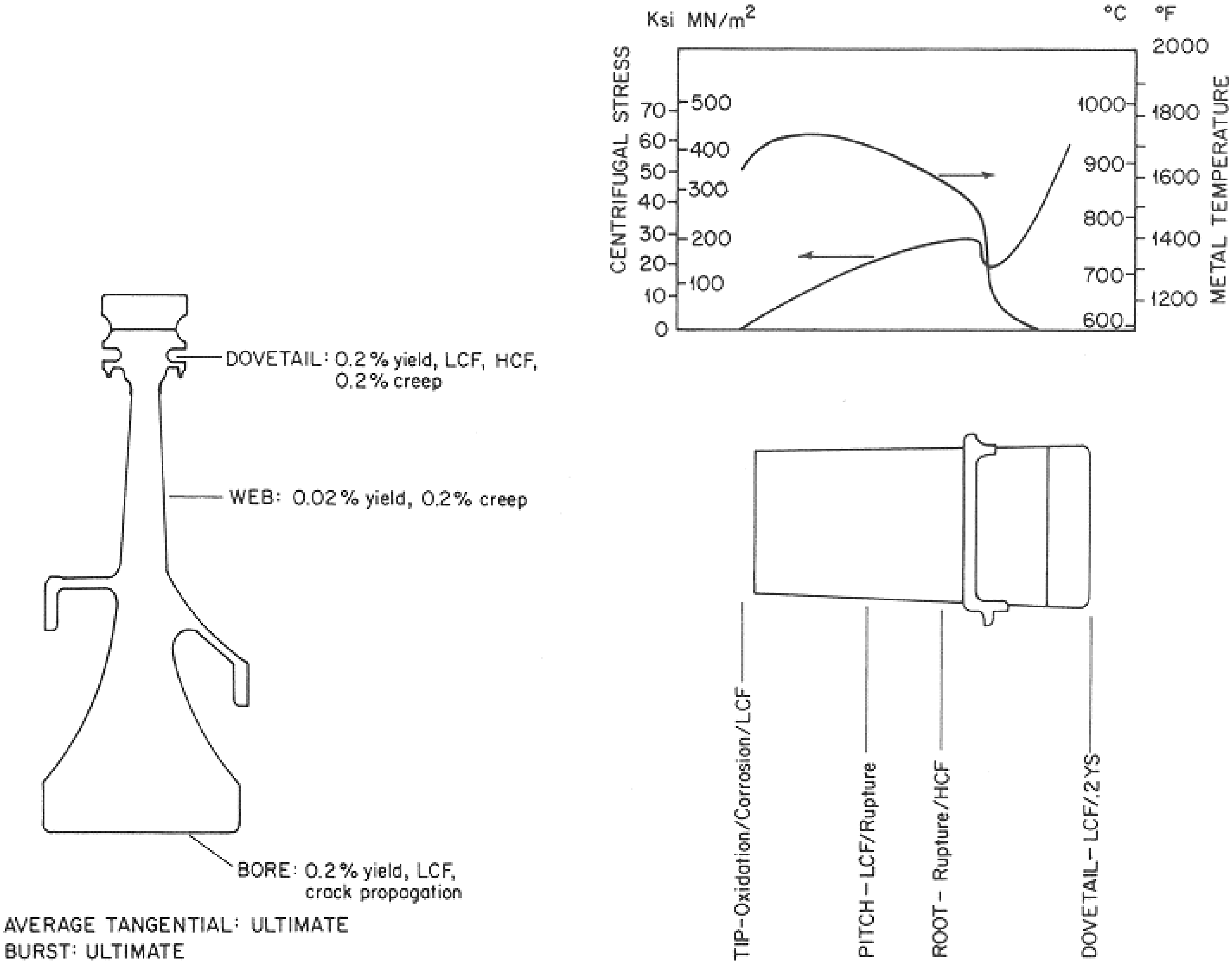
# Contents

## 1. Introduction

The objective of this document is to review the technological choices proposed in past experiences concerning gas turbines. Following the recommendations of the HTR-E/WP1 meeting held in Juelich [1], this report will be mainly devoted to the cooling systems and the deblading problems observed in HTR as well as other industries. Moreover, a first chapter will briefly summarize the materials used or proposed in several gas turbines.

## 2. Materials for turbine discs and blades

Turbine disks and blades are the most critically stressed components of gas turbines. These components can fail for several reasons including ultimate tensile strength, fatigue, creep and creep fatigue. Some classical criteria used for the design of these two components are described in figure 2-1.




(a) Design criteria for disks (b) Design criteria for blades

**figure 2-1 :** Design criteria for disks and blades (from Anderson 1979) [2]

By contrast, in the GT-MHR project, the first stages discs of the turbine will mainly be designed in terms of creep properties due to the foreseen loading conditions [18] (see table 3-1). In table 2-1, the working conditions and materials proposed for the rotor and the blades are displayed for several types of gas turbines : industrial, aero and helium gas turbines. When the choice of a material for the turbine components is driven by the creep properties, it is important to consider three parameters, namely the temperature, the applied stress and the estimated operating time, to compare the different choices displayed in this table.

| | TIT (°C) | cooling | Disc T (°C) | Turbine Disc diameter (m) | Disc material | Blades material | Overhauls (hr) |
|---|---|---|---|---|---|---|---|
| MS9001H (GE) | 1430 | N D B | < 600 | - | IN718 | SX (1st stage) | 20000 |
| M701F Mitsubishi | 1380 | N D B | - | - | 3-1/2 Ni alloy steel | IN738LC | 20000 |
| GT26 Alstom | 1425 | N D B | - | - | - | DS CM247LC | 20000 |
| CFM-56-5C | | N D B | - | ~0.8 | - | HPT : SX N5 | ~5000 |
| M88-2 | 1577 | N D B | < 650 | ~ 0.48 | PM N18 | SX AM1 | ~100 |
| GT-MHR Allied Signal [14] | 850 | D | < 600 | 1.36 | Alloy 718 forging | Cast alloy 713LC, Rene N4 | 60000 |
| GT-MHR OKBM [18] | 850 | - | - | 1.52 | EI698M, EP741H, EP975ID,EP962E,EP202 | ZhS6, ZhS40 | 60000 |
| GT-MHR / HTR-M [19][20] | 850 | - | < 700 | ~ 1.5 | U720 forging | IN792DS (CrOx)<br>CM247LCDS (AlOx) | 60000 |
| PBMR [11] | 900 | N D B | - | - | IN100 | IN100 | 50000 |
| EVO [9] | 750 | N D + blade feet | - | HPT 0.688 | High alloyed Austenitic steel | U520 | - |
| HHV [9] | 826 | N D + blade feet | < 400 | 1.6 | ferritic | Nimocast 713 LC | - |
| HTGR-GT JAERI [13] | 850 | D | < 600 | | IN706 | MARM247 | |

N : Nozzle, B : Disc, B : blade

**table 2-1 :** Gas turbines general comparisons

The major overhauls proposed by the manufacturers are displayed in the last column. It is important to notice that during these overhauls the hot parts are likely to be changed if necessary. Thus, the overhaul period corresponds to the minimum required lifetime of the hot components.

Although the turbine inlet temperature of the gas is higher in land-based as well as aero gas turbines than in the HTR ones, cooling systems are necessary for the nozzles, the blades and the rotors for the first stages. Even in the case of the military aircraft engine M88-2, the TIT is 1577°C, but the discs are cooled below 650°C. Moreover, these components are changed approximately every 100 hours, compared to the foreseen 60000 hours in the GT-MHR. Note that the loadings of the discs are of course very different in both cases.

The materials proposed for the discs and blades depend on the temperature of the components and in consequence of the cooling system design, as well as the material state of the art when the gas turbine has been designed. The more recent gas turbines use blades made of single crystals or directionally solidified superalloys to enhance creep properties. Concerning the disc, research is actually based on the development of forged U720 for large discs. The powder metallurgy process is also considered as an interesting route. The materials proposed for the PBMR, or used in the EVO and HHV rotors are based on a cooling of the disk. In the HHV turbines, the discs were cooled below 400°C allowing the use of a ferritic steel.

The up to date knowledge on high temperature materials for discs is based on the use of superalloys. These materials are strengthened by means of nanometric $\gamma'$ ($Ni_3$ (Al, Ti)) or $\gamma''$ ($Ni_3Nb$) precipitates. These precipitates are obtained by the following particular heat treatments :

- Solutionning at high temperature (typically in the range 1100°C-1200°C) followed by a cooling at a specific (and rather high) cooling rate.
- Ageing treatment at a temperature ranging between 650°C and 850°C, to grow the small precipitates up to the optimum size.

For temperatures close to the ageing treatment one, these small precipitates will coarsen leading to a higher mean free path for the dislocations and to a decrease of the material properties. This explains why the use of superalloys strengthened by $\gamma'$ phases is reduced, with the actual knowledge, to temperatures below 650°C, for high stress levels as well as long lifetime as foreseen in the HTR projects. In a near future, one can expect that this maximum allowable temperature will reach 750°C. For higher temperatures, other types of materials will have to be introduced.

