

Investigation and performance assessment of hydraulic schemes for the beam screen cooling for the Future Circular Collider of hadron beams

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Abstract. The international study at CERN of a possible future circular collider (FCC) considers an option for a very high energy hadron-hadron collider located in a quasi-circular underground tunnel of about 100 km of length. The technical segmentation of the collider foresees continuously cooled sections of up to 10.4 km; throughout the entire section length, more than 600 kW of heat mainly generated by the beam synchrotron radiation must be removed from the beam screen circuits at a mean temperature of 50 K. The cryogenic system has to be designed to extract the heat load dependably with a high-efficiency refrigeration process. Reliable and efficient cooling of the FCC beam screen in all possible operational modes requires a solid basic design as well as well-matched components in the final arrangement. After illustrating the decision making process leading to the selection of an elementary hydraulic scheme, this paper presents preliminary conceptual designs of the FCC beam screen cooling system and compares the different schemes regarding the technical advantages and disadvantages with respect to the exergetic efficiency.

1. Introduction

The Future Circular Collider (FCC) under conceptual study at CERN will have a circumference of about 100 km. The high energy proton beams will reach energies up to 50 TeV per beam, emit synchrotron radiation and generate resistive heating due to an image current producing a heat load of 31.4 W/m per beam. The direct impact of this heat load on the superconducting magnets operating at low temperature (1.9 K - 4.5 K) is technically and economically unacceptable. Consequently, the heat load must be intercepted by a beam screen which will be actively cooled at a higher temperature and will protect the magnets. Despite the thermal insulation, heat inleaks from the ambience to the thermal shield of about 9 W/m are expected, which are supposed to be absorbed in series by the beam screen cooling (BSC) system as well. The specific heat load to be extracted in the beam bending sections (arcs) will reach up to 71.8 W/m in nominal operation.

The actual segmentation of the FCC cryogenic system foresees ten sectors, which have to be cooled continuously and the supplying cryoplant located at one end. The cryogen (helium at a mean temperature of 50 K) partly has to be circulated over more than 20 km. Especially the pressure losses influence the feasibility and the efficiency of a cryogenic cooling system of this size.



2. Beam Screen Design

The beam screen (see Figure 1) is a metallic tube which contains the proton beam bent and focussed by the surrounding magnets. It is designed to fulfil several requirements for conservation of a stable and persistent beam, suitable for physical experiments. The main constraint is the scarce available space due to the restricted geometry of the magnets inner bore. Among other conditions the temperature of the core part of the beam screen (beam tube) must be kept between 40 K and 60 K to preserve the vacuum [1]. For this purpose cooling channels are integrated to extract the heat load within the available temperature range.

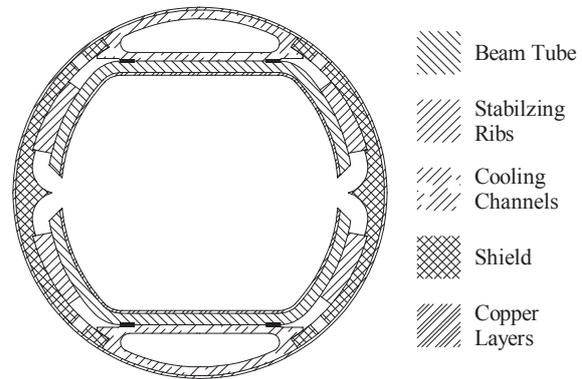


Figure 1. FCC beam screen design

3. Beam Screen Cooling Cycle

The actual baseline for the FCC cryogenic distribution system foresees ten sectors of two different designs; in eight of these ten sectors, the cryogen has to cool 8.4 km (≈ 550 magnets) of beam bending section, absorbing the heat generated by the synchrotron radiation and the image current. Depending on the final position of the cryoplants, the helium has to be led over an additional distance of 2 km of straight section, where no synchrotron radiation is emitted. Due to the longer distance the cryogen has to be delivered in the latter kind of sector, it was taken into consideration for the investigations presented in this paper (see Figure 2).

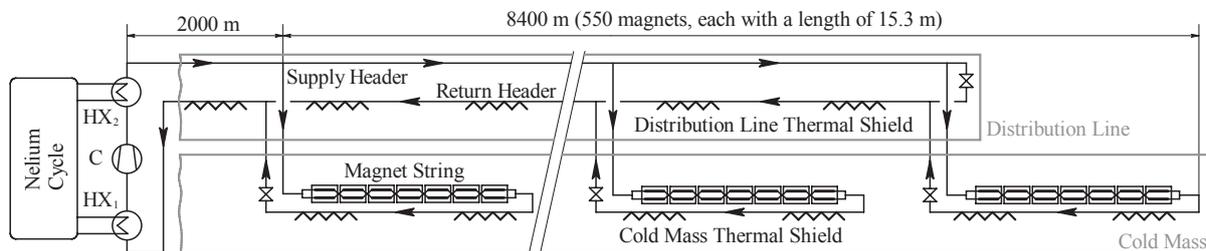


Figure 2. Schematic refrigeration design of a long FCC sector

The heat absorbed in the BSC cycle will be transferred to a refrigeration cycle using a helium-neon mixture (Neelium) as cryogen, which is under development by Dresden University of Technology [2].

