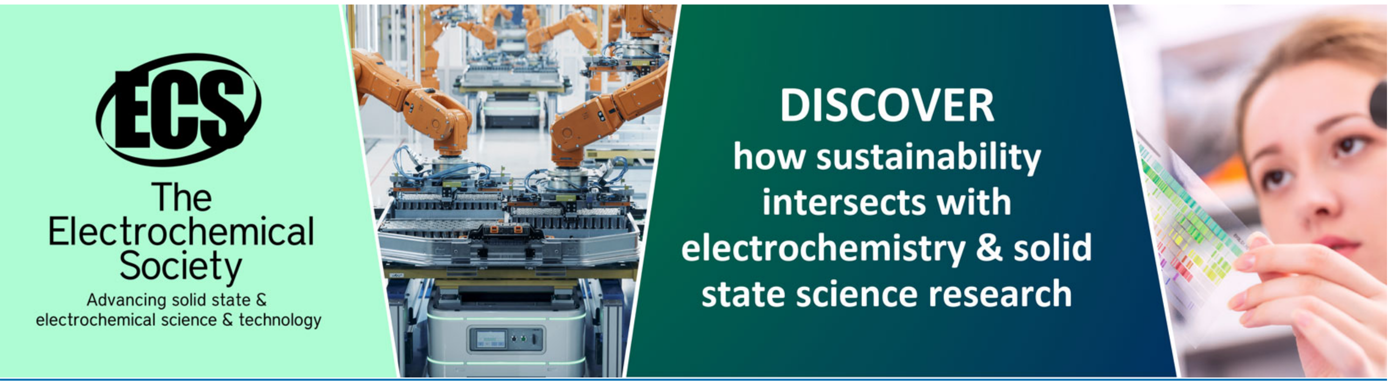
Thermodynamic and economic aspects of the Nelium Turbo-Brayton refrigerator development for the FCC-hh



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Thermodynamic and economic aspects of the Nelium Turbo-Brayton refrigerator development for the FCC-hh

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Abstract. Within the conceptual design study for the Future Circular Hadron Collider (FCC-hh) it was shown, that a large part of the total cryogenic heat load falls into the temperature range between 40 to 60 K. Thus, additional cryogenic refrigerators for this temperature level are specified for each of the 10 foreseen cryoplants. Such a cryogenic system was developed at the TU Dresden and is based on a Brayton cycle working with a neon-helium mixture as refrigerant and using multistage centrifugal compressors. The duty requirements comprise a 6.2 MW total heat load at 40 to 60 K for beam screens and shielding, additional 2.7 MW at 300 to 40 K for the pre-cooling of the helium cycle and a turndown ratio of up to 3.8. The optimisation of the referenced system was performed in order to obtain a high efficiency of the cryogenic cycle and tolerable costs for system components at the same time. An analysis of the mixture composition influence on the components and on the total gas mass was performed. Restrictions for industrially existing hardware were taken into consideration. Updated cycle parameters are subsequently described.

1. Introduction

Designing an efficient and reliable system is one of the main goals of the cryogenic system development. The Brayton cycle with a multi-stage turbo compressor for the beam screens and thermal shields cooling from 40 to 60 K has been previously designed [1] and was improved within this study. A mixture of neon and helium, so-called Nelium, is used as the refrigerant. The mixture composition is one of the main parameters of the cycle improvement. Thus, it influences the design of the cycle components, which determines the investment and operational costs of the cryogenic refrigerator. Moreover, the turbo compressor design imposes some limitations to the preferred helium content. To shift these limitations, the influence of the mixture composition on the cycle components was studied and the following cycle arrangement improvements were proposed in this paper.