However, using materials with the highest possible in-service temperature should help to decrease the derivated mass flow necessary to cool the discs, leading to a higher turbine efficiency.

| | Ni | Cr | Co | Mo | W | Nb | Ti | Al | Re | C | Ta | B | Hf | Fe |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| IN100 | bal | 10.0 | 15.0 | 3 | - | - | 5.0 | 5.5 | - | 0.18 | - | 0.01 | - | - |
| IN706 | bal | 16.0 | - | - | - | 2.9 (+Ta) | 1.75 | 0.2 | - | 0.03 | - | - | - | 37.5 |
| U720 | bal | 16.2 | 15.3 | 3.06 | 1.33 | - | 5.18 | 2.47 | - | 0.023 | - | 0.018 | - | 0.06 |
| IN718 | bal | 19.0 | - | 3.0 | - | 5.0 | 0.9 | 0.5 | - | 0.04 | - | - | - | - |
| PM N18 | bal | 11.5 | 15.7 | 6.5 | - | - | 4.35 | 4.35 | - | 0.015 | - | 0.015 | 0.45 | - |
| IN738LC | bal | 16.0 | 8.5 | 1.75 | 2.6 | 2.0 | 3.4 | 3.4 | - | ? | - | 0.01 | - | - |
| CM247LC | bal | 8.0 | 9.0 | 0.5 | 10.0 | - | 0.7 | 5.6 | - | 0.07 | 3.2 | 0.015 | 1.4 | - |
| IN713LC | bal | 12.0 | - | 4.5 | - | - | 0.6 | 6.0 | - | 0.05 | 4.0 | 001 | - | - |
| IN792DS | bal | 13.0 | 9.0 | 2.0 | 4.0 | 2.0 | 4.2 | 3.2 | - | 0.2 | - | 0.02 | - | - |
| U520 | bal | 19.0 | 12.0 | 6.0 | 1.0 | - | 3.0 | 2.0 | - | 0.08 | - | 0.005 | - | - |
| MARM247 | bal | 8.0 | 10.0 | 0.6 | 10.0 | - | 1.0 | 5.5 | - | 0.15 | 3.1 | 0.015 | 1.5 | - |
| N4 | bal | 9.0 | 8.0 | 2.0 | 6.0 | 0.5 | 4.2 | 3.7 | - | 0.06 | 4.0 | 0.004 | - | - |
| N5 | bal | 7.0 | 8.0 | 2.0 | 5.0 | - | - | 6.2 | 3.0 | - | 7.0 | - | 0.2 | - |
| AM1 | bal | 8.0 | 6.0 | 2.0 | 6.0 | - | 1.2 | 5.2 | - | - | 9.0 | - | - | - |

**table 2-2 :** Chemical composition (wt %) [3]

In table 2-2, the chemical composition of the main materials displayed in table 2-1 is given. Note that most of these materials have a high content of Cobalt. For nuclear applications such as HTR, the problematic of Cobalt content of turbine disks and blades materials in case of erosion by dust particles issued from the core has been addressed in the HTR-E project [4]. No significant Cobalt activation risk has been identified on the PCS components. Note that the Russian team working on the GT-MHR project still leaves this question opened [18].

## 3. Cooling systems

In the HTR-M project [20], the analysis of the stresses and of the temperature fields likely to occur within the GT-MHR project, along with the up to date materials mechanical properties, lead to the conclusion that it will be necessary to cool the turbine discs. By contrast, the blades and vanes should not necessary be cooled. As a consequence, in this chapter, we will mainly focus on turbine disc cooling.

The figure 3-1 displays a schematic of a two-stage cooled terrestrial gas turbine [5]. Capitals letters A to F correspond to the blade and vane cooling air stream, while letters a to f stand for the associated parasitic streams. The stator and the vanes are cooled by an air flow coming from the compressor exit (A) or a medium compressor stage (B) depending on the local pressure. In this figure, the discs are cooled by a windage (D, E, F) coming from the compressor exit. The pressure of these flows is higher than that of the main flow to avoid hot gas ingress towards the disks. This air flow is then conducted inside the blades and finally reintroduced in the main gas flow.

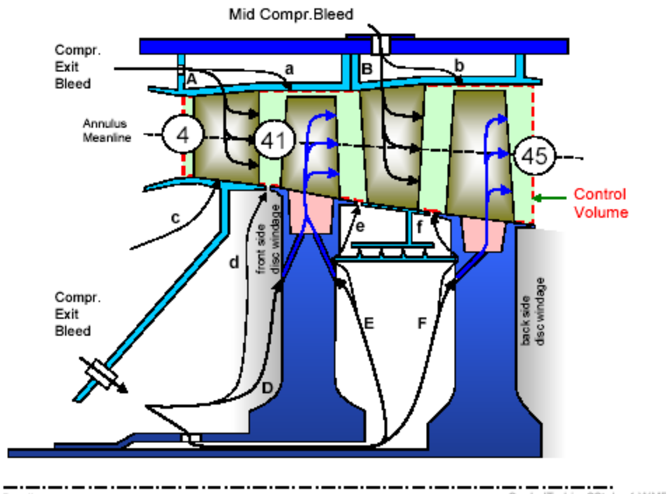



**figure 3-1 :** Schematic of a two-stage cooled turbine [5]

In figure 3-2, another drawing of the classical cooling circuits in a gas turbine is presented. Since turbine inlet temperatures are high, turbine cooling is very sophisticated : over 10% of compressor airflow goes to turbine blades, nozzle and disc cooling [6]. On the figure, one can see that the first rotor is cooled in a different way than the following ones.

Some requirements must be fulfilled by any applied cooling system [7] :

- The reduction in turbine efficiency and overall cycle efficiency compared to an uncooled turbine with the same inlet temperature should be kept as small as possible
- The performance level of the cooling system should not deteriorate during operation.
- The system should be mechanically simple and relatively easy to manufacture and to service.
- The application of the cooling system has to be justified on the basis of the overall economics of the installation.

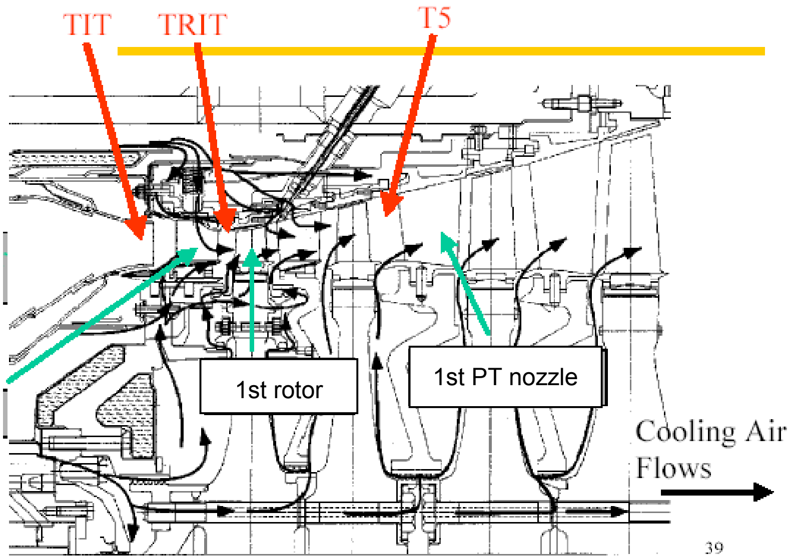



**figure 3-2 :** Cooling circuit in a gas turbine [6]

The main drawbacks due to a cooling system for the hot components of turbine are the following ones :

- When the derivated mass flow is reintroduced at the turbine inlet and at all the cooled stages, this leads to a decrease of the average temperature of the gas in the main flow leading to a decrease of the efficiency.
- The design of a cooling system is specific to each gas turbine. Adding a cooling system complicates the whole design and manufacturing and increases the cost.
- In the case of a closed cycle with a heat exchanger, the derivative mass flow also has an effect on the efficiency of the heat exchanger because the LP and HP mass flows in the heat exchanger will be unbalanced.

In the following, the cooling options chosen for some industrial and helium turbines are presented.

### *3.1. Industrial gas turbine*

In figure 3-3, the cooling circuit of the heavy duty General Electric MS7001F gas turbine is presented [8]. This gas turbine has the following main features :

Thermal power : 135.7 MW (simple cycle)

Shaft speed : 3600 rpm

TIT (°C) : 1260°C

Compressor : 18 stages

Turbine : 3 stages

The turbine rotor consists of three M-152 (12CrNiMoV steel) alloy wheels separated by spacers of the same alloy and with an aft bearing shaft of CrMoV steel. Each of the three rotor stages consists of 92 investment-cast buckets of GTD-111.

The first stage nozzle and the first and second stage buckets are air cooled via internal cooling circuits sourced from compressor discharge and the 18$^{th}$ compressor stage. The second and third stage nozzles are cooled via external circuits sourced from the 14$^{th}$ compressor stage. As can be seen in the figure, the cooling circuits also provide cooling of the first two turbine discs. The bucket cooling is supplied by air flowing radially inward at the 18th stage compressor wheel, then through 15 holes drilled axially through the distance piece, and then over the forward face of the first-stage turbine wheel. The bucket cooling air then flows through the bore of the first-stage turbine wheel into the chamber between the first and second-stage wheels to the root of the first and second-stage buckets.

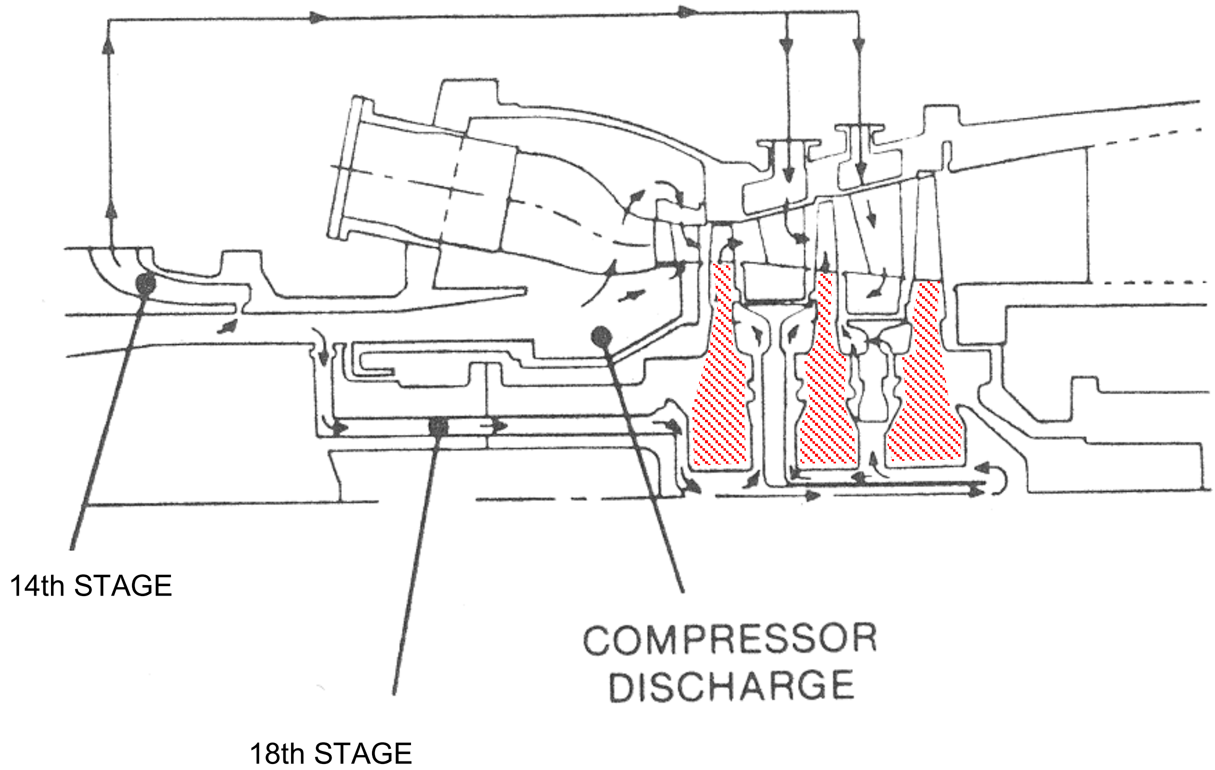



**figure 3-3 :** Cooling circuit in MS7001F gas turbine (GE) [8] (turbine disks shaded in red)

Despite the high turbine inlet temperature, the maximum temperature of the discs remains below 600°C due to the cooling circuits.

