Operational Safety of the JET In-Vessel Divertor Cryopump System

S Papastergiou, P Ageladarakis, W Obert, E Thompson.

JET Joint Undertaking, Abingdon, Oxon, OX14 3EA, UK.

ABSTRACT

A large condensation cryopump forms an integral part of the pumped divertor recently installed inside the JET torus. A detailed heat transfer model which incorporates radiative, conductive and convective heat transfer has been developed to predict the time-dependent behaviour of the high emissivity liquid nitrogen (LN) cooled chevron structure and also the low emissivity LN cooled back plate of the cryopump. In addition, the behaviour of surrounding components for a wide range of adverse scenarios has also been investigated. The model has been validated through a series of experiments.

These studies have enabled tokamak operation criteria to be defined which will ensure that the cryopump will not be over-stressed during normal operation and/or abnormal scenarios.

The studies of the behaviour of water cooled components in the vicinity of the cryopump which are at risk of freezing due to the proximity of the LN cooled structure of the cryopump have resulted in establishing criteria for the water flow and vacuum vessel temperature under all of the envisaged adverse scenarios.

1. INTRODUCTION

The tokamak environment is particularly hostile to a cryopump and high thermal stresses may occur during adverse scenarios like loss of vacuum, water, cryogens, etc. Figure 1 gives a cross-section of the JET Divertor Cryopump and its associated components, namely the water cooled baffles and target shields together with one of the divertor coils with its own thermal shields (1, 2).

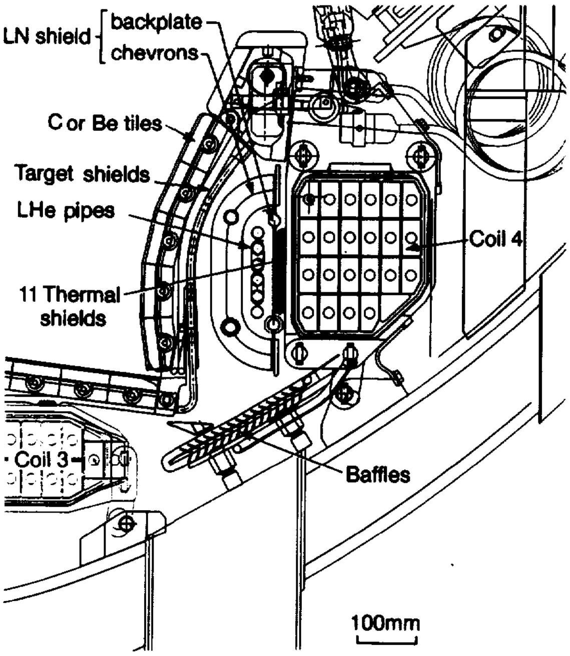

Figure 1. Cross-section of the JET Divertor Cryopump and surrounding components.

2. OPERATIONAL RISKS

a) The temperature differences (ΔT) between the black chevrons and the low emissivity backplate of the cryopump should not exceed 150°C. Linear stress analysis predicts yielding with a ΔT of approximately 40°C. However, expansion gaps and bellows in the system allow this larger value of 150°C.

b) Loss of water flow in the baffles may result in freezing or boiling of the water and stress the pipework.

c) Similarly, stop of water flow in the target shields may result in freezing of the water there.

3. MATHEMATICAL MODEL

When water is not running in the system, the general equations governing the behaviour of the system are:

$$\Sigma W_j = m_j C_{pj} dT_j/dt \qquad (1)$$

$$\Sigma W_j = \sum_{i=1}^{j}\left\{\phi_{ij}C_{ij}A_j\left(T_j^4 - T_i^4\right) + h_m A_j\left(T_j - T_i\right) + \left(K_m A_j/\ell_{ji}\right)\left(T_j - T_i\right)\right\} \qquad (2)$$

$$C_{ij} = \varepsilon_i \varepsilon_j \sigma$$

where:

j = 1,2,3 ... different masses/equations under investigation.

T_j = temperature of mass j

m_j = mass j
C_{pj} = specific heat of mass j
t = time
A_j = area of mass j
φ_{ij} = view factor between the masses ij
ε_j = emissivity of mass j
σ = Stefan-Bolzman constant
h_m = convection coefficient
K_m = conduction coefficient
ℓ_{ji} = conduction distance between masses i, j.

The SI system of units is used throughout the model.