4. Hydraulic Scheme

During the investigation for an efficient and dependable hydraulic scheme for the BSC cycle several technical possibilities have been assessed.

- **Control valves:** The use of control valves increases the investment costs and the controlling effort and could cause additional downtime due to component failure. The advantages of control valves though (e.g. individual control of single magnet strings and efficient cooling in different operational modes) outweigh the disadvantages. Therefore it was decided to use them, but to reduce the necessary number.
- **Flow direction in the return header:** The same flow direction of the fluids in supply and return header (parallel flow scheme) has advantages, when no control valves are used.

In the parallel flow scheme the pressure gradient in the headers are (qualitatively) the same leading to lower overall pressure losses, smaller pressure differences in supply and return header and therefore smaller overall mass flows. With the installation of valves the helium mass flow is not dictated by the pressure progresses in the headers (and the cooling channels) and can be minimized. It was to decided to install valves due to the simple and safe controlling of the refrigeration cycle, hence a flow scheme with different flow directions (counter flow scheme) in the headers was preferred to decrease the necessary piping.

- **Basic refrigeration cycle arrangement:** The arrangement of the basic components of the refrigeration cycle was chosen to be *heat sink (HX₁) - circulator (C) - heat sink (HX₂) - heat source (MS) - valve (V)* ('cold' circulator (Figure 3 left)). This setting minimizes the necessary mass flow as well as the pressure drop in the magnet strings and stabilizes the circulator inlet temperature.
- **Supply pressure:** The supply pressure was chosen to be as high as reasonable (50 bar) to reduce the pressure losses and the pressure ratio of the circulator.
- **Thermal Shielding:** The thermal shielding of the cold mass and the distribution line has to be installed, regardless of the emission of synchrotron radiation, hence over the entire sector length of 10.4 km. The reduction of the pressure drop in the last parallelly cooled magnet string is crucial to achieve reasonable exergetic efficiencies, which is accomplished by minimizing the temperature of the helium at the last magnet string inlet. For this reason the supply header will be led thermally insulated in the distribution line, whereas all the thermal shielding will be performed by the refrigerant after cooling the beam screen in the magnet strings. Therefore additional piping dedicated to the thermal shielding of the magnets vacuum jacket must be installed. These flows enter the return header in the distribution line after passing the control valves located at the end of the thermal shielding pipes. The return header will perform the thermal shielding of the distribution line itself.

5. Warm Circulator

With increasing pressure losses, additional circulation power has to be extracted at low temperature calling for a high exergetic effort. As an alternative option, the use of a circulator working at ambient temperature was considered ('warm' circulator (Figure 3 right)). The major part of the circulator power is extracted in a heat exchanger working at ambient temperature (HX_a); the terminal temperature difference (TTD) in the necessary internal heat exchanger (HX_i) though causes an additional amount of heat to extract at cryogenic temperature level.

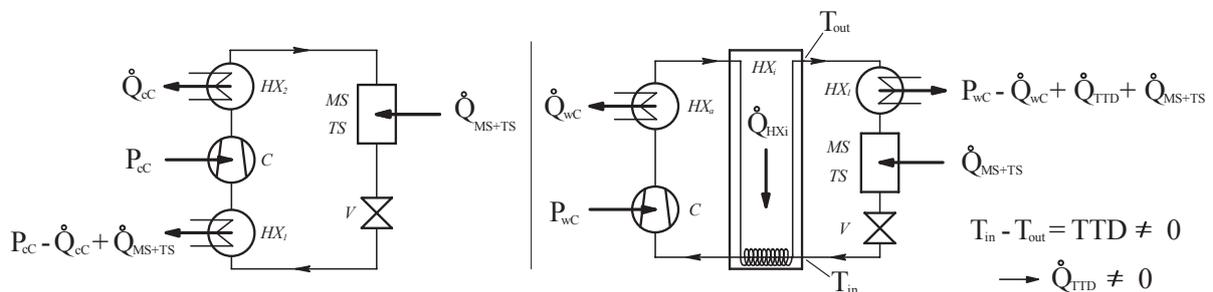


Figure 3. left: Cold circulator scheme | right: Warm circulator scheme

If high pressure drops are generated, the circulation power of a cold circulator (P_{cc}) and its extraction at cryogenic temperature level exceeds the circulation power at ambient temperature (P_{wc}) and the heat to extract at cryogenic temperature due to the terminal temperature difference of the internal heat exchanger (\dot{Q}_{TTD}) put together. Additional (operating) costs