2. Cycle design constrains

2.1. Turbo compressor design

The multi-stage turbo compressor is a key component of the cycle. The cost of one compressor unit is estimated to be up to 40 % of the total system investment costs. Only the usage of one-motor machine is justified in terms of process costs, otherwise the expenses for two separate compressors would be doubled [2]. Within the design of a multi-stage turbo compressor for

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light gases the limitations resulting in a restricted number of impellers and maximum achievable pressure ratio per machine are applied. From the cycle perspective, the required number of impellers can be estimated depending on the cycle parameters and a neon-helium mixture composition. Thus, from [3] it can be written:

$$y_p = \frac{1}{2}\psi_p U_2^2, \tag{1}$$

where y_p – polytropic head in kJ/kg, ψ_p – polytropic load coefficient, U_2 – tip speed in m/s. Therefore, the pressure ratio per stage Π_i can be calculated:

$$\Pi_i = \left(1 + \frac{k-1}{k\eta_p}\frac{\psi_p U_2^2}{2R_i T_1}\right)^{\frac{k\eta_p}{k-1}}, \tag{2}$$

where k – mean ratio of specific heats, η_p – polytropic efficiency, R_i – specific gas constant in J/kgK, T_1 – inlet temperature in K. According to [3], the optimal coefficient of polytropic head is in the range of 0.8 to 1.1 and can be assumed as 1.0.

Thus, the required number of compressor stages can be calculated for a specific cycle depending on the total required pressure ratio, mixture composition and tip speed (figure 1). The real number of turbo compressor impellers depends on the impeller geometry, number of inter- and after-coolers and other factors. However, the tendency to increase the number of required impellers for lighter gases remains. According to the compressor design at the University of Stuttgart in cooperation with MAN Energy Solutions based on the current manufacturing limitations, the application of one machine (with 9 impellers with the maximum diameter of 600 mm and the designed rotational speed of 9500 rpm) allows up to 40 vol.-% of helium [2].

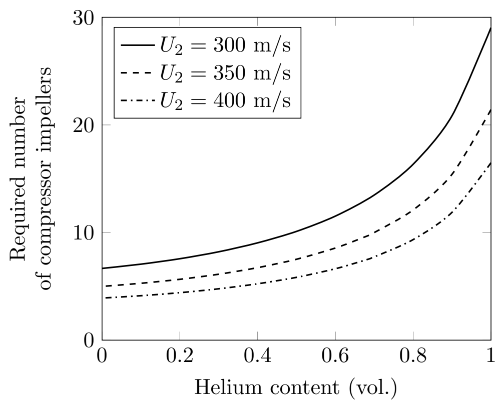

Figure 1. Influence of the mixture composition on the number of impellers.

2.2. Heat exchangers and cold box

The second important parameter influenced by the mixture composition is the size of the system. It is mainly determined by the size of the heat exchangers and subsequently of the cold box. Using the equation of the total heat exchanger volume and the Colburn factor from [4], the equation for the cross-sectional area from [5], and replacing all the parameters with functions of gas thermophysical properties, the cross-sectional area of the heat exchanger for the mixture over the one for the pure helium can be expressed:

$$\frac{A_{ci}}{A_{c0}} \simeq \frac{m_i}{m_0}\left[\left(\frac{C_{pi}}{C_{p0}}\right)^{\frac{1}{3}}\left(\frac{\mu_i}{\mu_0}\right)^{\frac{1}{3}}\left(\frac{\lambda_0}{\lambda_i}\right)^{\frac{1}{3}}\left(\frac{\rho_0}{\rho_i}\right)^{\frac{1}{2}}\right]\sqrt{\frac{NTU_i}{NTU_0}\frac{\Delta p_0}{\Delta p_i}} \tag{3}$$

The volume of the heat exchanger for the mixture over the one of the pure helium is as follows:

$$\frac{V_i}{V_0} \simeq \frac{NTU_i}{NTU_0}\left(\frac{A_{ci}}{A_{c0}}\right)^{0.52}\left[\left(\frac{Pr_i}{Pr_0}\right)^{\frac{2}{3}}\left(\frac{m_i}{m_0}\right)^{0.48}\left(\frac{\mu_0}{\mu_i}\right)^{0.48}\right] \quad (4)$$