The above set of equations (1) is solved in a step by step approach. The solution is given by the following equation.

$$T_{tj} = T_{t-1,j} + \Sigma W_j \, \Delta t/(m_j C_{pj}) \qquad (3)$$

where

T_{tj}, $T_{t-1,j}$ = the temperatures of mass j at time steps t and t-1 respectively.

Δt = Integration time step. This is small to avoid numerical instabilities but long enough for the water to go through the system.

If there is a fluid (water, gas) running through a mass, then the set of equations (1) takes the form

$$m_j C_{pj} \frac{dT_j}{dt} \pm \Sigma W_j = \dot{M} C_p' dT' = KAdT'' \qquad (4)$$

where

$\dot{M}$ = fluid flow rate
C_p' = specific heat of fluid
KA = characteristic of the heat exchanger
dT_j, dT', dT'' = temperature differences in component, fluid and heat exchanger respectively.

Equations (3) are also solved in a step by step approach assuming that in a small time step Δt, the temperature of the component is constant. Under these conditions the solution is given by the following equations.

$$\dot{M} C_p (T'_{t+1} - T'_{in}) = KA(T_t - (T'_{t+1} + T'_{in})/2) \qquad (5)$$

$$m_j C_{pj} (T_t - T_{t+1}) \pm \Delta t \Sigma W_j = \dot{M} C_p (T_{t'+1} - T'_{in}) \Delta t \qquad (6)$$

where

T'_{in} = fluid input temperature.

Equation (2) indicates that all three forms of heat transfer are taken into account. Radiation dominates only when there is good vacuum ($< 10^{-3}$mbar) Conduction is considered when there is physical contact or, in case of an air leak, when the Grashof number is less than 10^7. Convection starts when this number is higher and for the geometry involved this corresponds to approximately 2mbar. Typical conduction and convection parameters are: 0.026W/(mK) for gas nitrogen at 20°C and 5.5W/(m^2K) with 1bar air leak between chevrons (at 77K) and target shields (at 520K). In the analysis several assumptions have been made:

- The thermal mass and temperature of the graphite and inconel in the target modules are constant and unaffected by freeze-up scenarios. This is because the thermal mass of the graphite (0.9MJ/°C, 1200kg) and of inconel (1.28MJ/°C, 1300kg) are large compared to the thermal mass of the cryopump (26MJ, 330kg) for the full temperature excursion of 77K to 280K. In addition, the thermal mass of the hot air in case of air leaks again is small and does not affect significantly the inconel and graphite temperatures.
- Following water introduction into cooled components inside the machine, the temperatures of graphite and inconel are different. Graphite is always at higher temperatures (165°C when the vacuum vessel is at 250°C) than the water cooled inconel which is at ~20°C. In the event of an air leak in the machine, these temperatures tend to equalise but again during the phenomena under investigation, we assume that these temperatures do not alter. This was because it can be shown that the time constants of these events are different. It takes longer for the temperatures of graphite and inconel to equalise than for instance the onset of freezing.

It should be noted that in applying these models to simulate the abnormal operating scenarios of the cryopump, we experienced difficulties in some of the input parameters. These are:

- Empirical convection coefficients at low pressures. (It was assumed that the Nusselt number is proportional to the square root of pressure.)
- Characteristics of the heat exchangers. These, for instance, depend on many parameters some of them ill defined (ie, brazing joints).
- The dynamic flow and temperature of the gas inside the tokamak during air leaks.

4. EXPERIMENTAL MODEL VALIDATION

A series of experiments was undertaken to validate the model. Figure 2 gives comparisons between theory and experiment of cooling the baffles with ambient gas nitrogen from a vessel temperature of 220°C. The agreement is good.

Figure 3 shows also good comparison between theory and experiment in warm-up of a cryopump quadrant in a test tank.

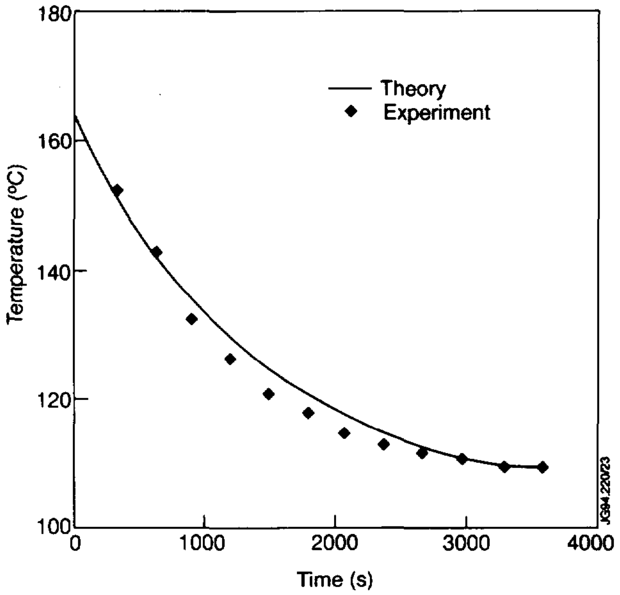

Figure 2. Exit temperatures of gas nitrogen during cooldown of baffles from 220°C.