## *3.2. Helium gas turbines*

### 3.2.1. EVO

Within the "Obheiz 2" HP turbine, a separate inner turbine housing was provided which was cooled by the passage of outlet helium from the HP compressor. The turbine rotor is provided with cooling for the rotor discs and the blade feet (figure 3-4, figure 3-5) : a cooling flow of helium was extracted from the HP compressor outlet and guided through the bore of the rotor shaft to the ring-shaped chambers of the turbine rotor discs and then finally, after the last turbine stage, ducted into the main helium-stream. Owing to this cooling, austenitic steel was chosen for the turbine rotor. The ring-shaped chambers were formed by the outer surfaces of the rotor discs and the blade shoulder [15].

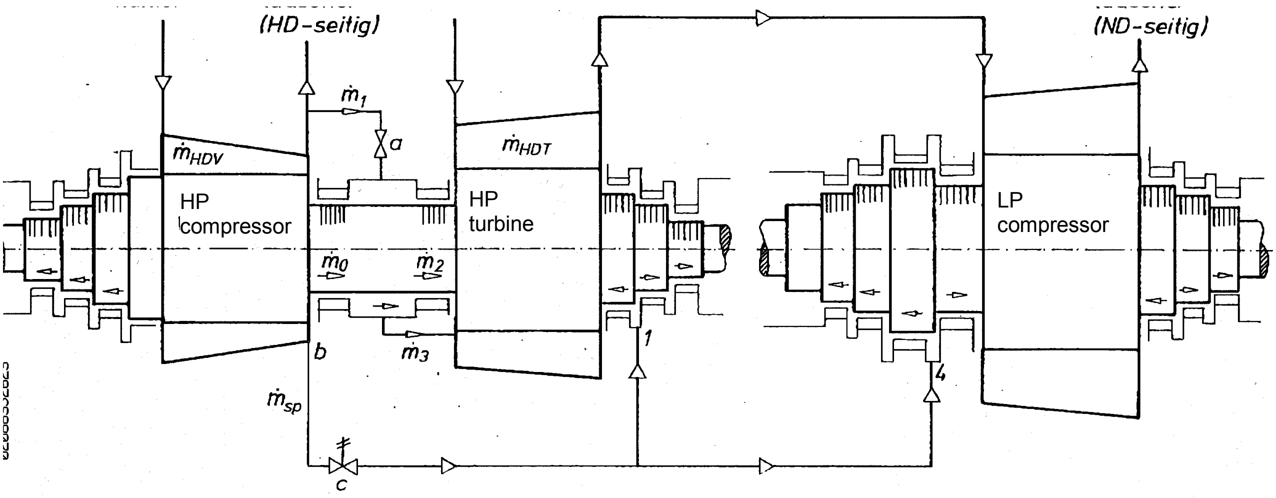



**figure 3-4 :** Cooling gas flow in the EVO "Obheiz 2" helium loop [16]

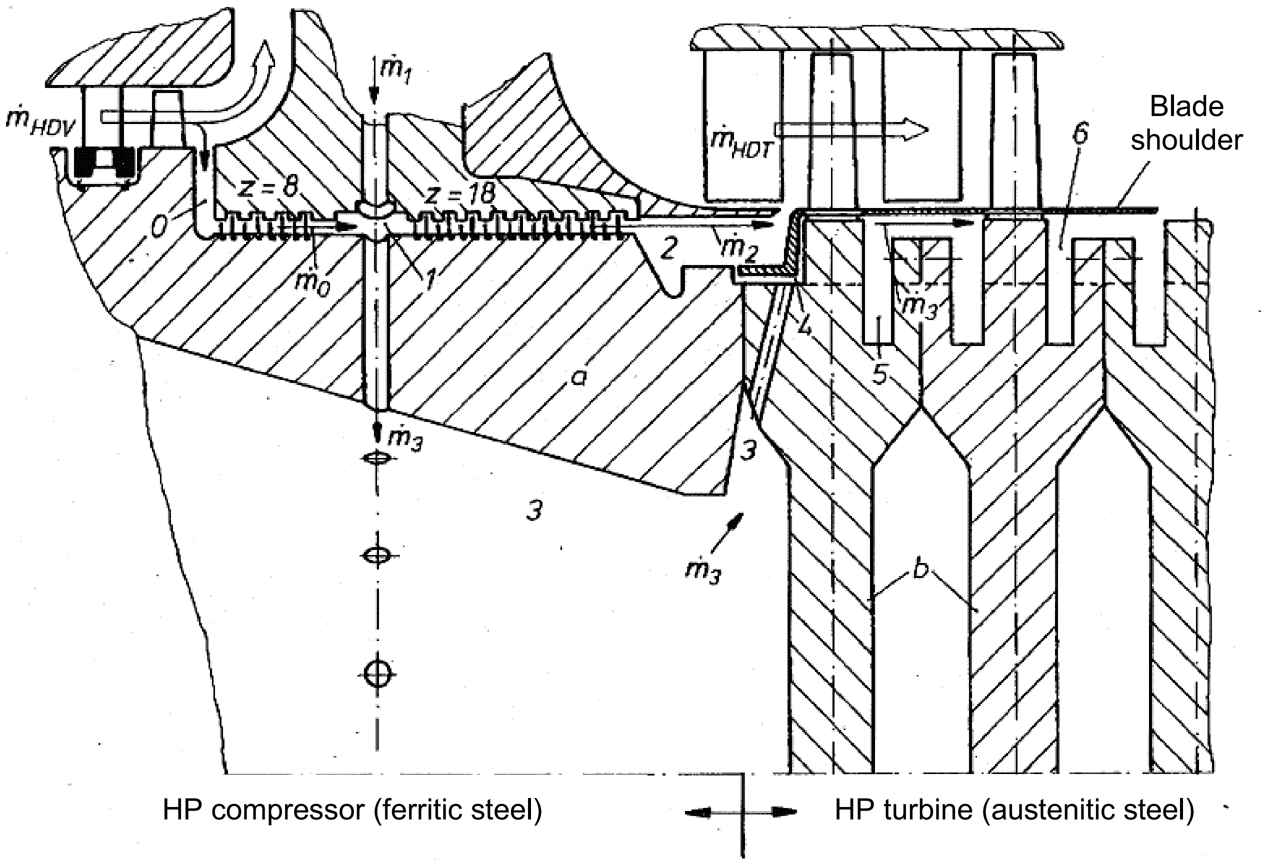



**figure 3-5 :** Cooling gas system - EVO "Obheiz 2" HP turbine (0-6 : ring shaped chambers) [16]

In these two figures, six different helium mass flows are identified :

$\dot{m}_{HDV}, \dot{m}_{HDT}$ : main helium mass flows in the HP compressor and the HP turbine.

$\dot{m}_0$ : cooling flow coming from the compressor outlet through the labyrinth seal to the ring-shaped chamber "1"

$\dot{m}_1$ : cooling flow coming from the compressor outlet and controlled by the valve "a" (figure 3-4).

$\dot{m}_2$ : cooling flow from the ring-shaped chamber "1" and going through the labyrinth seal to the ring-shaped chamber "2"

$\dot{m}_3$ : cooling flow from the ring-shaped chamber "1" towards the compressor ring-shaped chamber "3" (through the radial holes machined in the rotor) and then to the turbine ring-shaped chamber "4".

From these definitions and figures, it is clear that : $\dot{m}_0 + \dot{m}_1 = \dot{m}_2 + \dot{m}_3$.

During operations, it appeared that the helium mass-flow rates for the cooling and the sealing gas were much larger than predicted. This contributed to the deficiency of power production of the turbo-set. The power losses due to the cooling system have been estimated at 5.3 MW, to be compared to the total power loss in the circuit equal to 19.5MW [9].

### 3.2.2. HHV

In the HHV design, the blade feet of the turbine as well as of the compressor sections, the rotor and the housing were cooled by means of the cooling gas system or the sealing gas system (figure 3-6). The designed rotor cooling, by means of cold helium conducted through cooling channels, proved to be very effective, assuring that the disc temperatures could easily be kept below 400°C .

