TECHNISCHE UNIVERSITÄT DRESDEN

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

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Motivation

6.2 MW of cooling power at 40 to 60 K for the beam screens and thermal shields cooling and 2.7 MW at 300 to 40 K for the Helium-cycle pre-cooling need to be provided by 10 cryoplants for the *Future Circular Collider* operation. For this purpose a reverse Brayton cycle using a multi-stage turbo-compressor with magnetic bearings with a neon-helium mixture has been developed. The critical aspect of the cycle optimisation is an industrially realizable turbo-compressor design with affordable system cost and size. To fulfil manufacturing limitations and to increase process efficiency, the cycle arrangement has been improved with respect to the turbo-compressor design.

Design constraints

A single turbo-compressor presents up to 40 % of the investment costs. Therefore, only one machine is acceptable. The required number of compressor stages increases with increasing helium content (Fig. 1). Manufacturing limitations, such as maximum tip speed, number of impellers per casing and their diameters, rotor aerodynamics etc., restrict the application of one compressor to mixtures with high neon content. Presently, the compressor design for up to 40 vol. % of helium has been industrially confirmed. On the other hand, a higher helium content would be preferred, as it would allow reduced sizes of heat exchangers and the cold box (Fig.1). Thus, the target of the cycle development is to achieve high process performance using one-tandem compressor and a gas mixture with a maximum helium content.

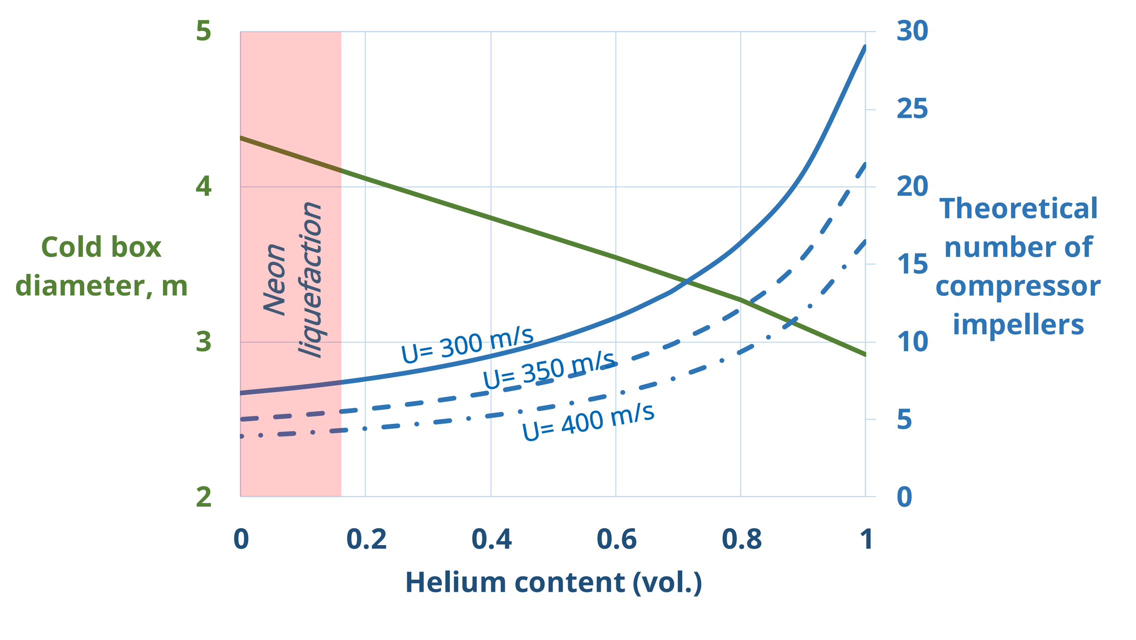

Fig. 1. Influence of the mixture composition on the components

Cycle development

The cycle includes the main and pre-cooling expansion turbines, approximately of 750 and 300 kW power respectively. The integration of a pre-cooling turbine increases the exergetic efficiency of the cycle. In the former cycle design the turbines were in series (Fig. 2 a). In contrast, the new cycle arrangement with a parallel pre-cooling turbine and the medium pressure flow redirected to the inlet of the second compressor casing is proposed (Fig.2. b). The cycles comparison showed that the new arrangement is more advantageous for the turbo-compressor design and the expected cycle efficiency.

