HELIUM REFRIGERATOR DESIGN FOR PULSED HEAT LOAD IN TOKAMAKS

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ABSTRACT

Nuclear fusion reactors of the Tokamak type will be operated in a pulsed mode requiring the helium refrigerator to remove periodically large heat loads in time steps of approximately one hour. What are the necessary steps for a refrigerator to cope with such load variations?

A series of numerical simulations has been performed indicating the possibility of an active refrigerator control with low exergetic losses. A basic comparison is made between the largest existing refrigerator sizes and the size required to service for example the ITER requirements.

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INTRODUCTION

In most of all cryogenic applications today an almost constant heat load is carried by the refrigerator. Load variations, required by the experiment, are achieved slowly and in smaller applications the load variations are often compensated by electrical heaters.

The main design criteria for helium refrigerators and liquefiers are performance and exergy efficiency. Hardly achievable specifications for transient modes are unusual. This schema will change with the refrigeration requirements for the future Tokamak fusion reactors. The Tokamak will be operated in a periodic process with a cycle time of about one hour. Peak load periods will alternate with low load periods. During the low load periods approximately half of the peak load capacity will be required.

To cope with the task, a refrigerator has to support rapidly increasing heat loads with maintainable exergetic losses. The power input to these refrigerators will be huge, thus a

TABLE 1. The estimated refrigeration requirement for ITER compared to existing refrigerator capacities.

Plant / Laboratory	No of units	4.5 K equivalent capacity per unit
DESY	3	5 kW
W7X	1	7 kW
CERN – LEP	4	12 kW
CERN – LHC	8	18 kW
ORNL - SNS	1	18 kW
ITER	3	24 kW

compensating heater to smooth the load variations is an immense waste of energy and consequently not an adequate solution. In TABLE 1 the estimated refrigeration requirement for ITER is compared to existing refrigerator capacities. To comply with the Tokamak requirements powerful and associated control mechanisms need to be developed. Prior to these control mechanisms it will be necessary to perform dynamic process simulations of refrigerators. Linde Kryotechnik AG has initiated the first steps.

DYNAMIC PROCESS CALCULATIONS

As a first step to meet with the use of dynamic process simulations a rough-and-ready computer program, based on an Excel spreadsheet, has been established. Using some simplifying suppositions, this program can illustrate the time flow of transient modes of a refrigerator which is equipped with several turbines at different temperature levels. Some basic properties of this program are outlined below:

- For data modelling the heat exchangers are divided into elements. Each element has its individual temperature value of the solid material. It is exchanging heat with the joining flows (FIGURE 1).
- At the interface points between the elements the temperature values of each of the flows are calculated by using the set of equations 1, whereby T means the temperature [K], the subscripts HP, MP and LP indicate the three different flows, HX indicates the solid and the variable subscript "i" indicates the element number. These equations perform exactly 0.5 transfer units per element.
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$$T_{HP,i} = \frac{2 \cdot T_{HP,i-1} + T_{HX,i}}{3} \qquad T_{MP,i-1} = \frac{2 \cdot T_{MP,i} + T_{HX,i}}{3} \qquad T_{LP,i-1} = \frac{2 \cdot T_{LP,i} + T_{HX,i}}{3} \tag{1}$$

