

OPERATION OF THE HE-LIQUEFIER OF THE AGOR CYCLOTRON

L. P. ROOBOL, S. BRANDENBURG, H. W. SCHREUDER, AND S. VAN DER VEEN

Kernfysisch Versneller Instituut, Zernikelaan 25, NL-9747 AA Groningen, the Netherlands

Due to a cryogenic leak in the main cryostat, the cooling power required for the operation of the AGOR cryogenic system is significantly larger than originally expected. Although the liquefier has been upgraded substantially, it has to operate at maximum capacity under certain conditions. Measurements indicate that the maximum capacity is well below the rated capacity. A thermodynamic model of the liquefier has been developed to gain a better insight in the operation of the system. Due to a lack of diagnostics in the liquefier no detailed comparison between model and liquefier is possible, yet clear discrepancies are found.

1 Introduction

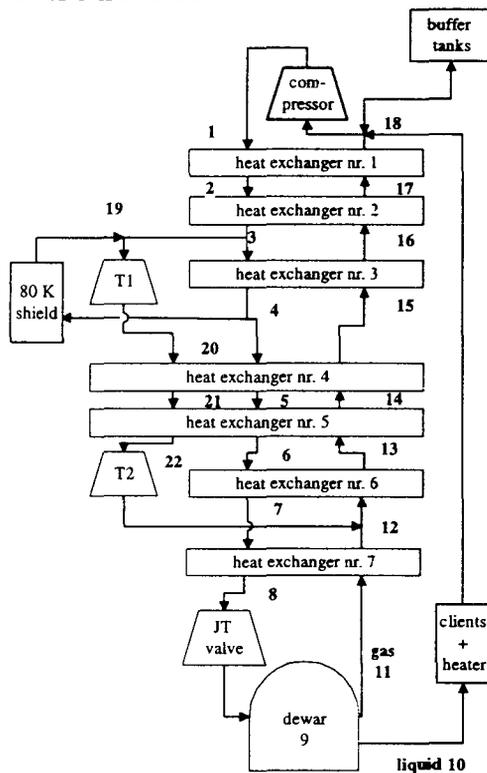


Figure 1: Diagramme of the liquefier.

Since AGOR only is a superconducting cyclotron below 10 K (the approximate critical temperature of the magnet coils, made from NbTi), the main magnet coils need to be cooled down to cryogenic temperatures. The same is true for the second electromagnetic bending channel, EMC2, and the quadrupole lens at the exit of the cyclotron.

For this a liquefaction plant, a TF50, has been bought from Linde, see figure 1 for a block diagramme. It consists of three compressors, taking the helium pressure from slightly above ambient up to 13 bar, with a mass flow of up to 55 g/s. A part of this gas stream is led through two turbines connected in series, over which the

gas is expanded and cools. This cooling power is used to cool the incoming gas stream with a series of heat exchangers. The remaining gas, being precooled to around 12 K, enters the Joule Thomson stage, where it expands over a needle valve, and transforms into a mixture of cold gas and liquid. This mixture is fed into a storage dewar, where the liquid remains. The evaporating gas goes back through the heat exchangers into the compressor inlet line.

Because of the large number of parameters in this system, it is necessary to make a physical model of the liquefier plant in order to make predictions about its performance, and about the role which the individual parameters play in the cooling process. The following article describes the development of such a model, and some of the conclusions derived from it.

The factory specifications of the setup are: 660 W of non-isothermal refrigeration around 80 K, 55 W of isothermal refrigeration at 4 K, and, additionally, a liquid supply of 2.2 g/s, around 65 l/h, depending on pressure. These numbers are given under the constraint of a constant liquid level in the storage dewar.

2 Thermodynamics

The thermodynamic potentials of interest are, in our case, the enthalpy H and the entropy S . The enthalpy is defined as

$$\begin{aligned} H &= U + pV + \frac{1}{2}mv^2 \\ dh &= du + d\left(\frac{p}{\rho}\right) + vdv \\ &= Tds + \frac{dp}{\rho} + vdv \end{aligned}$$

with p the pressure, V the volume, m the mass, v the speed of the flow, h the specific enthalpy, u the specific

internal energy, ρ the mass density, T the temperature, and s the specific entropy.