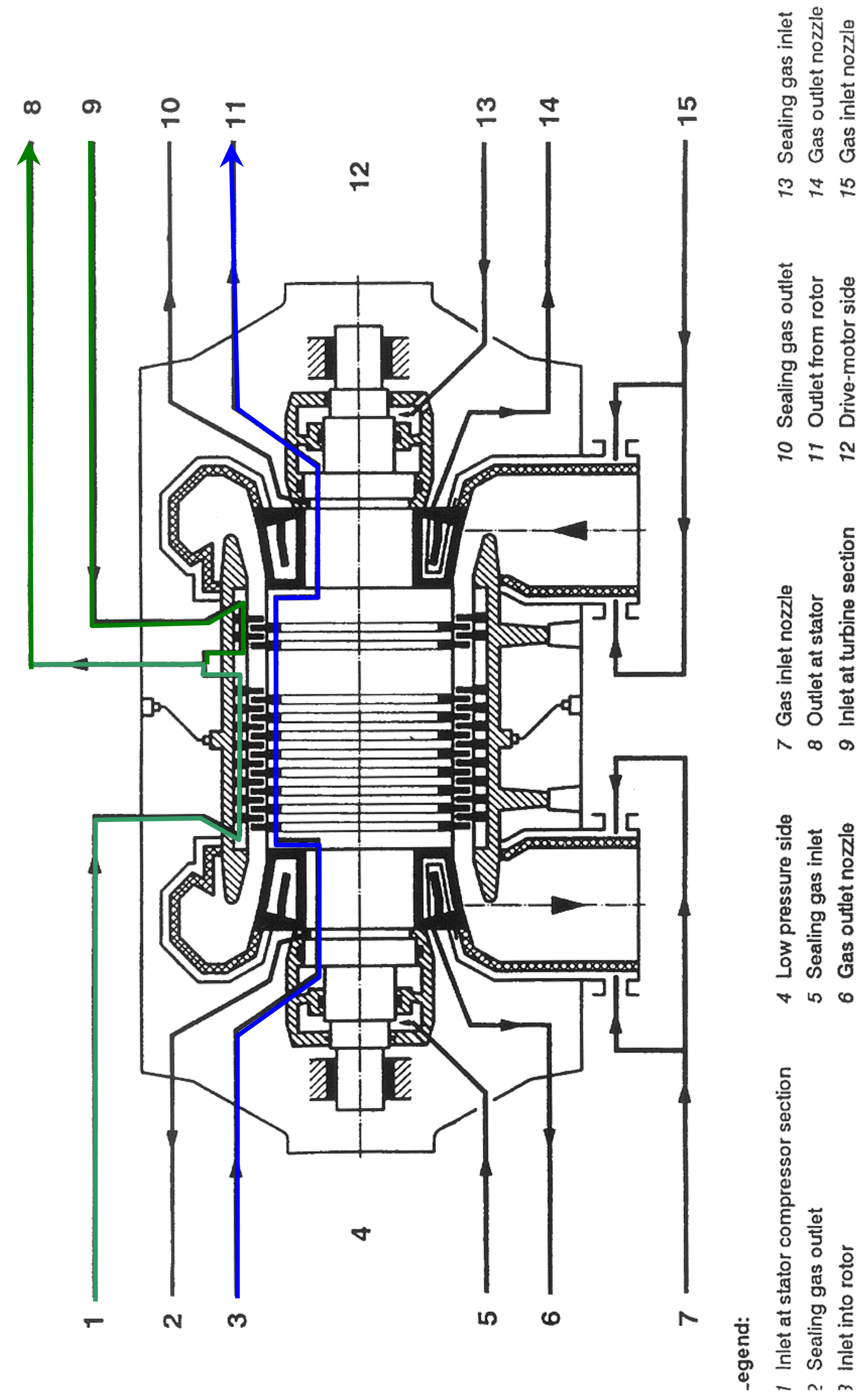



**figure 3-6 :** Cooling and sealing gas flow in the HHV turbomachine (rotor cooling : blue, stator cooling : green) [9]

The measured coolant gas flow rates were 10 to 20% larger than the planned flow rates, so the actual temperatures of the rotor discs and blade feet were distinctly lower than forecasted. Only at the turbine inlet were the measured temperatures 20-30°C higher than estimated by the design, but still below permissible values.

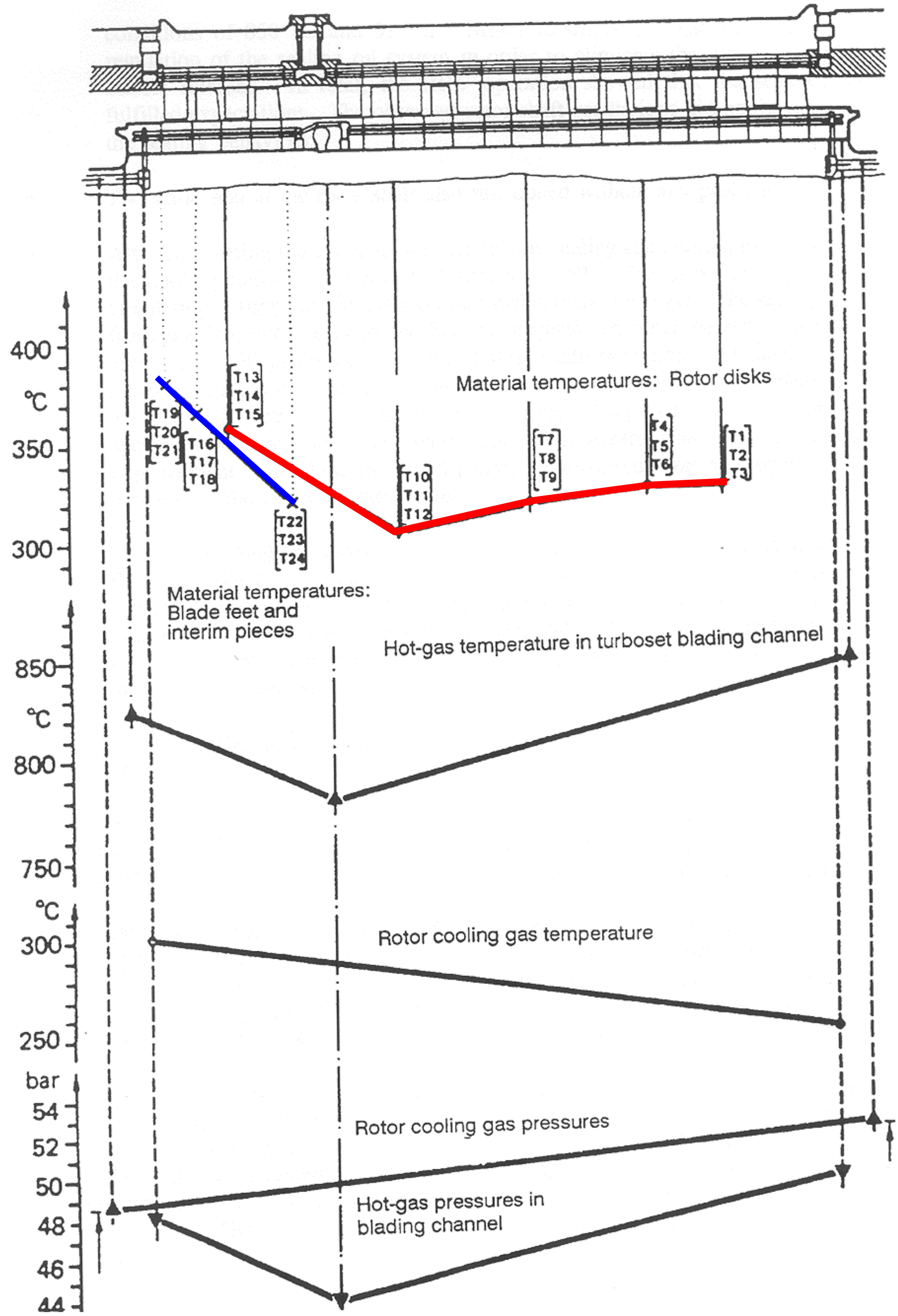



**figure 3-7 :** HHV temperature and pressure patterns in blading channel, cooling channels and rotor discs [9]

The figure 3-7 shows the pressure and temperature patterns in the blading channel and in the cooling channels as well as the rotor disc and blade feet temperature. The cooling system functioned so well that an operating temperature of 1000°C for the turbine seems to be possible [9].

### 3.2.3. HHT

Within the HHT project that was initiated in Germany in 1968 and has been cancelled in 1981, studies have shown that a proven rotor cooling system based on HHV feedback, absolutely safe in operation, is available. Such a cooling system could ensure that the rotor temperature in the helium turbine remained below 400°C [10].

### 3.2.4. PBMR

The data concerning the PBMR design have been found in the public literature. As a consequence, they are out of date since the turbomachinery is being completely redesigned.

In [11], Liebenberg discussed the design options for a 220 MWt PBMR. In this project, it was planned to cool the rotor blades LP turbine (inlet temperature of 820°C) to increase creep life. Moreover, the stator blades had a single cooling channel which should also scavenge hot gas ingress to the disc front and the inter disc space. The HP turbine was a scaled-down version of the LP turbine. The higher inlet temperature (900°C) made rotor blade cooling more critical and it required a larger cooling flow rate [11].

In a design discussed more recently [17], the three turbines are cooled. Concerning the power turbine (inlet temperature : 720°C), the cooling of the discs was foreseen as follows : each set of stator blades support a seal ring on the inside, which has a sealing surface towards the rotor center-line and upper disc. Cooling gas is passed through the hollow blade shank and seal ring to a cavity between the two seal faces. The quantity of gas flow is such that a temperature diluted stream will pass through the seals and adequate cooling of the disc will take place. The cooling flow was estimated at 2.6 kg/s for the power turbine, 2.1 kg/s for the HP Turbine and 3.0 kg/s for the LP Turbine.

### 3.2.5. JAERI

In the HTGR-GH project developed by JAERI, it has been decided to cool the turbomachine hot components in order to use existing material and technology. The turbine disc has a large diameter and the heat-resistant temperature of the material suited to such a large disc is approximately 600°C. Approximately 1% (~3 kg/s) of the total mass flow is extracted from the HP compressor outlet. The performance drop due to this cooling is estimated to be 0.5 to 1% [12] [13]. Unfortunately, a clear design of the cooling system has not been found in the literature.

### 3.2.6. GT-MHR

In the GT-MHR design analysed by Allied Signal [14] in 1995, an active cooling is considered necessary to lower the temperature of the first stage turbine discs and shafts to around 650°C. According to these authors, the secondary flow distribution needs to be addressed early in the conceptual design due to its impact on the thermal growths of the rotating components. A preliminary cooling circuit is presented in figure 3-8. The cooling flow originates at the HP compressor discharge. However, the authors state that unless a more specific cooling flow path is defined, this vague introduction of helium in the vicinity of the hot turbine wheel will do little to cool the first stage turbine wheel.