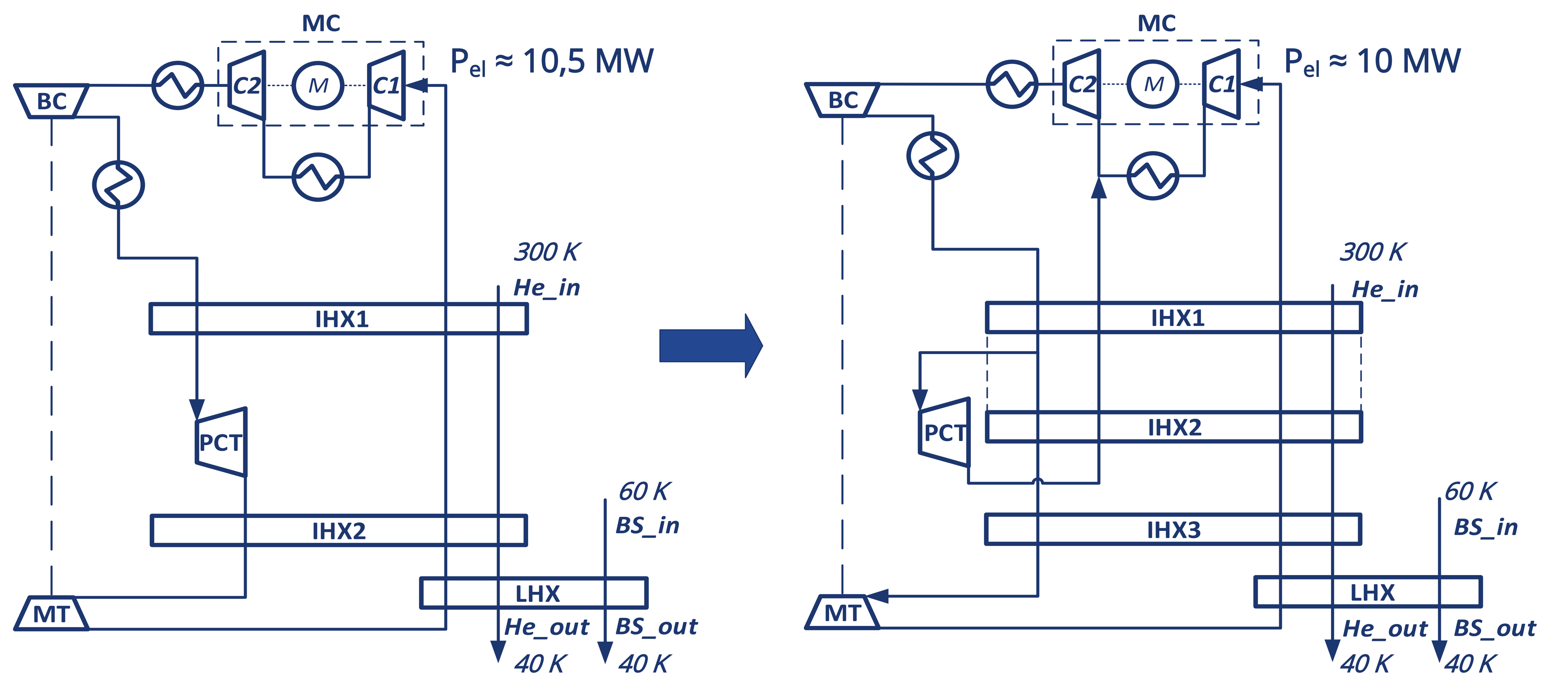

Fig. 2. Flow diagrams of the baseline (a) and improved (b) Turbo-Brayton cycle

The improved cycle arrangement ***b*** has the following advantages over the baseline ***a***:

— increased volumetric flow V_{in2} on the inlet of the second compressor casing (Fig. 3);

— reduced (by at least 7 %) cycle pressure ratio (Fig. 4).

The latter results from the higher inlet and subsequently outlet pressure of the main turbine at the fixed cycle high pressure to provide the same required cooling power. On the other hand, the higher volumetric flow on the inlet of the second compressor casing helps to increase the flow coefficient φ or diameters of the casing impellers. This improves the polytropic efficiency or the achievable pressure ratio of the compressor. Thus, two strategies are available for the process improvement: efficiency increase at 40 vol. % of helium or reduction of the cold box size and cost by the helium content increase.