2.1 Compressors

The helium gas enters the compressors at $h_{in}(p_{in}, T_{in})$, and leaves at $h_{out}(p_{out}, T_{out})$, which means the work done is equal to

$$\dot{W} = \dot{m}(h_{in} - h_{out})$$

2.2 Heat exchangers

An ideal counterflow heat exchanger, as depicted in Figure 2, just shifts heat between the upwards and downwards stream. In practice, there will always be some entropy production, which can be modelled by a heat leak \dot{Q} . Also, the flow might not be balanced in the sense that the mass flow in the two streams is not the same. The enthalpy balance becomes

$$\dot{m}_{1 \rightarrow 2}(h_2 - h_1) + \dot{Q} = \dot{m}_{3 \rightarrow 4}(h_4 - h_3)$$

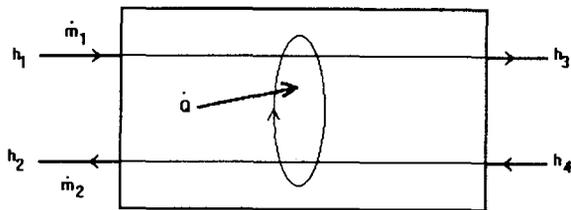


Figure 2: An ideal heat exchanger.

2.3 Joule Thomson valve

The Joule Thomson expander is a (needle) valve over which gas is expanded in such a way that at either side of the valve the pressure is constant, while the gas does not pick up any significant speed in the process. The gas then is in equilibrium at both sides of the valve, from all of which it follows that $\Delta H = 0$: the expansion is isenthalpic. The temperature change due to this effect is

$$\left(\frac{\Delta T}{\Delta p}\right) = \int_{p_1}^{p_2} \frac{1}{C_p} \left[T \left(\frac{\partial v}{\partial T}\right)_p - v \right] dp$$

The sign of this effect can be either positive or negative: below the inversion temperature $T_{inv} = v \left(\frac{\partial v}{\partial T}\right)_p$, around 35 K for helium, cooling is obtained.

2.4 Dewar: refrigeration vs. liquefaction

There are two types of energy drains from the storage dewar:

- Demand for Liquid: A mass flow of $\dot{m}_{extracted}$ is taken from the dewar, and only returned to the system at room temperature.
- Demand for Cooling: At the same time, some liquid is taken from the dewar to cool the main magnet. This liquid evaporates, and the cold gas is returned to the system. In other words, this can be modeled by a refrigeration load at 4 kelvin: $\dot{Q}_{refrigeration}$.

The enthalpy balance becomes

$$\begin{aligned} \dot{Q}_{dewar} &= \dot{m}_{JT} (h_{gas,4K} - h_{JT}) \\ &= \dot{m}_{extracted} (h_{gas,300K} - h_{liquid}) + \dot{Q}_{refrigeration} \end{aligned}$$

where \dot{m}_{JT} is the mass flow through the Joule Thomson valve, and h_{JT} the entropy of the gas in that stream. This describes the static situation, in which the liquid level in the dewar stays at constant height.

From the last equation it is clear that extracting liquid is very enthalpy costly, about 1.5 kJ/g, while refrigeration only takes 20 J/g. From entropy considerations it follows that withdrawing 1 litre of liquid per hour diminishes the cooling power of the liquefier by somewhat more than 4 W, the precise number depends on the pressure in the dewar.

3 Turbines

The expansion machines used to precool the incoming gas stream are turbines, which let the gas expand adiabatically over a nozzle wheel. To keep the energy constant, the kinetic energy of the gas increases, and is transferred to a spinning wheel with fins on it, the rotor wheel. The energy picked up by the rotor wheel is dissipated at room temperature in a brake circuit, see figure 3.

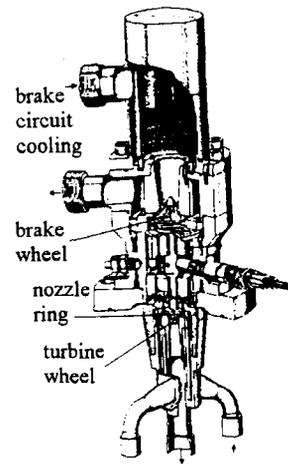


Figure 3: View of the inside of the turbines.

The mass flux density j increases with the speed of the gas, until it reaches a maximum j_{max} at the speed of sound. Then it decreases again, to vanish at some supersonic velocity. For the turbine, this means that the largest mass flow which can be obtained is given by

$$\dot{m}_{max} = j_{max} A_{min} \propto \sqrt{p\rho} \propto \frac{p}{\sqrt{T}}$$

with A_{min} the smallest cross section of the nozzle wheel. Writing $\gamma = c_p/c_v$, with $c_{p(v)}$ the heat capacities at constant pressure (volume), it is possible to obtain expressions for the pressure- and temperature drop over the turbine at the point of maximum throughput:

$$\frac{p_{out}}{p_{in}} = \left[\frac{2}{\gamma + 1} \right]^{\gamma/(\gamma-1)} = 0.487$$

$$\frac{T_{out}}{T_{in}} = \frac{2}{\gamma + 1} = 0.75$$

The numbers at the right hand side of the equation are calculated for an ideal mono-atomic gas, for which $\gamma = \frac{5}{3}$.

