Neon helium mixtures as a refrigerant for the FCC beam screen cooling: comparison of cycle design options

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Neon helium mixtures as a refrigerant for the FCC beam screen cooling: comparison of cycle design options

S Kloeppel[1], H Quack[1], C Haberstroh[1], F Holdener[2]

[1] Technische Universitaet Dresden, Bitzer-Stiftungsprofessur fuer Kaelte-, Kryo- und Kompressorentechnik, 01062 Dresden, Germany
[2] Shirokuma GmbH, 8620 Wetzikon, Switzerland

E-mail: `steffen.kloeppel@tu-dresden.de`

Abstract. In the course of the studies for the next generation particle accelerators, in this case the Future Circular Collider for hadron-hadron interaction (FCC-hh), different aspects are being investigated. One of these is the heat load on the beam screen, which results mainly from the synchrotron radiation. In case of the FCC-hh, a heat load of 6 MW is expected. The heat has to be absorbed at 40 to 60 K due to vacuum restrictions. In this range, refrigeration is possible with both helium and neon. Our investigations are focused on a mixed refrigerant of these two components, which combines the advantages of both. Especially promising is the possible substitution of the oil flooded screw compressors by more efficient turbo compressors. This paper investigates different flow schemes and mixture compositions with respect to complexity and efficiency. Furthermore, thermodynamic aspects, e.g. whether to use cold or warm secondary cycle compressors are discussed. Additionally, parameters of the main compressor are established.

1. Introduction

The Future Circular Collider (FCC) study aims to evaluate the possibilities and limits of a new generation particle accelerator. There are different designs under investigation, one being the hadron-hadron collider concept. In this, protons will be accelerated in a ring of 100 km circumference to a center of mass energy of 100 TeV. The high energy of the protons causes the synchrotron radiation, which was a minor effect at the LHC, to become a new challenge. The power of this radiation will add up to 6 MW in total. To reduce the heat input into the cold magnets a beam screen at an intermediate temperature will be applied. This beam screen is situated inside the cold bore of the magnets around the beam. Its temperature is limited by two factors: the overall exergetic balance and vacuum requirements. The higher the beam screen temperature, the lower is the power required for cooling, but the higher is the heat input into the magnet which themselves have to be cooled. An optimum temperature for an ideal refrigeration system would be around 80 K, but due to restrictions from the vacuum and the surface impedance, the cooling has to take place between 40 and 60 K [1]. The beam screen is not the only part of the accelerator that requires cooling in this temperature range. The thermal shields as well as the superconducting current leads require an appropriate cooling, too. The state of the art approach would be a helium refrigerator. In a previous paper the Nelium concept was already described [2]. The main point of this concept is the substitution of the screw compressors by centrifugal compressors, which is made feasible by using a mixture of neon and helium called Nelium. The proposed cycle setup consists of a primary cycle for providing the refrigeration capacity and a secondary cycle for the circulation in the beam screen. Different design options for both cycles are presented in this paper. In addition the basic parameters of the main compressor are defined.

2. Requirements

The parameters of the FCC-hh are subject to change. To facilitate the design of the cooling cycle in the ongoing study, some parameters were specified by CERN as illustrated in Figure 1. The accelerator is divided into twelve segments, each of which will have its own refrigerator. The beam screen of one segment consists of 230 pipes with 5.55 mm inner diameter and 30 m length which are connected in parallel. The thermal shield is cooled by a pipe with 100 mm inner diameter and 16 km length. It is part of the project to propose an optimized geometry for the current leads, therefore the only data provided for them is the total current of 2000 kA per segment.