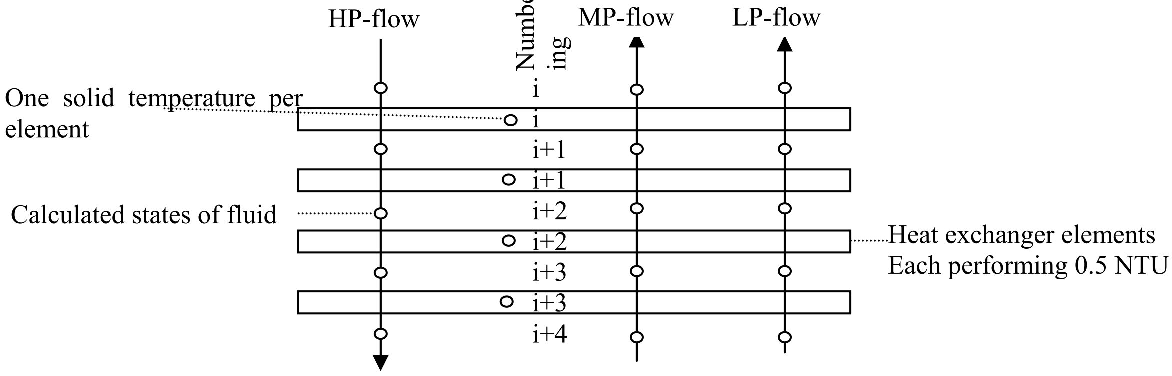

FIGURE 1. Data modeling for dynamic process calculations in heat exchangers.

- The temperature of the elements changes with the quotient of the residual heat load from the flows and the heat capacity of the element. This is calculated with equation (2), whereby T means the Temperature [K], E the Enthalpy [J/kg], M the mass [kg], m^* the mass flow [kg/s], ρ the fluid density [kg/m^3], cp the specific heat [J/kg/s], V the volume of the flow passages[m^3] and t the time [s].

$$\frac{dT_{HX,i}}{dt}=\frac{m^*_{HP,i-1}\cdot\left(E_{HP,i}-E_{HP,i-1}\right)+m^*_{MP,i-1}\cdot\left(E_{MP,i}-E_{MP,i-1}\right)+m^*_{LP,i-1}\cdot\left(E_{LP,i}-E_{LP,i-1}\right)}{M_{HX,i}\cdot cp_{HX,i}+V_{HP,i}\cdot\rho_{HP,i}\cdot cp_{HP,i}+V_{MP,i}\cdot\rho_{MP,i}\cdot cp_{MP,i}+V_{LP,i}\cdot\rho_{LP,i}\cdot cp_{LP,i}} \quad (2)$$

- The 3rd group of equations, with variables and subscripts as in equations (1) and (2), performs the conservation of mass. The changing fluid density in (3) is calculated by equation (4), where P_T and P_ρ are partial derivations from the pressure with respect to temperature and density.

$$\begin{aligned}
m^*_{HP,i} &= m^*_{HP,i-1}-V_{HP,i}\cdot\frac{1}{2}\cdot\left\{\frac{d\rho_{HP,i}}{dt}+\frac{d\rho_{HP,i-1}}{dt}\right\}\\
m^*_{MP,i-1} &= m^*_{MP,i}-V_{HP,i}\cdot\frac{1}{2}\cdot\left\{\frac{d\rho_{MP,i}}{dt}+\frac{d\rho_{MP,i-1}}{dt}\right\}\\
m^*_{LP,i-1} &= m^*_{LP,i}-V_{LP,i}\cdot\frac{1}{2}\cdot\left\{\frac{d\rho_{LP,i}}{dt}+\frac{d\rho_{LP,i-1}}{dt}\right\}
\end{aligned} \quad (3)$$

$$\frac{d\rho}{dt}=\frac{1}{P_\rho}\cdot\left\{\frac{dP}{dt}-P_T\cdot\frac{dT}{dt}\right\} \quad (4)$$

- The program applies the real gas properties for helium from NIST[1] and real temperature dependant values for the specific heat of the heat exchanger material.
- The time step for the process simulation is 0.1 second. Ten successive time steps are proceeded with constant values for cp, P_T and P_ρ. Then, consequently once per second of process time, the fluid and material properties are recalculated.

The simplifying assumptions are:

- The pressure drops of the flows are constant
- The heat transfer numbers between the fluids and the heat exchanger are constant
- The turbines have a constant polytropic efficiency
- The turbine mass flows are calculated like the flow through a nozzle
- The deviations in the compressors capacity control are not considered

Before presenting the results of computing, the simulated refrigerator is considered and explained.