For a non-ideal turbine, in which entropy is produced, the enthalpy difference between inlet and outlet will be less than calculated above. One defines the turbine efficiency η_T through $\Delta h_{actual} = \eta_T \Delta h_{isentropic}$. Typical values for the efficiency are $\eta \approx 0.6 - 0.8$. Taking the efficiency into account, we get for the temperature drop at maximum throughput:

$$\left(\frac{T_{out}}{T_{in}} \right) = 1 - \eta_T \frac{\gamma - 1}{\gamma + 1} = 1 - \frac{1}{4} \eta_T$$

Where the last "=" sign again is valid for the ideal gas only.

Another important parameter for the turbine is $x = v_{rotor}/v_{gas}$, the ratio of the rotor speed to the speed of the gas. Because of the (constant) friction in the shaft of the rotor, higher speeds produce relatively less entropy than low speeds. At high speeds, however, the friction between the gas and the rotor blades becomes sizeable, growing as x^3 . Hence there will be an optimal value for x , the precise value depending on the design of the rotor.

From the definition of the enthalpy it follows that the maximum speed of the gas is $v_{max} = \sqrt{2\Delta h}$, so that in order to keep the value for x at its optimum, the revolution frequency should follow the enthalpy drop over the turbine:

$$\omega_{rotor} \propto \sqrt{\Delta h}$$

4 The model

From thermodynamic considerations, applied to the liquefier setup with its clients as a whole, it follows that

$$\dot{Q}_{turbines} - \dot{W}_{compressor} - \dot{Q}_{heatleaks} = \dot{Q}_{refrigeration} + \dot{m}_{liquid} (h_{ambient} - h_{liquid})$$

which simply means that the combined work done by the turbines and the compressor, should equal the sum of the refrigeration power, the heat leaks and the energy cost of removing liquid at 4 K while returning gas at ambient temperature.

Taking the compressor power and the heat leaks as design constants, we can still optimize the liquefier performance using the turbine power, given the refrigeration need of the clients. Clearly, the more work is done by the turbines, the more cooling power is produced.

The turbine throughput can be written as $\dot{m} = \beta p_{in} / \sqrt{T_{in}}$, with β an empirical constant. For a given mass flow, temperature and pressure at the exit of the second turbine, we can work our way through the whole chain, and determine the conditions at the inlet of the first turbine.

The temperature at the exit of the turbine circuit determines the efficiency of the Joule Thomson stage, the optimum differs with the type of cooling needed. The exit pressure is more or less equal to the turbine inlet pressure, which means that we essentially have two variables left: the temperature difference ΔT across the last heat exchanger, and the branching ratio

$$z = \dot{m}_{Joule-Thomson} / \dot{m}_{compressors}$$

We assume some reasonable value for the temperature difference, $\Delta T = 0.15K$, and take z as an optimizing parameter, to be adjusted such that the cooling power be as large as possible. There should exist an optimum, because if $z = 1$, there would be a large mass flow through the Joule Thomson stage, but it would not be precooled. On the other hand, if z were close to 0, the gas would be extremely cold, but there would be nearly no gas to liquefy.

5 Conclusions

It has proven possible to calculate most properties of the AGOR cryoplant by using thermodynamic and hydrodynamic principles. The only ad hoc assumption we had to make was for the temperature difference ΔT across the last heat exchanger. All other "free" parameters are related to the design of the subsystems, and have been extracted from data sheets supplied by the manufacturer or found empirically.

It is quite difficult to make a detailed comparison between the predictions made by the model and experiment, because our diagnostic tools are not adequate for that purpose. For example, we can only measure the total mass flow through the compressors, and not the branching ratio z . Also, we lack thermometers at some crucial points in the set-up.

The model agrees with the cooling power quoted by the manufacturer, but empirical data show a discrepancy: the turbine- and compressor throughput are considerably (10 - 20 %) lower than specified. This would not be a problem if we did not have a cryogenic leak in the magnet dewar. When active, the leak doubles the heat load at 4 K from 50 to 100 W (about 25 W per 10^{-5} mbar pressure increase), which is just too much for the cryoplant to handle, according to both calculation and experiment.

Fortunately, the leak is not very active at all: only after temperature excursions, caused e.g. by liquefier down-time (regeneration), it appears. For unknown reasons, the time for which it is active has decreased from several days in 1997 to less than a day during recent months.

References

- [1] L. D. Landau and E. M. Lifschitz, *Fluid Mechanics*, (Pergamon Press, Oxford, U.K., 1986).
- [2] C. J. Adkins, *Equilibrium Thermodynamics*, (Cambridge University Press, Cambridge, U.K., 1983).
- [3] Hartmut Frey und René A. Haefler, *Tieftemperaturtechnologie*, (VDI-Verlag GmbH, Düsseldorf, Germany, 1981).
- [4] NIST technical report PB90-183351, *Thermophysical properties of helium-4 from 0.8 to 1500 K with pressures to 2000 MPa*, (NIST, Washington DC, U.S.A., nov 1989).