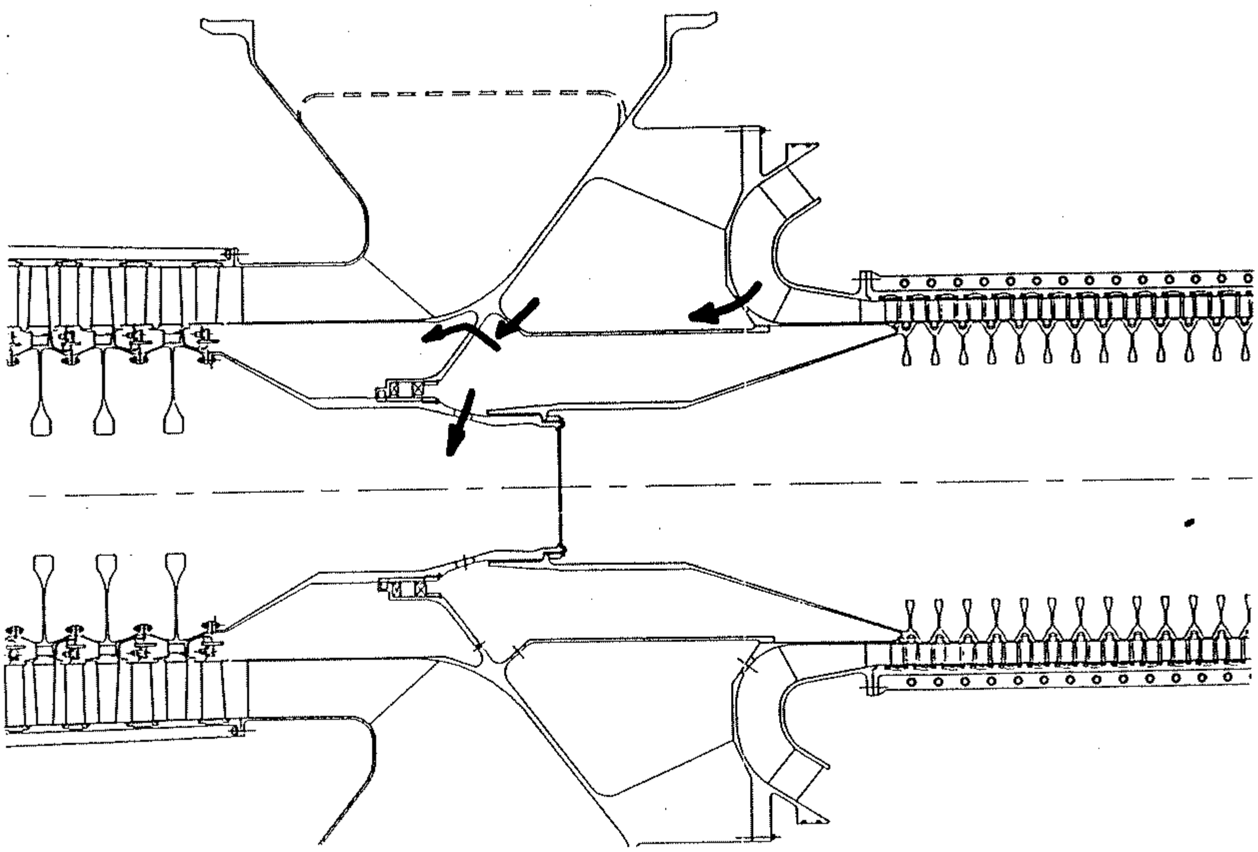


**figure 3-8 :** Preliminary GT-MHR cooling flow design (Allied Signal) [14]

More recently, Romantsov et al. [18] have performed a new analysis of the turbomachine components considering welded or assembled discs. From thermomechanical calculations, the authors note that the specified 60000h lifetime, associated to a quite high stress level in the turbine discs leads to high requirements on the mechanical characteristics of the used structural materials. Thus, they analysed both the non cooled as well as the cooled options.

To investigate the number of stages necessary to cool, the assumption of constant enthalpy drop per stage can be made. Neglecting the variations of the helium specific heat with temperature, the temperature drop per stage is then (as the turbine has 12 stages) :

$$\Delta T_{stage} = \frac{848-511}{12} \approx 28K$$

Considering that as long as the gaz temperature is greater than 650°C, it will be necessary to cool the discs, the number of stages is :

$$n \approx \frac{(848-650)}{28} \approx 7$$

Thus, approximately the 7 first turbine discs should be cooled in order to allow the use of the materials proposed in the HTR-M projects. This rough estimation is confirmed by the temperature fields in the non cooled stator casing displayed in figure 3-9 [18]. The blue dash line on this figure corresponds roughly to the 650°C iso temperature line. The temperature of the stator casing decreases below 650°C within the 7$^{th}$ stage.

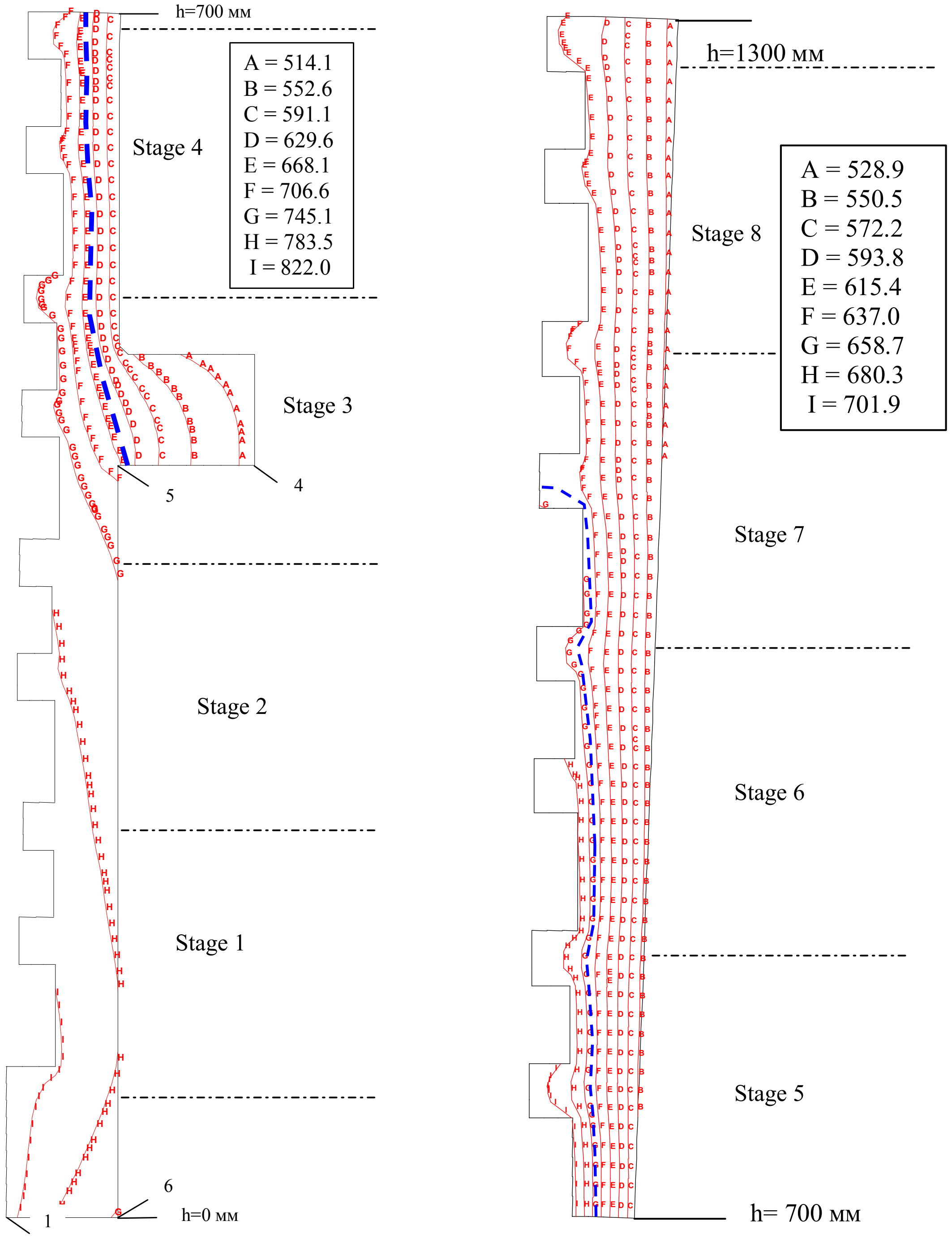



**figure 3-9 :** Results of the calculation of the stator casing temperature state ($^0$C) [18]

The authors also displayed the temperature fields in the turbine discs of the first three stages for the cooled option. Due to material limitations, the maximum temperature reach in the first disc in 100% mode operation is below 650°C as seen in figure 3-10.

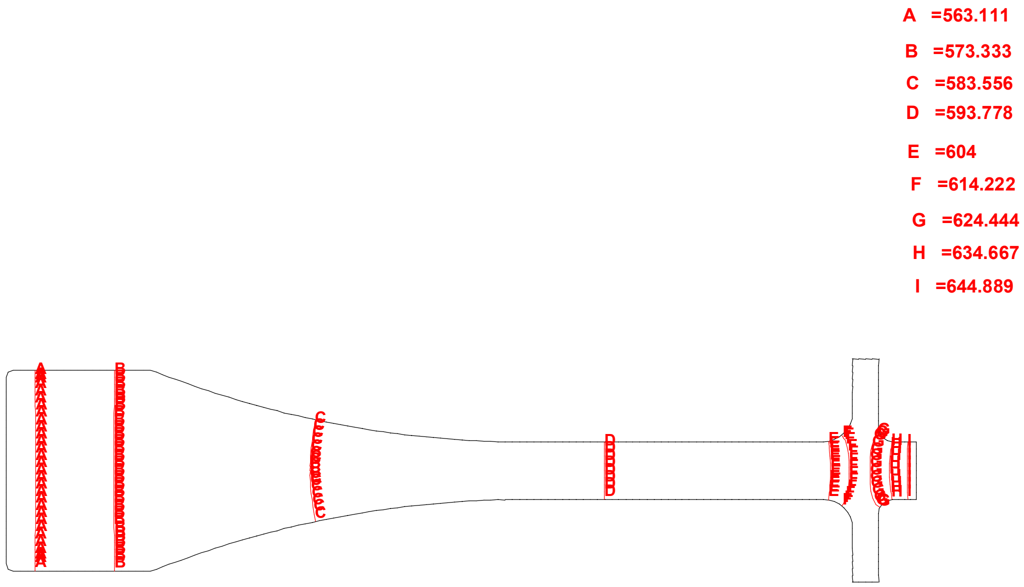



**figure 3-10 :** Temperature field (°C) of the first stage turbine disc of the welded rotor in 100% operation mode [18]

From their calculations, the authors proposed minimal required mechanical properties for the materials of the turbine most stressed components. Considering the turbine disc, the following data are provided :

| | Calculation T (°C) | Creep strength (MPa) for 0.2% deformation after 60000 h | Creep strength (MPa) for 1% deformation after 60000 h |
|---|---|---|---|
| cooled | 650 | 270 | 295 |
| | 700 | 174 | 200 |
| non cooled | 820 | 74 | 100 |

**table 3-1 :** Minimal creep stresses for the turbine disc material [18]

# 4. Deblading

The HTR projects (mainly the PBMR and the GT-MHR) analysed within this European contract are based on a direct cycle. Thus, it is an important safety concern to analyse the consequences of a possible deblading. In case of failure in a direct cycle turbomachine several scenarios have been considered in the literature :

- Past experiences on gas turbines operations indicate that parts from a gas turbine may break off and block through the turbine passages. This blockage of flow could result in a large axial pressure drop that damages the core support structure and challenges the safety functions of heat removal and control of core heat generation [21].
- A blockage could also cause a reversal of flow through the core, which in turn may cause an ejection of control rod and challenge control of core generation [21].
- By contrast, an additional scenario considers a total deblading in the turbine leading to high rate depressurization [26].