THE VIRTUAL REFRIGERATOR

The process which has been selected for this simulation is neither the process of an existing plant nor the proposed process solution for a new refrigerator project. It has merely been sized for the purpose of this study and for this reason only one refrigeration consumer is supported and the pre-cooling with liquid nitrogen is not foreseen.

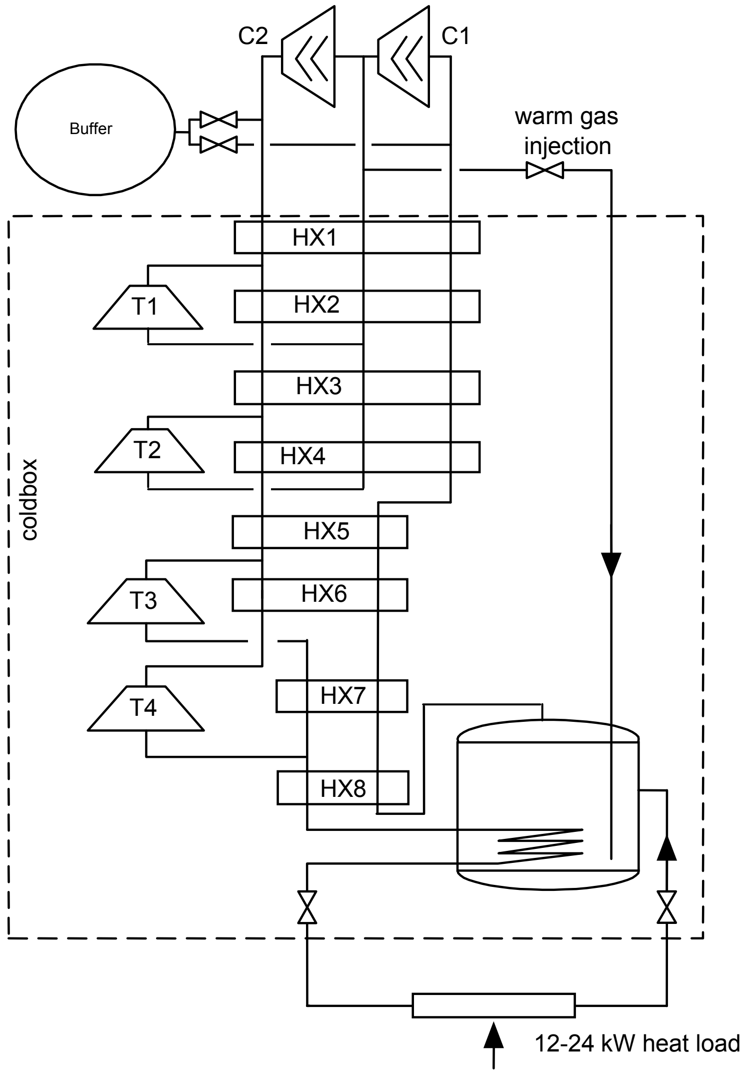

FIGURE 2. Flow scheme of the refrigerator.

Suppositions:	Heat exchangers:	14 tons / 145 ntu	
	Polytropic turbine efficiencies:	80%,80%,77%,75%	
	Capacity of the compressors:	1^{st} stage	1320 g/s at 103 kPa
		2^{nd} stage	1580 g/s at 400 kPa

Refrigeration is performed by a supercritical helium flow which leaves the coldbox with approximately 400 kPa and 4.5 K and which returns to it with an enthalpy value near to saturated vapor. The flow after return is throttled to the low pressure level of 125 kPa. At steady state operation the capacity is 18 kW, but boosted with the supply of liquid helium the refrigerator can perform up to 24 kW during a limited period of time. It is supposed that the used liquid helium is re-liquefied in low load periods by a heat load of about 12 kW and that the helium inventory over the timeframe of one operation loop of the Tokamak is balanced.