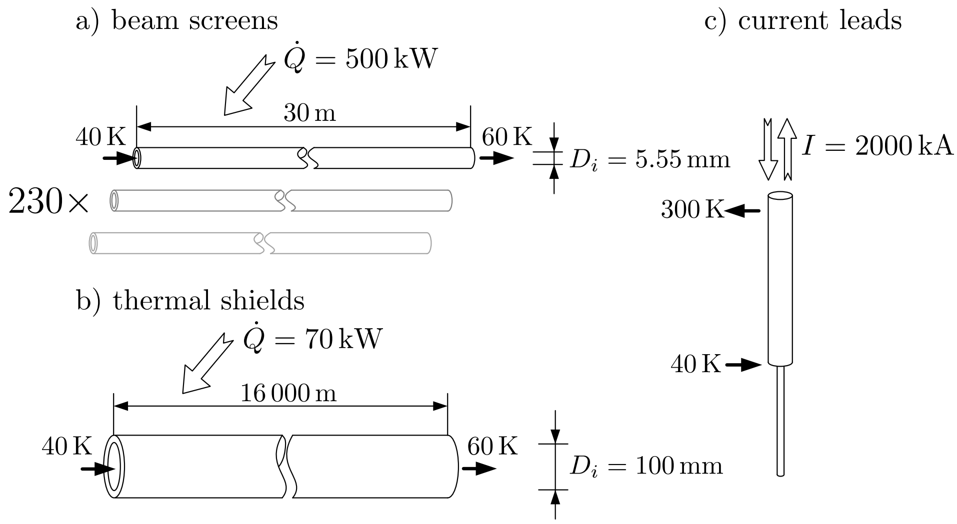

Figure 1. Heat loads per segment and geometrical constraints, twelve segments in total

3. Cycle Setup

3.1. Primary cycle

The division into a primary and secondary cycle is necessary because of two reasons. The first is the high pressure drop in the tubes of the beam screen. A single cycle configuration would have a very high outlet pressure of the compressor to overcome the pressure losses. On the other hand, there must not be neon in the beam screen to avoid induced radioactivity. Both cycles are connected via the load heat exchanger which absorbs the heat of the secondary cycle.

In a first step three different options for the primary cycle were examined. For the calculation of these cycles some additional parameters had to be assumed. During the project, these parameters will be adjusted according to the values that result from the examination of each component. The maximum pressure in the system was set to 40 bar. This pressure always occurs at the outlet of the brake compressor (2) (see Figure 2). The isothermal efficiency of the main compressor was set to 70 %, whereas the isentropic efficiencies of the brake compressor and the expansion turbines were set to 80 % and 85 %, respectively. To take into account losses between the expansion turbine and the brake compressor, a power recovery factor of 98 % was introduced. The outlet temperatures of the recoolers were set to 300 K. The number of transferunits (NTU) of the first and second main heat exchangers were set to 40 and 30, respectively. For the load heat exchanger, a temperature difference of 1 K was presumed, so that the inlet temperature is 39 K. The inlet temperature on the secondary side will be higher than 60 K, because of the energy input required for the circulation. For the first calculation a temperature of 62 K is assumed. The highest efficiency can be achieved if both heat capacity rates are the same. Therefore the temperature differences on both sides will be the same so that the outlet temperature of the primary cycle will be 61 K. The pressure drop in each heat exchanger was presumed to be 0.2 bar. The pressure ratio of the pre-cooling turbine in option C is a free parameter. It was chosen so that the input power of the main compressor becomes minimal.

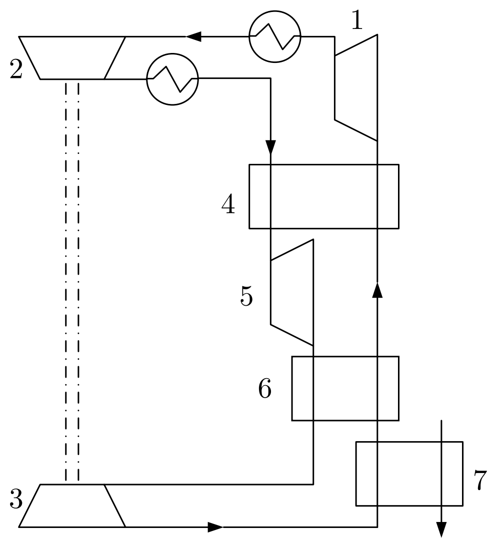

Figure 2. Proposed primary cycle (complete setup): (1) main compressor, (2) brake compressor, (3) expansion turbine, (4) primary main heat exchanger, (5) pre-cooling turbine, (6) secondary main heat exchanger, (7) load heat exchanger; option A: without (5) and (6), option B: without (5)

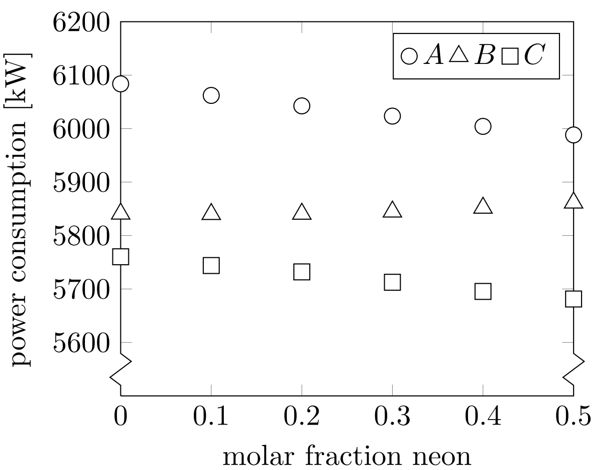

Figure 3. Cycle power requirements for the different design options of the primary cycle; A: without secondary main heat exchanger and without pre-cooling turbine (○), B: without pre-cooling turbine (△), C: complete (□)