Here V_i/V_0 – ratio of heat exchanger core volumes assumed to be equal to the ratio of total volumes, C_p – isobaric heat capacity, A_c – cross-sectional area, m – mass flow, NTU – number of transfer units calculated using equations for counterflow heat exchangers [6], Δp – heat exchanger total pressure drop, μ – viscosity, λ – thermal conductivity, ρ – density, Pr – Prandtl number. The indexes 0 and i are related to pure helium and neon-helium mixture respectively. The NTU values and pressure drops are assumed to be constant within the analysis. The thermophysical properties are calculated as mean integral values depending on flow parameters and the mixture composition. Therefore, the ratios of heat exchanger volumes, cross-sectional areas and following cold box diameters can be found as functions of the mixture composition (figure 2). Moreover, the ratio of the cycle gas mass to the pure helium cycle can be calculated in respect to the system volume and mean gas density (figure 3).

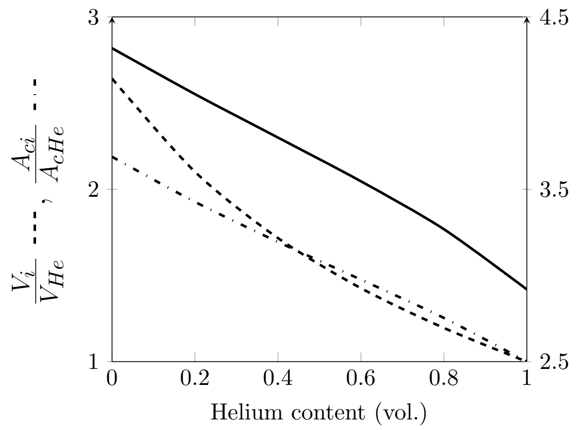

Figure 2. Influence of the mixture composition on the specific exchanger volume $\left(\frac{V_i}{V_{He}}\right)$, cross-sectional area $\left(\frac{A_{ci}}{A_{cHe}}\right)$ and cold box diameter (D_{CB}).

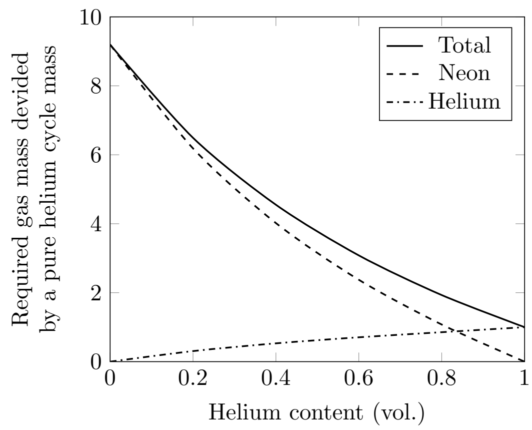

Figure 3. Influence of the mixture composition on the required total gas mass.

The increase of the system size as well as of the required gas mass, especially neon, results in an increase of the process cost, so a higher helium content would be preferable for the system design. Thus, the target of the cycle development is to achieve high process performance using one compressor and a gas mixture with a maximum helium content.

3. Cryogenic cycle development

The flow diagram of the baseline Turbo-Brayton Cycle is shown in figure 4 a. It includes the main turbo compressor (MC) with one motor (M), two casings (C1, C2) and after-coolers, inner and load heat exchangers (IHX, LHX), pre-cooling (PCT) and main turbines (MT). The input power of the main turbine is approximately 750 kW, which is partially recovered in the booster compressor (BC). This system has been previously optimised and a turbo compressor for 40 vol.-% of helium has been industrially developed. To shift the turbo compressor design limitations, a new cycle arrangement has been proposed (figure 4 b). In the baseline cycle the main and the pre-cooling turbine were connected in series. The improved arrangement with a