The HP-supply pressure to the coldbox as well as the LP-return pressure from the coldbox are kept constant. The MP-return pressure is floating, which is defined by the constant volume flow to the second stage compressor. Basically the MP-return pressure is low during the low load periods and high during the high load periods.

During the transient mode from a 12 kW heat load with liquefaction to a liquid helium supported 24 kW heat load, the temperature of the process equipment, like heat exchangers

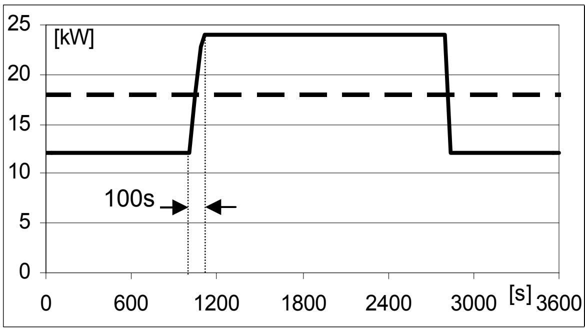

FIGURE 3. The applied heat load to the refrigerator versus time. This load progression is repeated in time loops of one hour. The load increase from 12 to 24 kW takes 100 s. The drop of load from 24 to 12 kW is done in 1 s.
This heat load curve was input to the dynamic process simulation. The dotted line indicates the designed steady load of 18 kW.

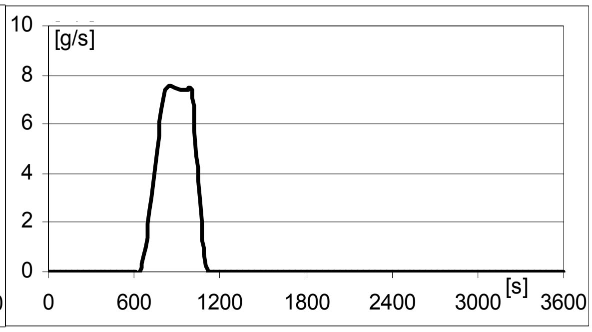

FIGURE 4. Warm gas injection to the liquid helium versus time. The warm gas injection is started 300 s prior to the load increase. First it is ramped on with 0.05 g/s, then it is limited by maintaining a low pressure flow of 1320 g/s. By this restriction it reduces automatically with the rising heat load. It is stopped as soon as maximal heat load is reached.

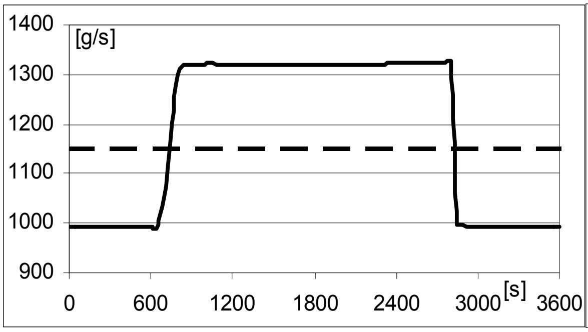

FIGURE 5. The low pressure flow versus time. The flow starts increasing with the start of the warm gas injection. It keeps high during the high load period. The capacity control of the 1st stage compressor was not yet part of the study. The dotted line indicates the flow at steady stade conditions with 18 kW heat load.

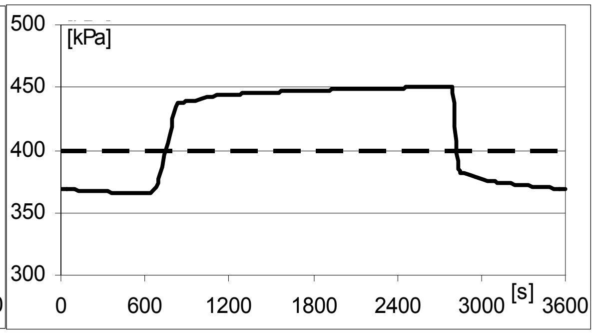

FIGURE 6. The inter stage pressure versus time. It is supposed that there is no active control of this pressure. The pressure is reached by the given flow volume of the second stage compressor. At a constant load of 18 kW this pressure is 400 kPa.