The composition of the working fluid was changed in steps of 10 % from 0 % to 50 mol − % neon. The results of the calculation for a heat load of 500 kW are given in Figure 3. It can be derived that the addition of the second heat exchanger as well as of the precooling turbine improve the performance of the primary cycle. The results of this first calculation are only valid for an ideal configuration. Neon has a lower thermal conductivity, lower specific heat as well as a higher viscosity than helium, which results in heat exchangers that are larger, have a higher temperature difference and/or have a higher pressure drop. Since the maximum size of a heat exchanger is limited by the availability of vacuum furnaces, the cycles with a higher neon content will have a lower efficiency than given here. The cycle that is taken as a reference is option C with a neon content of 25 mol − %. This configuration has an input power of 5.7 MW for a refrigeration capacity of 500 kW, which equals a COP of 8.8 % or 57 % of the Carnot efficiency.

3.2. Secondary cycle

The secondary cycle distributes the cold along the long and narrow pipes of the beam screen. The resulting pressure losses and possible measures to reduce these are currently under investigation by a group at CERN [3]. For the given parameters and a maximum pressure in the secondary cycle, the pressure drop is 7.8 bar.

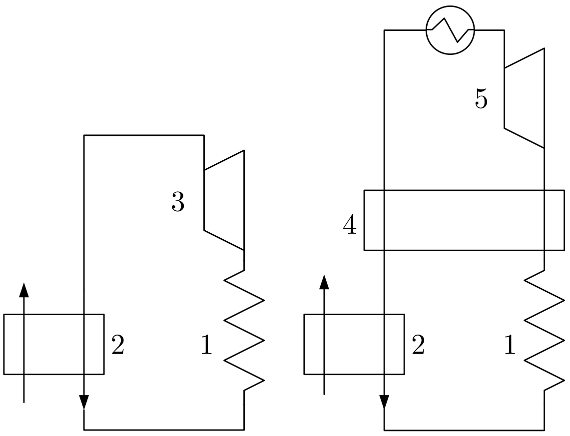

Figure 4. Proposed secondary cycles: (1) beam screen, (2) load heat exchanger, (3) cold circulation compressor, (4) warm circulation compressor, (5) inner heat exchanger

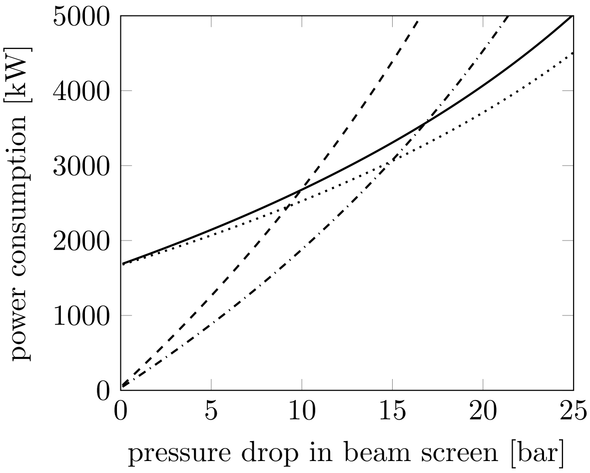

Figure 5. Total power for distribution:
—— warm compressor with $\eta_{is} = 83\%$,
······ warm compressor with $\eta_{is} = 100\%$,
- - - cold compressor with $\eta_{is} = 70\%$,
-·-·- cold compressor with $\eta_{is} = 100\%$

There are two options considered for the distribution of cooling power as depicted in Figure 4. The first is to use a cold compressor that increases the pressure of the secondary refrigerant after it leaves the beam screen. In this option, the power of the compressor has to be absorbed by the refrigerator. The other option is to use a counterflow heat exchanger in which the secondary refrigerant is warmed to ambient temperature where it is compressed in a warm compressor, recooled to ambient temperature and pre-cooled by the upstream. In this case, the power of the compressor is rejected by the after cooler but the cold end losses of the heat exchanger have to be absorbed at the low temperature. In both variants additional heat has to be absorbed by the load heat exchanger, which increases the refrigeration rate of the primary cycle. For the comparison of both options of the secondary cycle, the drive power of the circulation compressor as well as of the main cycle compressor have to be taken into account. For the warm circulation compressor, an isentropic efficiency of 83 % was assumed, whereas for the cold compressor 70 % is a realistic value. To show the theoretical limit of possible future improvements in the next years, also compressors with an isentropic efficiency of 100 % are included. The NTU of the heat exchanger of the system with a warm compressor is set to 40. The results are included in Figure 5. For the non-ideal case, the break even, where for a higher pressure drop the warm compressor is the better option, is at 10 bar. It would be at 15 bar for the ideal case. With the pressure drop of 7.8 bar a cold circulation compressor would be the better option with a power consumption of 2 MW in comparison with 2.4 MW for the warm compressor. For the total power requirement the values for refrigeration (Figure 3) have to be added.