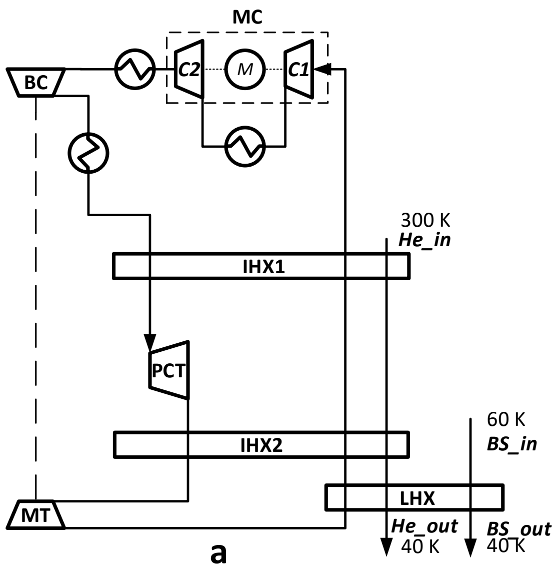

Figure 4. Flow diagrams of the baseline (a) and improved (b) cycle.

parallel pre-cooling turbine with medium outlet pressure and redirection of the flow to the inlet of the second compressor casing is proposed. Both cycles were calculated using the Refprop 10.0 library, and the advantages of the new cycle had been shown.

3.1. *Advantages of the proposed cycle*

The first advantage of the improved cycle (b) is the reduced required compressor pressure ratio. Assuming $NTU_{IHX1b} = NTU_{IHX2b} = \frac{1}{2}NTU_{IHX1a}$ for cycle (b), the system parameters were calculated depending on the helium content. According to the calculations, the required pressure ratio can be reduced by at least 7 % (figure 5).

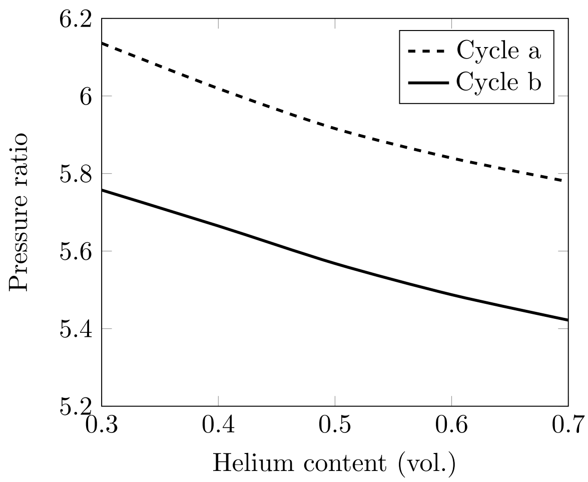

Figure 5. Required compressor pressure ratio of the baseline (a) and improved cycle arrangement (b) depending on the mixture composition.

Moreover, it can be further reduced by varying the pre-cooling turbine flow parameters and the NTU values of IHX1 and IHX2. A dedicated optimisation was performed for a 40 vol.-% helium content and the optimal required pressure ratio of 5.11 was achieved.
The second advantage is the increase of the volumetric flow at the inlet of the second compressor casing. In a multi-stage turbo compressor the volumetric flow usually reduces progressively from stage to stage. This results in a reduction of the flow coefficient φ [3]:

$$\varphi = \frac{4Q_{in}}{D^3\pi^2 n}, \tag{5}$$

where D – impeller diameter in m, n – rotational speed in rps, Q_{in} – inlet volumetric flow in m^3/s. The flow coefficient influences the stage efficiency and has an optimum at around 0.06 [3]. Thus, its reduction below the optimum needs to be compensated by a reduction of the impeller diameter. This subsequently leads to a reduction of the pressure ratio and is especially significant for the last stages. Therefore, the increase of the mass flow at the inlet of the second compressor casing should bring down this effect and improve the compressor performance.