The transients due to these several scenarios should be carefully analyzed. Moreover, the risks of possible blades or discs parts ejection have to be considered for obvious safety reasons, the reactor vessel being in the vicinity of the power conversion vessel.

As described by McCloskey [25] for land based gas turbines, most failures in turbine blading of fossil units occur in low pressure blades particularly concentrated in the L-0 and L-1 stages. Each turbine blade row of a particular turbine design tends to have distinctive problems. For example, the operating life of L-0 rows is generally controlled by Low Cycle Fatigue caused by start-stop cycling, although in extreme off-design conditions stall flutter may also cause fatigue damage by high cycle damage mechanism. By contrast, the L-2 and L-3 rows of the same design are generally controlled by high cycle fatigue typically as a result of resonance with particular blade-disc modes with per-rev harmonics of rotationnel speed.

Once a blade failure occurs, other blades may fail from impact with the previous one. The entire rotor becomes dynamically unstable because of the mass imbalance. These vibrations may cause extensive damage to all components of the rotor. Indeed, missiles (blades, large disc parts) will impact the casing. It is an important safety concern to analyse the consequences of such missiles. In the following, the problems and solutions encountered in various types of gas turbines are presented.

## *4.1. Aeroengines*

Blade failure in compressors and turbine at high rotational speed is a critical event in aircraft engines. The aviation authority regulations require that containment of the blade loss is provided by the casing structure of the respective component, which has to be demonstrated by experiments or analysis [22]. According to the Federal Aviation Administration [23], few published results on design methods for the containment of rotating parts within the gas turbine engine case are available. The nature of such research, requiring a large amount of expensive test data, is such that most development has occurred at the engine manufacturers. Large engine manufacturers state that their extensive research and development efforts into rotor containment advances had cost them a lot of money. The work is thus considered to be highly proprietary and details are not generally released by the companies involved. Although some finite element simulations are mentioned, these tools are in no case presented as a simple solution. While time histories of deflection, stress levels, and reaction forces are fairly simple to obtain, selection of an appropriate failure model and hence the ability to predict containment capability, remains a problem [23].

As an example, the analysis of deblading in a low pressure turbines is presented below [22]. The analytical procedure established for containment design involves an energy balance method based on the comparison of the kinetic energy of the released blade and the strain energy of the containment zone. In figure 4-1, a comparison between a deblading experiment and the dynamic simulation is displayed. However, these kinds of simulations show the influence of the adjacent blades on the results. The choice of the material mechanical models is also crucial to the representation of the expected failure mode.

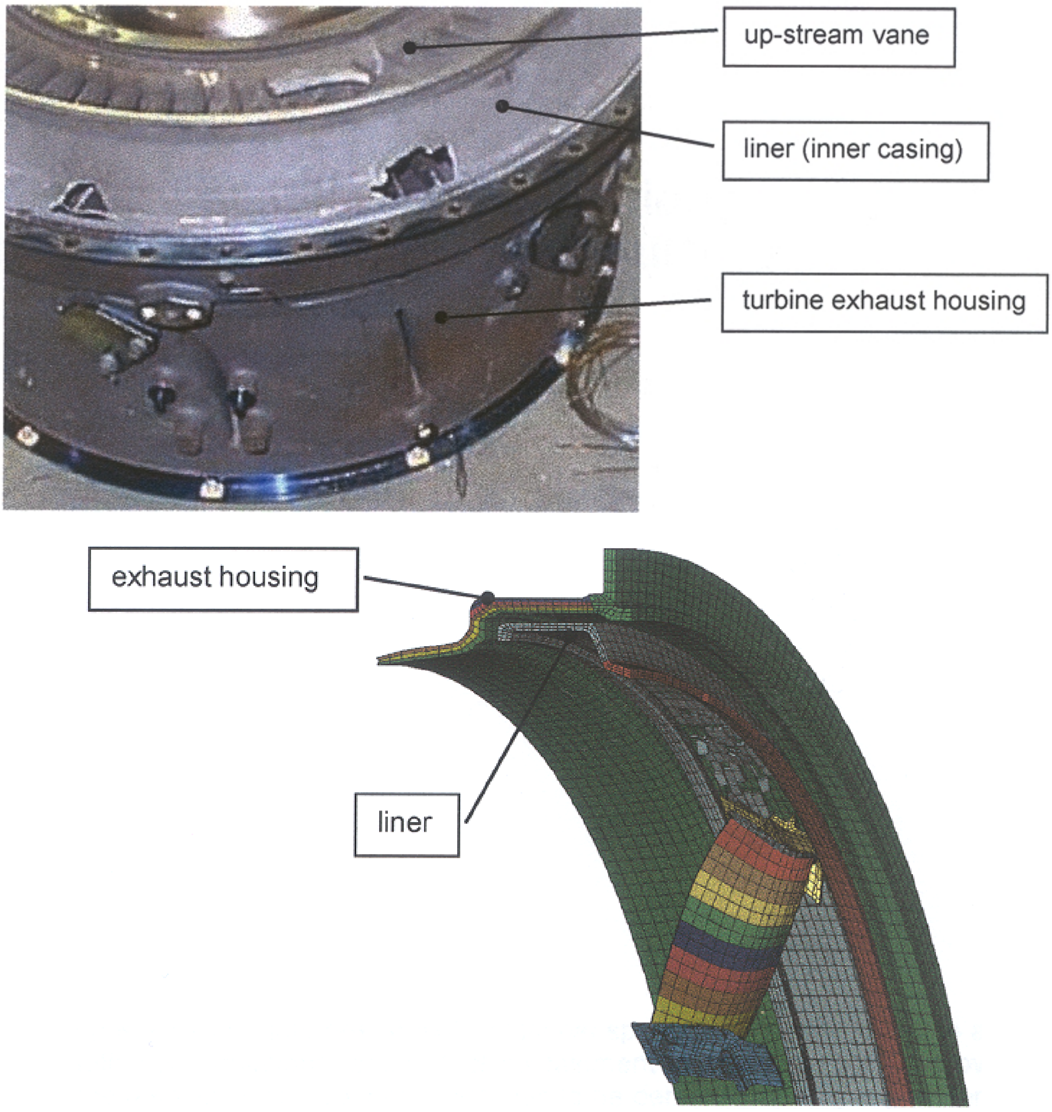



**figure 4-1 :** Penetration of the liner in the blade-off rig test (top) and in the LS-DYNA simulation (bottom) [22]

The new European Aviation Safety Agency (EASA [24]) assessed specifications for certification of aero engines, based on previous Joint Aviation Authorities statements. Concerning the safety analysis of the turbine engines, the following points corresponding to deblading and missiles problems are interesting to note in this paper.

- *Uncontained debris cover a large spectrum of energy levels due to the various sizes and velocities of parts released in an engine failure. The engine has a containment structure which is designed to withstand the consequences of the release of a single blade and which is often adequate to contain additional released blades. The engine containment structure is not expected to contain major rotating parts should they fracture. Discs, hubs, impellers, large rotating seals, and other similar large rotating components should therefore always be considered to represent potential high-energy debris.*
- *Because the failure of an engine critical parts is likely to result in a hazardous engine effect, it is necessary to take precautions to avoid the occurrence of failures of such parts.*
- *It shall be shown that Hazardous Engine Effects are predicted to occur at a rate not in excess of that defined as Extremely Remote : probability less than $10^{-7}$ per engine flight hour. The non-containment of high energy debris shall be regarded as a Hazardous Engine Effect.*
- *It is recognised that the probability of primary failure of certain single elements (for example, disks) cannot be sensibly estimated in numerical terms.*
- *It shall be demonstrated that any single compressor or turbine blade will be contained after failure and that no hazardous engine effect can arise as a result of other engine damage likely to occur before engine shut down following blade failure. It shall be demonstrated that failure of the shaft systems will not result in hazardous engine effect or it shall be shown that failures are predicted to occur at a rate not in excess of that defined as extremely remote.*

The main points to be retained for the HTR projects with a direct cycle are :

- A single blade failure should not lead to any rupture of the rotor and the blade itself should be contained by the casing.
- Discs parts can not be contained by the casing. In consequence, it must be proved that such accident has a very low probability to occur.

## *4.2. Steam turbines*

### 4.2.1. Main problems

In [25], McCloskey listed the most prevalent turbine steam path damage mechanisms :

- Creep in High Pressure and Intermediate Pressure rotors
- Creep and high-temperature damage in HP and IP blades
- Overheating caused by blade windage
- Solid particle erosion
- Deposit effects and removal in HP, IP and LP turbines
- Low and High cycle fatigue in HP, IP and LP blades.
- Fretting wear an fatigue
- Environmentally related damage in blades and blade/disc attachments
- Pitting and crevice corrosion
- Corrosion fatigue in LP blades

- Stress corrosion cracking in LP blades and around the disc rim and blades attachments
- Liquid droplet erosion of rotating and stationary blading
- Water induction damage, flow accelerated corrosion and moisture-related damage

Note that the size of the blades in a steam turbine is larger (> 1m) than in a helium turbine (~10 cm). Thus, a single blade failure should have less consequences (for example, the rotor will be less unbalanced) in a helium turbomachine than in a steam one.