3.3. Main Compressor Design

The primary cycle requires a pressure ratio of 6 over the main compressor if a side stream for the cooling of the current leads is included (not shown in Section 3.1). The state of the art compressor for such a task would be a screw compressor. The substitution of a such a compressor by a turbo compressor causes the latter to have a high number of stages to provide the same pressure ratio. The pressure head of a stage of a centrifugal compressor is determined by the pressure coefficient, the circumferential speed u_2 and the properties of the working fluid. These three parameters can be used to increase the pressure head per stage and therefore reduce the number of stages required for the compression. The pressure coefficient, which is defined as $\psi = 2\Delta h_{is}/{u_2}^2$ is set to 1. Current centrifugal compressors that might be suitable for the application are limited to a circumferential speed of $340\,\mathrm{m\,s^{-1}}$. Increasing this value, for example to $410\,\mathrm{m\,s^{-1}}$, would decrease the number of the stages required. The main approach in this project to reduce the number of stages is to change the properties of the working fluid. With the isentropic head given, the pressure ratio per stage for an isentropic process and the required number of stages can be calculated. The results of this calculation are given in Figure 6. A pure helium compressor with a circumferential speed of $340\,\mathrm{m\,s^{-1}}$ would require 20 stages. By increasing the speed to $410\,\mathrm{m\,s^{-1}}$, the number of stages can be lowered to 14. When using Nelium instead of pure helium, the number of stages can be brought down to 7 for a neon content of 25 %. The higher the neon content is, the lower is the effect of adding more neon. Since it has a negative impact on the heat exchange, it is favorable to limit the neon content. The exact content is to be determined as part of the optimization process, but a content of 25 % will be used for the preliminary design.

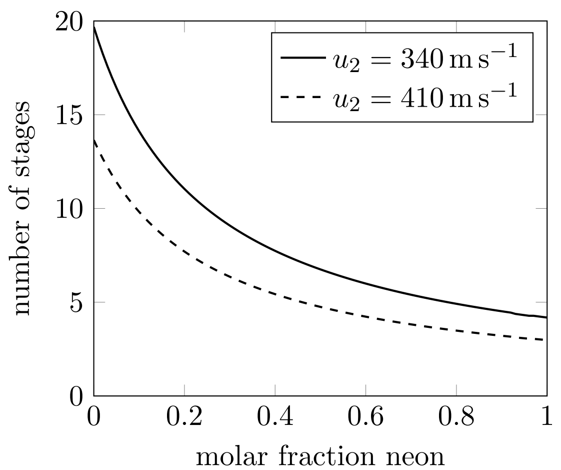

Figure 6. Required number of stages over neon content for an isentropic compression with re-cooling to 300 K after every stage and an overall pressure ratio of 6

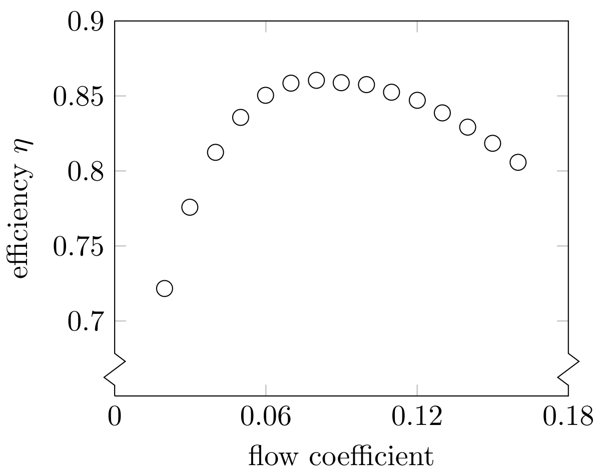

Figure 7. Achievable efficiency of one stage of a centrifugal compressor over flow coefficient, according to [4]