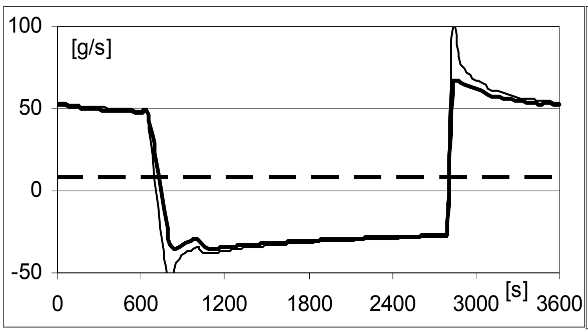

FIGURE 7. The gas exchange with the buffer (bold) and the liquefaction rate (thin) versus time. The gas exchange flow is never more than 4% of the total circulating mass flow. The dotted line indicates the liquefaction rate at steady stade conditions with 18 kW heat load.

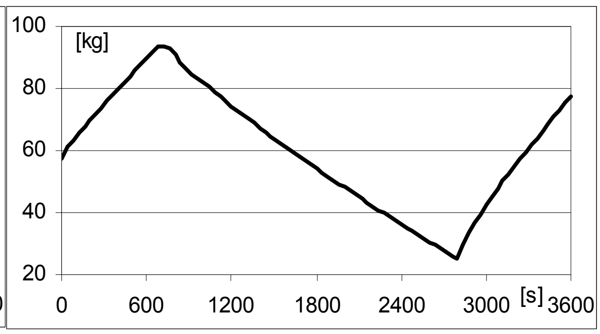

FIGURE 8. The liquid helium inventory in the liquid helium separator versus time of the coldbox. This curve is the integral of the liquefaction curve in Figure 7.

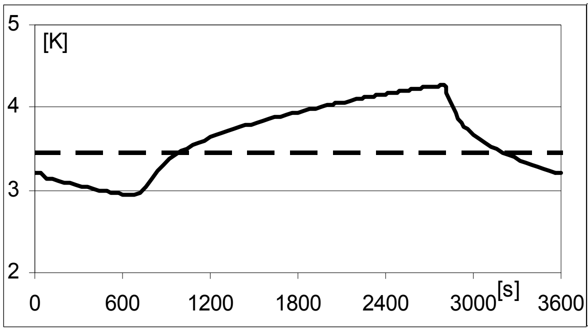

FIGURE 9. The warm end temperature difference versus time.

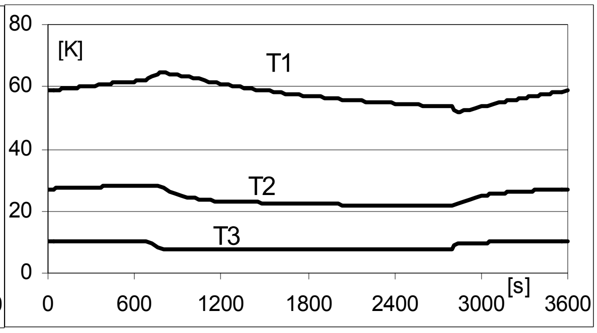

FIGURE 10. The temperatures of turbines T1 to T3 versus time.

and turbines, have to be lowered. The extracted heat appears as additional load to the system and constrains the maximal rate of increase in external heat load. Exceeding this maximal rate of increase results in a high evaporation rate of liquid helium from the separator attended by an overloaded 1st stage compressor unit.