### 4.2.2. EDF-Porcheville steam turbine accident feed-back

During the 1970ies, a deblading accident occurred at Porcheville (France) in an EDF steam turbomachine. The accident took place during an overspeed test after an overhaul. The scenario of the accident has been identified and described as follows. A blade of the low pressure turbine failed leading to a strong unbalanced rotor. The bearings were not mechanically dimensioned to support such a loading. The rotation of the whole rotor (3000 rpm, weight ~200tons) has been stopped in less than 1 round due to the failure of the bearings. The huge amount of energy lead to the failure of the rotor in many parts. Some parts have been found few hundred meters away from the turbomachine building. Hopefully, nobody was injured by this catastrophic accident.

After this bad experience, EDF took the following decisions :

- The bearings must be able to support the rotation of an unbalanced rotor due to the failure of one LP turbine blade.
- The turbomachine location and orientation must be such that no missile can move towards any safety buildings.

## *4.3. HTR*

In this paragraph, the feedback from previous High Temperature Reactor experiments are displayed.

### 4.3.1. EVO

The EVO helium loop has undergone a deblading problem in the HP turbine after 11000 operating hours. Prior to the incident, the electric power was around 10 MWe. Suddenly, a short decrease down to 2.5 MW followed by an increase up to 8.5 MW has been observed [15]. Meanwhile, the vibrations level in the HP turbine increased (figure 4-2).

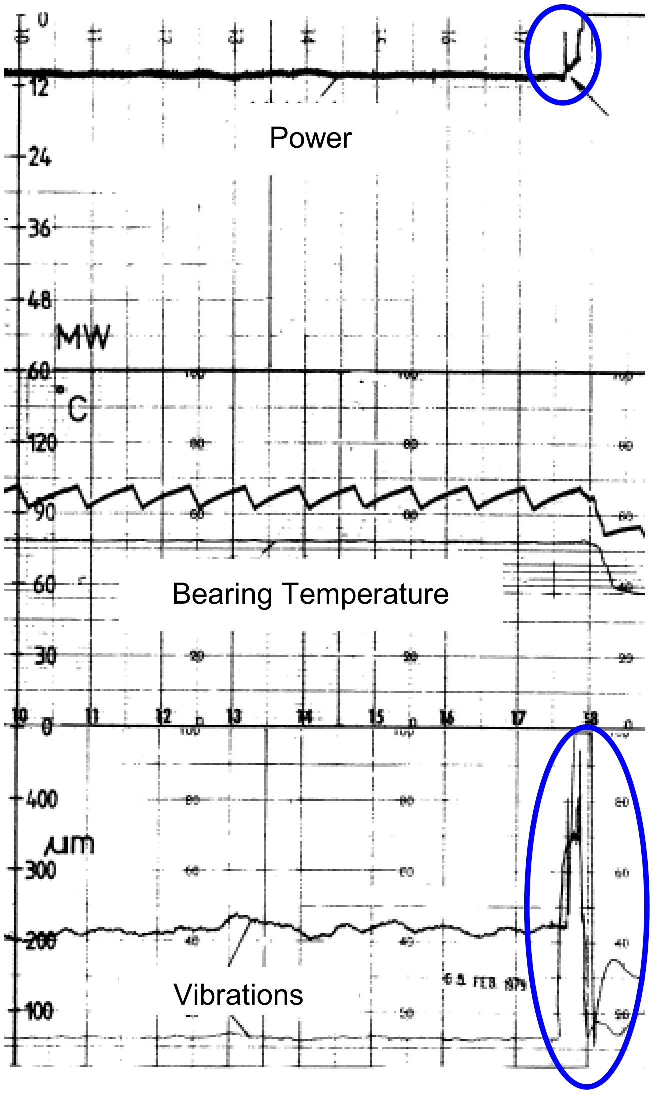



**figure 4-2 :** Power, bearing temperature and vibration in the Oberhausen HP turbine during the deblading accident [15]

The generator has then been shut down manually, and the casing has been opened. As seen in figure 4-3, a blade in the second stage of the HP turbine was broken. Subsequent blades have then been damaged. Some part of these blades reached the BP turbine where they also induced some damage.

After analysis of the broken blades, it seemed that an initial crack was present at the bottom of the blade. This defect has not been observed during the preliminary inspection probably due to he polishing of the surface that occurred before. The SEM observations showed that this crack slowly grew by a creep mechanism until it reached approximately half the section of the blade.

Compared to the catastrophic consequences observed in the case of the steam turbine accident in Porcheville described above, the deblading accident in the EVO helium turbine had minor consequences. This can be explained by the small size of the blades in a helium turbine compared to that of a steam turbine, but also by the low power reached before the accident.

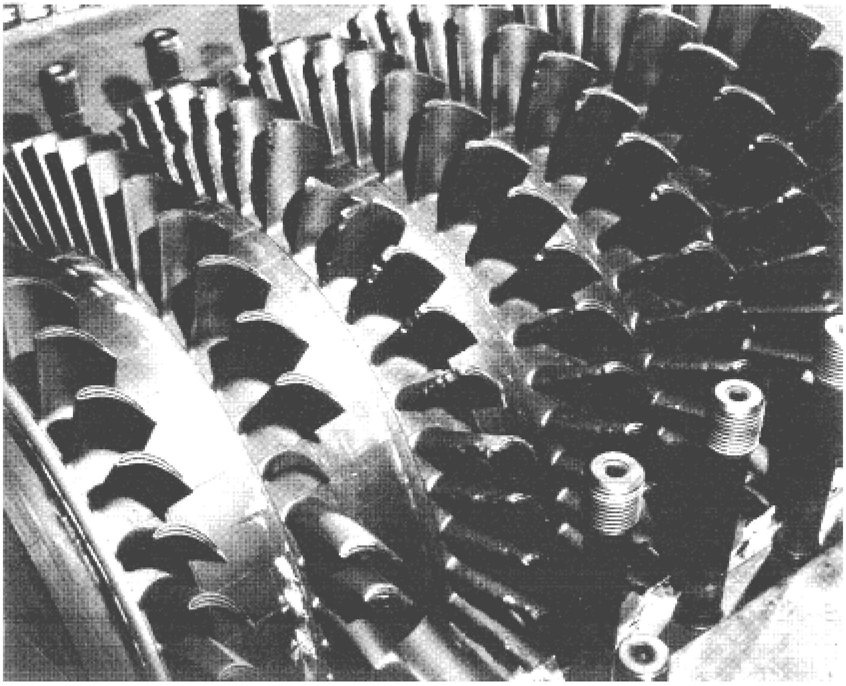


**figure 4-3 :** Deblading in the EVO HP turbine [15]

### 4.3.2. HHT

Previous studies carried out on the HHT project [26] (one single shaft) analysed the consequences of a deblading in terms of severe pressure equalization transients. Indeed, within the gas cycle, the high and low pressure levels are kept separated dynamically by the turbomachines and statically by the recuperator. Failures of these barriers may lead to potentially very severe pressure transients. To describe precisely how fast and how complete the turbine deblading can be is very difficult. Thus, the authors analysed the pessimistic assumption of an instantaneous and total deblading. This has been simulated by replacing the turbine by a nozzle with the appropriate flow area. The resulting mass flow transients are shown in figure 4-4. The peak value in the turbine is almost 5.2 times the nominal value and is reached in less than 0.1 s.