The achievable efficiency of a stage in a real compressor can be estimated depending on the flow coefficient ϕ which is defined as $\phi = \dot{V}_{in}/D_2^2 u_2$ [4]. It is advantageous to limit the flow coefficient between 0.06 and 0.12, so that the efficiency is higher than 0.85. In a single shaft compressor, the RPM of every stage is the same. The highest pressure ratios per stage can be achieved if the diameter is also constant. The flow coefficient therefore only depends on

the volume flow, which itself is a function of the outlet pressure and temperature of the stage before. The higher the pressure and the lower the temperature the lower will be the volume flow and therefore the flow coefficient. Therefore the value for the flow coefficient for the first stage of a group should be 0.12. It follows from this calculation that the achievable pressure ratio for a stage group is limited by the lower limit of the flow coefficient. For an ideal gas, the pressure ratio is the inverse of the volume ratio for a constant temperature. With the above, the achievable pressure ratio per stage group is $\Pi_g = (\phi_1/\phi_n)\Pi_n$ where ϕ_1 and ϕ_n are the flow coefficients of the first and last stage and Π_n is the pressure ratio of the last stage which has to be included because of the definition of the flow coefficient at the inlet of the stage. For the given values and re-cooling to 300 K after every stage, the achievable pressure ratio is 2.6. To fulfill the requirement of a pressure ratio of 6, at least two stage groups have to be used. For these, two options exist. The first is to use a multi shaft design where the second group is in a separate machine with RPM different from the first. This allows to choose optimal RPM but comes at the expense of having two drives, bearings etc. The other possibility is to reduce the outer diameter of the second group and mount it on the same shaft. This allows reduced investment on the drive side but requires a higher number of stages. The diameter is determined so that the flow coefficient of the first stage of the second group is again 0.12.
In a real compressor it is not viable to re-cool after every stage. A cooler between for example every second or third stage is sufficient to keep the temperature below 400 K and stay relatively close to the ideal case of isothermal compression. Cooling can also be used to influence the achievable pressure ratio per group. A commercially available chiller can cool the inlet stream to 279 K. A possible inlet temperature for the last stage is 350 K. With the equation for the pressure ratio per stage and the ideal gas law, a factor of 1.25 for the achievable pressure ratio can be of a stage group can be calculated. The exact layout of the compressor has to be determined iteratively. One possible configuration with a single shaft is given in Figure 8. The pressure increase from 31.9 bar to 40 bar is generated by the brake compressor.

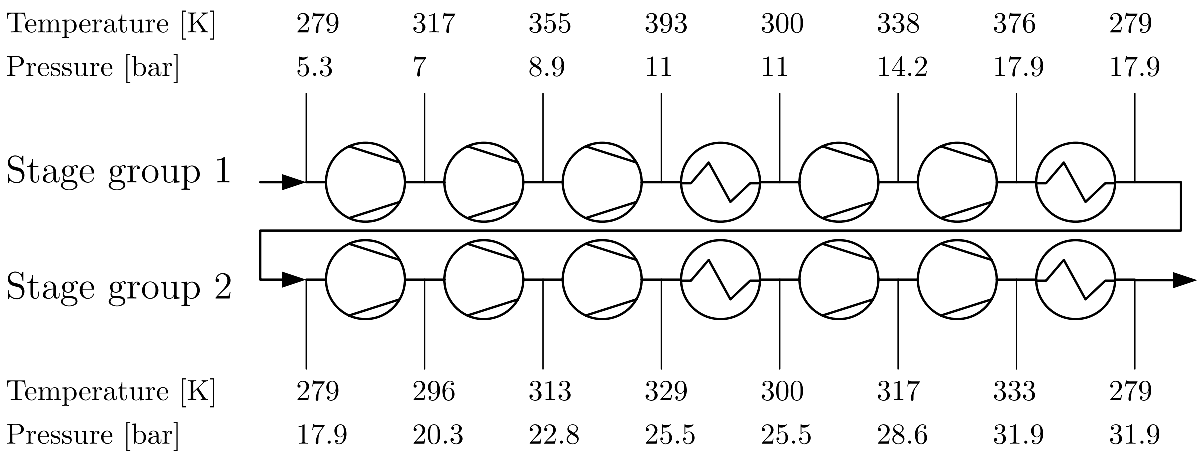

Figure 8. Compressor configuration in single shaft arrangement

4. Conclusions

For the primary cycle, the option that includes both pre-cooling turbine and second inner heat exchanger is most promising. The higher the neon content, the more feasible the pre-cooling turbine becomes. For the secondary cycle a cold compressor is slightly better than a warm compressor. The main compressor has to include at least two stage groups with either different speed or outer diameter.

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