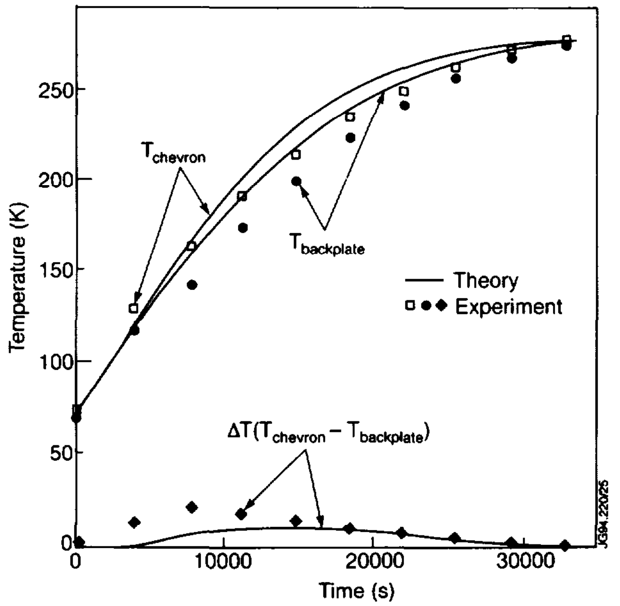

Figure 3. Cryopump warm-up in a vacuum test tank with good vacuum.

5. RESULTS

Figure 4 gives the predicted temperature evolution of the baffles, chevrons and backplate during warm-up of the cryopump with bad vacuum vessel at 250°C, stop of LN flow and loss of water flow. The temperature difference between backplate and chevrons of less than 150°C is allowable.

The model shows that at vessel temperatures above 100°C, no freezing of water in the baffles is possible if there is loss of flow.

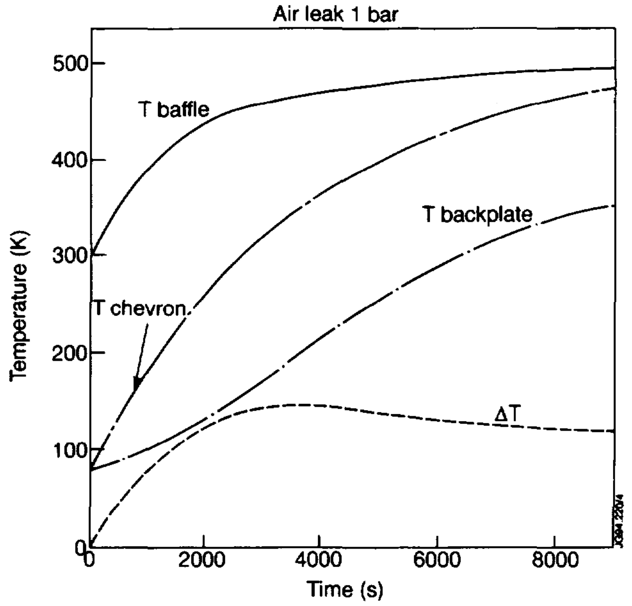

Figure 4. Temperature evolution in the cryopump with bad vacuum, vessel hot (250°C), loss of water flow and stop of LN flow.

Figure 5 attempts to analyse the risk of freezing of target shields, as a function of vessel temperature in the event of stoppage of the water flow. It is assumed that the water enters the system at 300 K (27 °C). The effect of the high temperature Carbon (C) or Beryllium (Be) tiles is demonstrated. The low emissivity Be results in higher probability of freezing since it protects less than C the target shields even with loss of vacuum when convection (not radiation) dominates the heat transfer. In addition Figure 5 shows the effect of different heat transfer assumptions, with regard to the gas dynamic mechamisms of the air leak, on the model predictions. It is shown that the freeze-up risk of the target shields is high even at elevated vessel temperatures. For instance with a vessel temperature of 250 °C (523 K), and inlet water temperature of 18 °C the model predicts freeze-up of the shields under any assumption.

With good vacuum however no risk of freezing the target shield exists provided the vessel temperature is above 400 K.