3.2. Upper heat exchanger design

The new cycle arrangement includes an additional heat exchanger IHX2 (figure 4 b). However, to keep a cross-sectional area of the heat exchanger and a coldbox diameter unchanged, a joint design of the upper heat exchanger (IHX1+IHX2) is considered (figure 6). At the same time, the total NTU value and the pressure drop of the joint upper heat exchanger is equal to the baseline cycle upper heat exchanger. The entire high pressure flow is extracted from the heat exchanger and a part of it is expanded in the pre-cooling turbine, while the rest is returned to the heat exchanger. This increases the total length by the size of the middle distributor, which is relatively small compared to the total cold box length. The position of the flow extraction depends on the optimal pre-cooling turbine flow and pressure with respect to the energy balance.

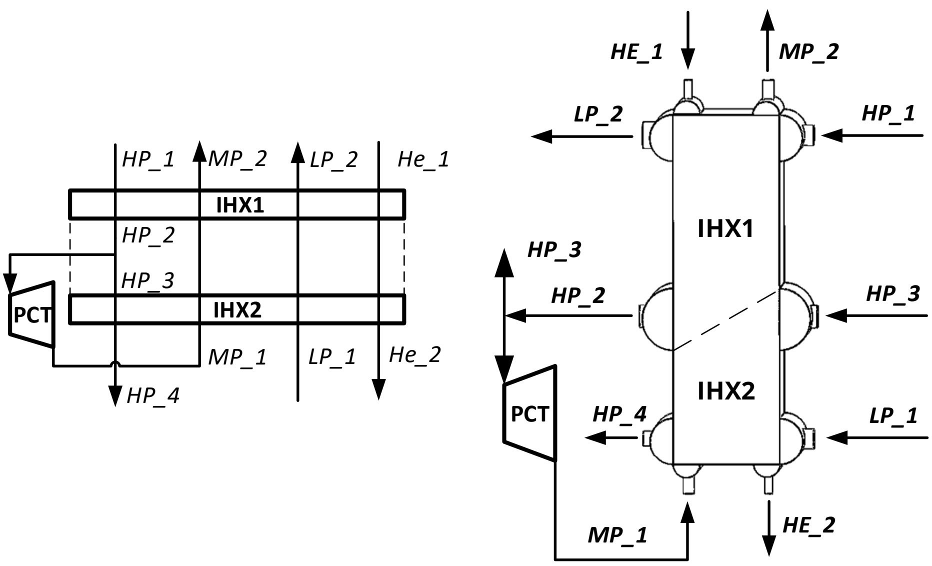

Figure 6. Upper heat exchanger design.
LP, MP, HP – low, medium and high pressure streams; PCT – pre-cooling turbine; IHX – inner heat exchanger.

3.3. Improved cycle performance
The cycle was optimized for 40 vol.-% of helium. The total power consumption depends on the pre-cooling turbine outlet pressure and mass flow. This mass flow $\dot{m}_{PCT}$ increases linearly to the middle pressure (figure 7). At the same time, the required pressure ratio reduces, but the power of the second compressor casing increases. Thus, the optimum for the minimum cycle power can be found (figure 8). However, this optimum needs to be matched with the preferred design middle pressure of the turbo compressor. The power estimations (figure 8) were based on the calculation of a typical stage performance depending on the flow coefficient [7]. The actual compressor power will depend on real stage losses. For comparison, the power consumptions of the baseline cycle designed for the load of 680 kW at 60 to 40 K for the beam screen cooling including the secondary cycle losses recovery and 270 kW at 300 to 40 K for the helium cycle pre-cooling were estimated to be approximately 10.5 MW.