To obtain a higher rate of load increase the refrigerators temperatures have to be lowered prior to an expected external heat production. The most effective way to achieve this is to inject a small flow of warm gas to the liquid helium separator. This warm gas allows a larger flow of helium to be evaporated which then produces refrigeration in all the heat exchangers. Unlike electrical heaters, the warm gas injection has no thermal inertia. Its response time is consequently very short. When the external heat load starts increasing, the warm gas injection reduces well-regulated back to zero.

During the transient mode from the high load to the low load operation the coldness of the heat exchangers performs temporary excessive liquefaction and the low pressure flow reduces considerably. In a real plant, resulting pressure fluctuations would require speed reduction of the turbines prior to the drop of the heat load.

SIMULATION RESULTS

The operating schedule for the periods of heat loads and the starts and stops of the controlled warm gas injection has been as follows:

12 kW load during	1749 s	
12 to 24 kW load increase within	100 s	
24 kW load during	1750 s	
Drop from 24 to 12 kW within	1 s	
Start of warm gas injection has been	350 s	prior to the start of increasing load
Stop of warm gas injection has been	10 s	after end of load increase

Before the computer simulation has been started the temperatures of the flows and the heat exchangers have been initialized with the temperature values of the 12 kW steady state operation. From these starting conditions to the almost repetitive behaviour of the characteristic process parameters which is plotted in the figures 3 to 10, the computer has stepped through 8 loops, thus 8 hours of operation time. This has taken approximately two hours of computing time.

Interesting is the development of the residual liquid helium production (respectively consumption) after each loop (FIGURE 11). It shows a pointwise convergence to a constant value.

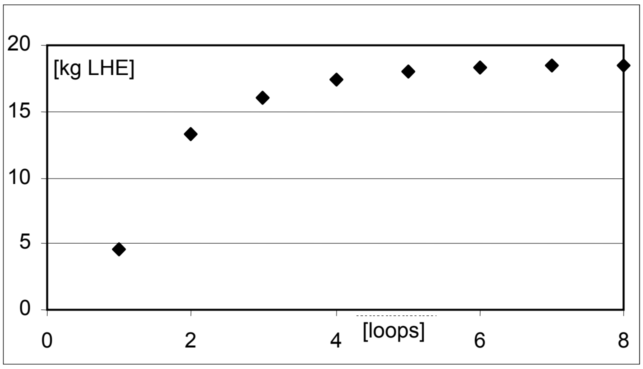

FIGURE 11. Residual Liquefaction per loop after startup from a steady state 12 kW heat load.

The average exergy efficiency of the coldbox over one repetitive loop is 50.8 %. This value may be low compared to the exergy efficiency values of the same refrigerator in steady state operation which are 57.1% at 12 kW, 56.1% at 18 kW and 51.2% at 24 kW. By using a compensation heater and operating continuously at 24 kW the efficiency would be in the range of 40 to 45%.

CONCLUSIONS

The operating performance of a helium refrigerator at the specific conditions of periodic heat load variations has been simulated. Peak load periods of 24 kW at 4.5 K have taken turns with low load periods of 12 kW at the same temperature.

The supposed refrigerator has been equipped with four turbines and with heat exchangers of a total of 145 NTU. To perform a higher exergy efficiency a real refrigerator of this capacity would be equipped with more turbines and with increased heat exchanger surfaces. However, within the scope of this study the selected process equipment is a feasible choice. More equipment would have required more data points and consequently more computing time, an adequately sophisticated computer program and a more powerful computer.

The objective of this study is in a first step the development of refrigerators which can sustain the pulsed and rapidly increasing heat loads of the future Tokamak fusion reactors. The results are encouraging. Nevertheless, further investigations are necessary, first in the development of improved process simulation programs, then in the control of the compressor units when three or four parallel operating coldboxes are interacting.

REFERENCE

1. Physical Properties of Helium, Technical Note 1334, US Department of Commerce / National Institute of Standards (NIST).