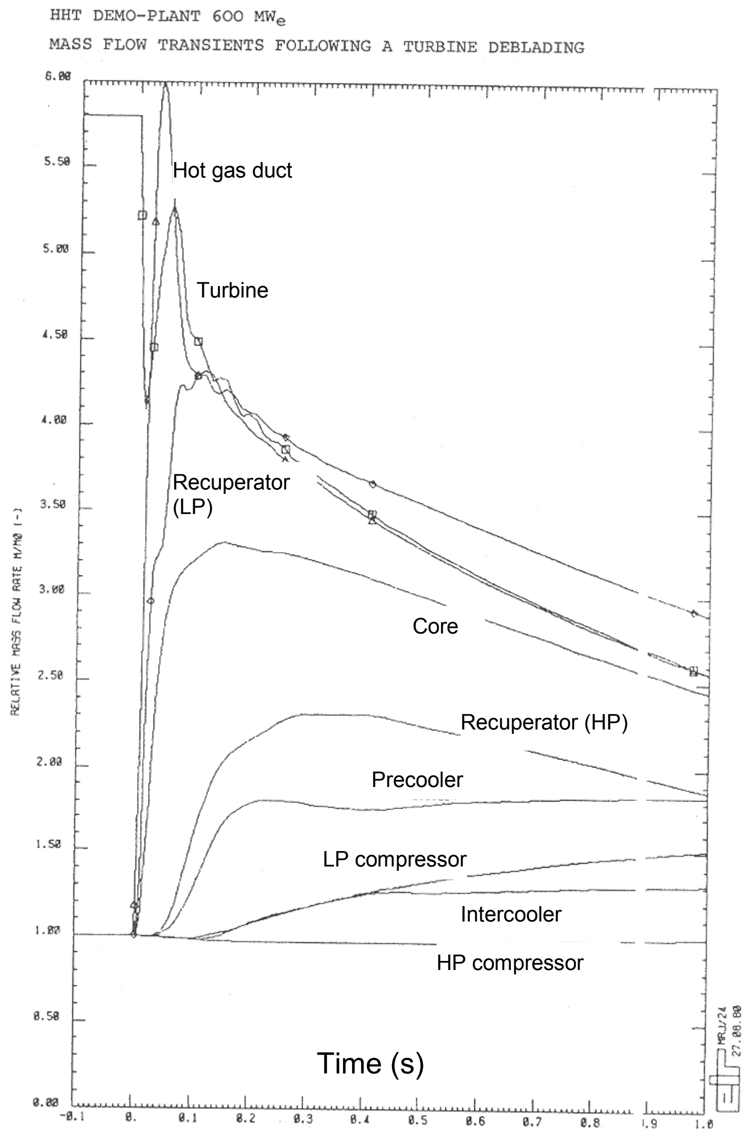
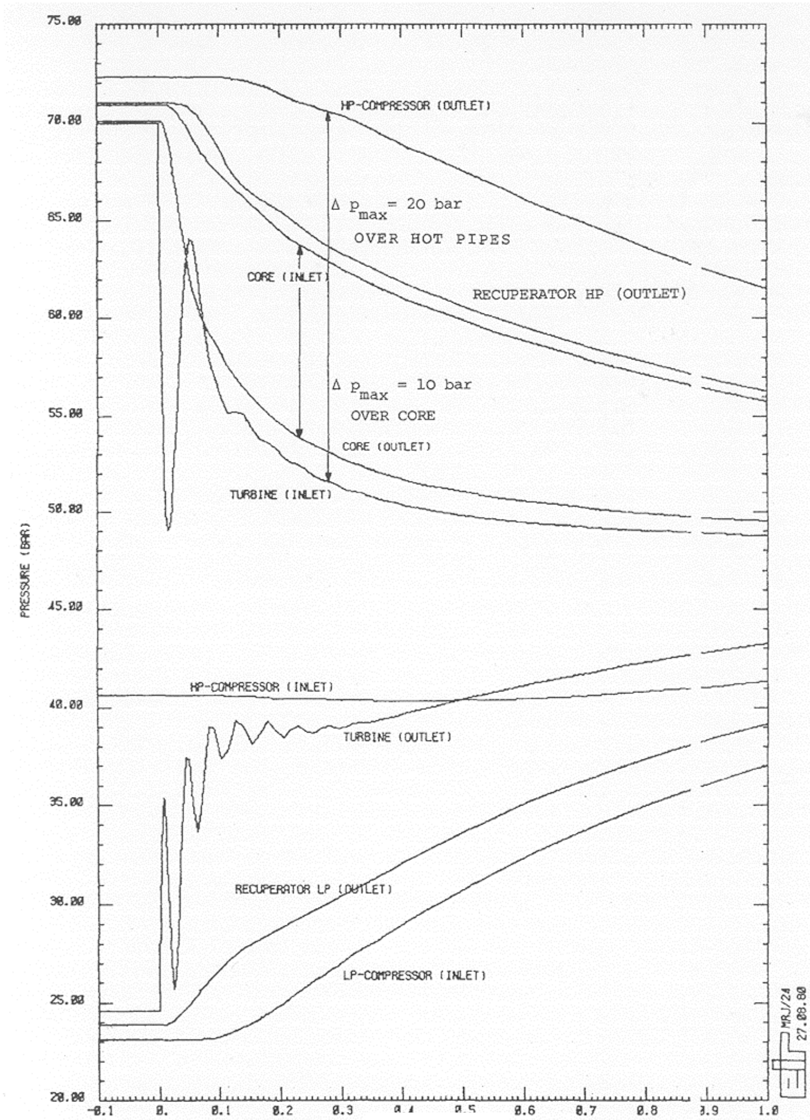



**figure 4-4 :** HHT demo plant 600 MWe, mass flow transients following a turbine deblading [26]

The corresponding pressure transients are displayed in figure 4-5. The depressurization rates on the HP side of the cycle are close to 1000 bar/s at the turbine inlet as well as in the hot gas duct and up to 200 bar/s in the core outlet plenum. These conclusions consider the worst cases of deblading. A partial deblading would reduce the pressure differences. Moreover, the time needed for partial deblading should also affect the maximum pressure differences occurring across the core and the hot gas duct.



Time (s)

**figure 4-5 :** HHT demo-plant 600 MWe, pressure transients following a turbine deblading [26]

These transients will have a strong influence on the safety valves system design.

### 4.3.3. Other HTR projects

In the previous PBMR project, the housings of the LP, HP and power turbines were designed such that a rotor disintegration due to an overspeed burst will be contained within its own housing [17].

In a previous GT-MHR project, both the turbine and compressor stators form a single load bearing cylindrical structure protecting the other PCS components and the vessel failures from turbo-compressor accidents such as deblading [17].

In the reactor proposed by JAERI [13], it is specified that the casing consists of an inlet scroll with thermal insulation structure, a thick barrel to protect turbine missiles and a large outlet diffuser. Moreover, the choice of a horizontal rotor is done for several reasons including safety ones.

In Germany, for big nuclear plants, there is a guideline (17.1) by the Reactor Safety Commission dated October 1981, which stated that the site plan should be so that safety important buildings should not be located inside a sector +/- 20° normal to the turbine axis.

## 5. Conclusion

In this report, some technological choices proposed in past experiences have been reviewed. Three topics have been addressed in this report : materials for turbine discs, cooling options and deblading accidents.

The main conclusions concerning materials and cooling options are the following :

- In order to fulfill the required mechanical properties in terms of stresses, temperature and life time, the turbine discs should be cooled if a turbine is to be built in a near future. Indeed, the development of new materials for a non cooled disc option is not foreseen before at least twenty years.
- Since the Turbine Inlet Temperature in High Temperature Reactor is low (850°C) compared to aero and land based gas turbines, the derivated mass flow of helium necessary to cool the discs is of the order of 1% of the main flow, to be compared with about 10% in classical gas turbines. Thus, the decrease of total efficiency of the helium loop should not decrease dramatically.
- Due to the rather low inlet temperature it seems that it will not be necessary to cool the nozzles nor the blades.

The case of a deblading accident has been reviewed and the following conclusions can be drawn :

- The casing should be able to contain a blade loss. By contrast, disc parts can not be contained. It will be necessary to prove that such accident has a very low probability to occur.
- The magnetic bearings should remain safe in case of a blade loss.
- The orientation and location of the turbomachine rotor should geometrically avoid any risk of missiles ejection towards safety buildings (as the reactor vessel).
- It is thus recommended to have a horizontal shaft except if it can be proved that the concrete vessel between the Power Conversion Unit and the Reactor can contain any disc parts ejection and that no missiles are likely to move through the gas duct toward the reactor. Note that such a horizontal shaft is also a better design solution for maintenance and repair.

## 6. References

[1] Briottet L., *Minutes of WP1 technical meeting*, FZJ, 14-15/11/2002, report n° HTR-E-02/11-M-1-0-2, 2002.

[2] *The design of high-efficiency turbomachinery and gas turbines*, 2nd ed., D.G. Wilson and T. Korakianitis, Prentice Hall, 1998.

[3] Heat resistant materials, ASM specialty Handbook, ASMI, Ed. J.R. Davis, 1997.

[4] Mauget C., *HTR recommendations of cobalt content in helium turbine materials*, HTR-E WP1, n° HTR-E-02/12-D-1-1-1-1, 19/11/2002.

[5] Kurzke J., *Performance modelling methodology : efficiency definitions for cooled single and multistage turbines*, Proc. of ASME TURBO EXPO 2002, Amsterdam, The Netherlands, June 3-6 2002.

[6] Kurz R., *Introduction to gas turbine and applications*, ASME Turbo expo 2002, Amsterdam, The Netherlands, 2002.

[7] Arts T., *Introduction to heat transfer phenomena in gas turbines*, Course note 127, von Karman Institute for Fluid Dynamics, Belgium.

[8] Brandt D.E., *The design and development of an advanced heavy-duty gas turbine*, J. of Eng. for Gas Turbines and Power, **110**, pp. 243-250, 1988.

[9] Weisbrodt I.A., *Summary report on technical experiences from high-temperature helium machinery testing in Germany*, Beijing, 1995, IAEA Tecdoc 899, pp. 177-248.

[10] Haselbacher H., Simon M., *The HHT INT gas turbine*, HHT presentation at the BMFT, Bonn, February 25 1977.

[11] Liebenberg J.J., *Power conversion unit for the South African direct cycle HTGR*, Johannesburg, 1996, IAEA Tecdoc 988, pp. 205-212.

[12] Matsuo E., Tsutsumi M., Ogata K., Nomura S., *Conceptual design of helium gas turbine for MHTGR-GT*, 1995, IAEA Tecdoc 899, pp. 95-109.

[13] Muto Y., Ishiyama S., Shiozawa S., Tanihira M., Miyoshi Y., Ohashi K., *Design and economics of the HTGR-GT power plant by JAERI*, Proc. of ASME TURBO EXPO 2002, Amsterdam, The Netherlands, June 3-6 2002.

[14] Allied Signal, *GT-MHR turbo compressor design report*, report 41-14039, September 29, 1995.

[15] EVO, *Troubles encountered during operating times*, *Documentation F*, *Capitel 14.1-14.8*, HTR-E/WP1.

[16] EVO, Documents concerning EVO turbine cooling system, fax 26/02/2003.

[17] *Current status and future development of modular high temperature gas cooled reactor technology*, IAEA tecdoc 1198, 2001.

[18] Romantsov A. A., *Testing and investigations of materials for turbocompressor high temperature components (from July 1,2001 to June 30, 2002)*, Report for ISTC project #1313-2001, 2002.

[19] CEA/G, HTR-M1, Deliverable D1, *Report on selection of materials and testing requirements*, 11/2002.

[20] Buckthorpe D.E., Earl C.J., HTR-M, Deliverable 2.7, *Materials for turbine : Synthesis of material database*, 06/2002.

[21] Siefken L.J., Harvego E.A., *Transient analysis needs for generation IV reactor concepts*, 2002.

[22] Kraus A., Frischbier J, *Containment and penetration simulation in case of blade loss in a low pressure turbine*, DYNAmore LS-DYNA Foru 2002, 19-20 September 2002, Bad Mergentheim, 2002.

[23] *Design procedure and analysis of turbine rotor fragment hazard containment*, Final report, DOT/FAA/AR-96/121, March 1997.

[24] *EASA Certification specifications for engines, CS-E, Issue 1 (draft)*, March 2003.

[25] McCloskey T.H., *Troubleshooting turbine damage steam path damage mechanisms*, Proceeding of the 31st Turbomachinery Symposium, pp. 105-143, 2002.

[26] Dang M., Dupont J.F., Jacquemoud P., Mylonas R., *Pressure transients analysis of a high temperature gas-cooled reactor with direct helium turbine cycle*, Lausanne, 1-3 september 1980, AIEA report #IWGGCR1, pp. 137-153, 1980.