As far as boiling of water inside the system (with abnormal operation, loss of water flow and vessel hot) is concerned, it was shown that the time constants of such an event are much longer than the time constants of re-instating flow or draining. Thus pressurisation of the pipework due to water boiling is not possible.

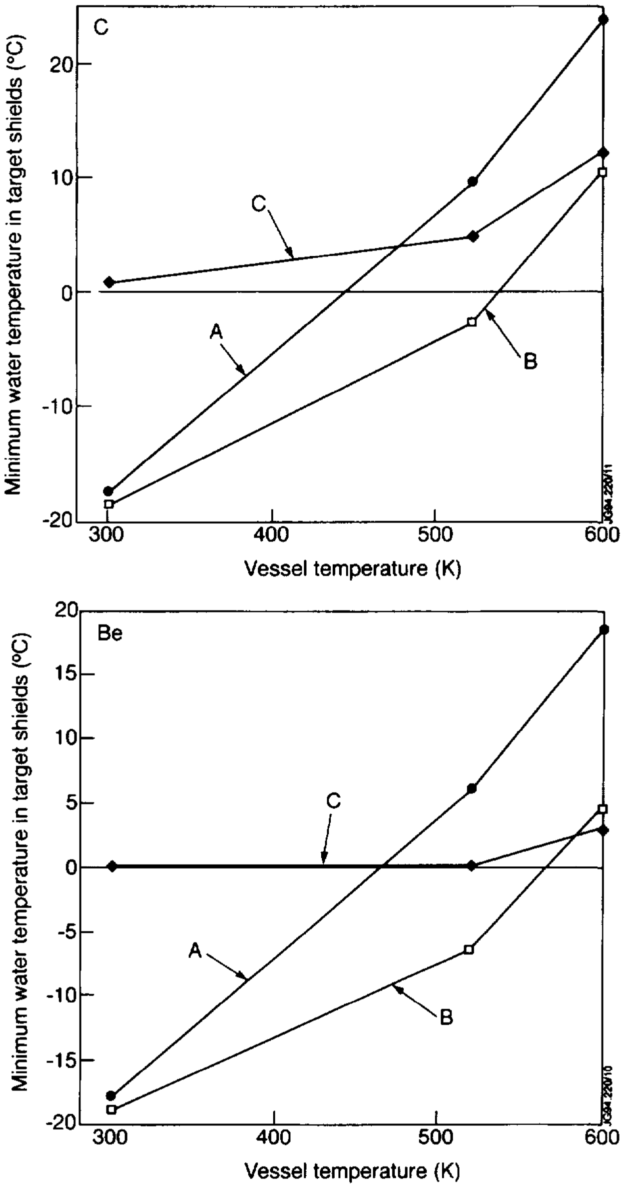

Figure 5. Minimum target shield temperatures as a function of vessel temperature and different assumptions in the heat transfer modes.

A: Optimistic assumptions (hot air at the rear of the target shields), 1 bar air leak.
B: Realistic, 1 bar air leak.
C: 15 mbar air leak.

6. CONCLUSIONS

The analysis resulted in guidelines to protect the cryopump and the associated in-vessel components against abnormal scenarios. These are:

- No cooldown of the cryopump without water inside the internal components of the vacuum vessel, while the vessel temperature is above 100°C.
- No freeze-up risk of baffles with vessel temperature above 100°C.
- No freeze-up risk of the target shields with good vacuum and vessel temperature higher than 400 K.
- The risk of freezing the target shields with loss of vacuum is appreciable even at high vessel temperatures. Thus an effective gas flushing system* is triggered to remove water from pipework in such an event.
- No risks occur due to boiling of the cooling water.

REFERENCES

1. W. Obert, S. Papastergiou, JET Pumped Divertor Cryopump, proceedings Symposium of Fusion Technology, London, UK (1990) 488-492.
2. S. Papastergiou, W. Obert and E. Thompson, Material Selection, Qualification and Manufacturing of the In-vessel Divertor Cryopump for JET, proceedings of Inter. Cryogenic Mater. Conf., Albuquerque, USA (1993) to appear.

ACKNOWLEDGMENT

The experimental data of cooling the baffles with nitrogen, used in Figure 2, were provided by M Cooke and P Butcher of the JET Machine Services Group.

---

* In designing such gas flushing system, care has to be taken to account for possible parallel loops in the pipework. These may result in residual stagnant water, being present in pipes with high hydraulic resistance, despite the gas flushing. Such water pockets are at risk of freezing.