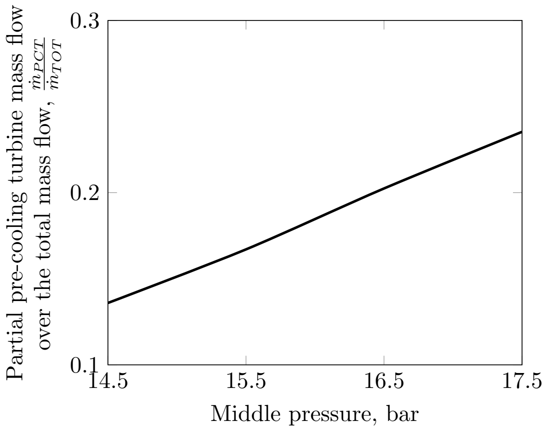

Figure 7. Pre-cooling turbine mass flow related to the total mass flow as a function of the 2^{nd} casing inlet pressure for 40 vol.-% helium.

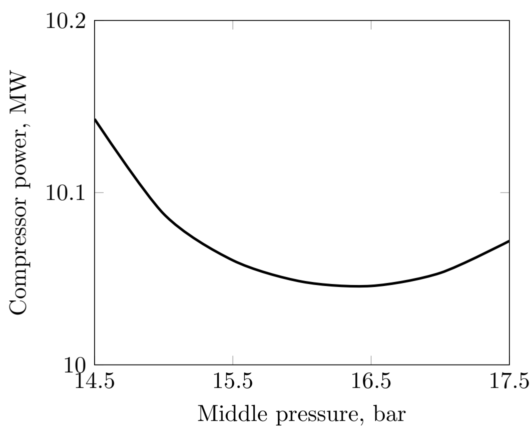

Figure 8. Estimated compressor power as a function of the 2^{nd} casing inlet pressure for 40 vol.-% helium.

4. Conclusion
This study suggests improvements for a Turbo-Brayton cycle arrangement as input value for the turbo compressor design. In contrast to the conventional cryogenic cycles with screw compressors, the cycles using turbo compressors are more sensitive in terms of changes in cycle parameters. Thus, the new cycle arrangement allows two strategies of the system improvement. The first strategy corresponds to the reduction of the system cost by increasing the helium content compressible in the one compressor. In contrast, the second strategy suggests increasing the compressor efficiency by reducing the number of required impellers for the current composition (40 vol.-% of helium). The final strategy needs to be chosen according to the most advantageous industrial turbo compressor design for the improved cycle. For the mixture containing 40 vol.-% of helium the power savings could be up to 0.5 MW per cryogenic plant with respect to the baseline. Therefore, assuming energy costs of 60 CHF/MWh, 1800 days of the collider operation within 10 years and a turndown ratio of 3.6 within 50 % of the operation time, a cost reduction of up to 8 Mio. EUR within ten years could be estimated for the ten cryoplants.

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References

[1] Kloeppel S, Quack H, Haberstroh C and Holdener F 2015 Neon helium mixtures as a refrigerant for the FCC beam screen cooling: comparison of cycle design options *IOP Conf. Ser.: Mater. Sci. Eng.* **101** 012042
[2] Podeur M and Vogt D M 2019 Design of a multi-stage turbocompressor for the FCC pre-cooling cycle *Poster presented at FCC Week in Brussels*
[3] Luedtke K H 2004 *Process Centrifugal Compressors. Basics, Function, Operation, Design, Application* (Berlin, Heidelberg: Springer) p 37
[4] Guo K, Zhang N and Smith R 2015 Optimisation of fin selection and thermal design of counter-current plate-fin heat exchangers *Appl. Therm. Eng.* **78** pp 491-499
[5] Quack H, Haberstroh C, Seemann I and Klaus M 2015 Nelium, a Refrigerant with High Potential for the Temperature Range between 27 and 70 K *Phys. Procedia* **67** pp 176-182
[6] Shah R K and Sekulić D P 2003 *Fundamentals of heat exchanger design* (Hoboken, New Jersey: John Wiley & Sons, Inc.) p 128
[7] Aungier R H 2000 *Centrifugal compressors. A strategy for aerodynamic design and analysis* (New York: Asme Press) p 8