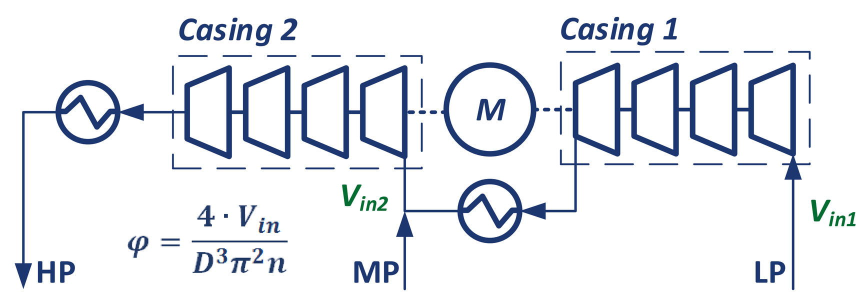

Fig. 3. Compressor casings inlet flows

(LP, MP, HP—low, medium and high pressure; V—volumetric flow)

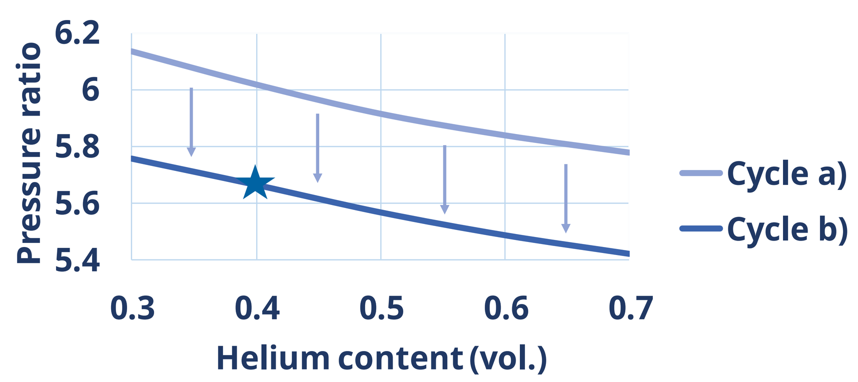

Fig. 4. Required compressor pressure ratio

Performance evaluation

As the cycle ***b*** provides significant advantages for the compressor design, the compressor power can be reduced from 10,5 MW to approximately 10 MW for the mixture with 40 vol. % of helium. The cycle ***b*** can be optimised by varying the pre-cooling turbine mass flow and outlet pressure. On the other hand, the middle pressure between the casings is given by the optimal compressor design. Thus, the best cycle design needs to be adjusted together with compressor manufacturers.

The analysis of exergy losses distribution for the designed cycle shows relatively high exergy efficiency of the cold box and cycle with currently developed compressor (68 % and 48 % respectively, Fig. 5). Thus, the compressor performance improvement due to the new cycle arrangement will lead to reduction of main compressor and followed after-coolers exergy losses and increase in the cycle exergy efficiency by 4 %.

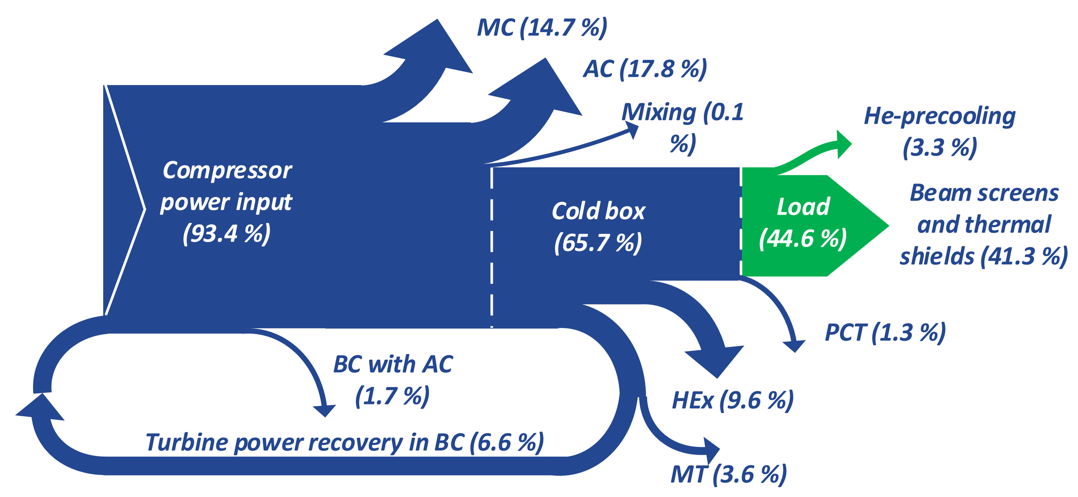

Fig. 5. Exergy balance of the designed system

MC—main compressor (C1 and C2— casings); BC—booster compressor; AC—after-coolers; HX—heat exchangers (IHX—inner; LHX—load); MT and PCT—main and pre-cooling turbines

Conclusion

In contrast to traditional systems with screw compressors, systems with turbo-compressors are much more sensitive to changes in cycle parameters. Thus, introduced cycle design improvements are beneficial for the main turbo-compressor design and allow to achieve higher system performance. The expected power savings compared to the baseline are estimated to be up to 500 kW per cryoplant which would result in 18 Mio. EUR savings for 10 plants within 10 years of the operation.

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