Table 1. Additional simulation input data

Input Value	Unit	Cold Circulator	Warm Circulator
Ambient temperature	K	300	300
Ambient pressure	bar	1.013	1.013
Circulator outlet pressure	bar	50	50
Circulator inlet temperature	K	40	$f(NTU_{HXi})$
Circulator isentropic efficiency [2]	-	0.7	0.83
Heat exchanger 2 outlet temperature	K	40	40
Internal heat exchanger NTU [2]	-	-	40
External heat exchanger outlet temperature	K	-	300
Valves maximal inlet temperature	K	57	57
Cooling channels hydraulic diameter	mm	5.61	5.61
Cooling channels cross section area	mm ²	53.8	53.8
Cooling channels absolute roughness	μm	1.5	1.5
Headers absolute roughness	μm	15	15
Thermal shielding pipe diameter	mm	50	50
Thermal shielding pipe absolute roughness	μm	15	15
Heat load on cold mass thermal shield	W/m	5	5
Heat load on distribution line thermal shield	W/m	4	4
Heat load on beam screen	W/m	62.8	62.8

of the warm circulator cycle (e.g. the circulation of the water in the ambient temperature heat exchanger) have not been taken into account.

6. Numerical Simulation

The temperature and pressure progresses in the headers and the beam screen cooling channels have been obtained by a numerical simulation. The helium properties were taken from the library 'HePak' (Cryodata Inc.). Presumed values and numbers and boundary conditions which have not been mentioned in the previous text are listed in Table 1.

The pressure losses in the heat exchangers were neglected. The Darcy friction factor to calculate the pressure drop in the tube flows was obtained with the explicit formula of Swamee-Jain [3]. To be able to control the mass flow in the magnet strings, the valves were assumed to generate 20% of the pressure loss of the adjacent magnet string, but at least 1 bar [4].

7. Exergetic Analysis

The exergy \dot{E} transported by the cryogen can be calculated with the enthalpy h and the entropy s of the cryogen at actual state and at ambient condition state (subscript a) as per

$$\dot{E} = \dot{m} [h - h_a - T_a (s - s_a)] \quad (1)$$

For each component of the BSC system an exergy balance can be calculated. The exergy contained in the incoming flow plus the exergy effort \dot{X} is equal to the exergy contained in the outgoing flow plus the exergetic benefit \dot{A} plus the exergetic loss \dot{B} .

$$\dot{E}_{in} + \dot{X} = \dot{E}_{out} + \dot{A} + \dot{B} \quad (2)$$

In steady-state operation the exergy contained in the helium at each point of the cycle does not vary with time - the total exergy content is constant. Therefore the sum of all exergetic efforts put up to run a refrigeration cycle working in steady-state, exit either as exergetic benefit or as exergetic loss.

$$\dot{X}_{tot} = \dot{A}_{tot} + \dot{B}_{tot} \quad (3)$$

The exergetic efficiency ζ is a good indicator to validate the costs of a thermodynamic process. It is defined as the ratio of the exergetic benefit and the exergetic effort.

$$\zeta = \frac{\dot{A}_{tot}}{\dot{X}_{tot}} = \frac{\dot{X}_{tot} - \dot{B}_{tot}}{\dot{X}_{tot}} = 1 - \frac{\dot{B}_{tot}}{\dot{X}_{tot}} \quad (4)$$

7.1. Exergetic Efforts

The exergetic efforts are the electrical power of the circulator (P_C) and the exergy gained during cool down in the heat exchangers.

$$\dot{X}_C = P_C = \dot{m} \frac{(h_{out} - h_{in})_{s=const.}}{\eta_s} \quad (5)$$

The exergetic effort to extract the heat in the heat exchangers connected with the Helium cycle was obtained by calculating the difference of the exergy contained in the entering and exiting flows. The sum of the circulator power and the exergetic gains in the heat exchangers connected to the Helium cycle yields the total exergetic effort.

$$\dot{X}_{tot} = \dot{X}_{HX1} + \dot{X}_{HX2} + P_C \quad (6)$$

7.2. Exergetic Benefits

Although the major part of the heat load due to the synchrotron radiation \dot{Q}_{SR} is extracted at the (higher) temperature level of the beam screen shield, the highest temperature of the beam tube was chosen to define the exergetic benefit. All efforts will be undertaken to keep the beam tube temperature below 60 K. As the task would have failed if the temperature of one point of the beam tube exceeded the maximal allowed temperature, this temperature (T_{BT}) is the technical basis for the calculation of the exergetic benefit of the BSC (subscript BS). Additionally the resistive heat due to the image current \dot{Q}_{IC} has to be extracted.

$$\dot{A}_{BS} = (\dot{Q}_{SR} + \dot{Q}_{IC}) \left(\frac{T_a}{T_{BT}} - 1 \right) \quad (7)$$

This argumentation also applies for the exergetic benefit of the thermal shielding (subscript TS). Although all the used exergy is transferred to ambient temperature level (and therefore becomes anergy by classic definition), the technical purpose only can be achieved by this transformation, hence the exergy difference in the headers due to the heat load is counted as benefit.

$$\dot{A}_{TS} = \dot{Q}_{TS} \left(\frac{T_a}{T_{TS}} - 1 \right) \quad (8)$$

7.3. Exergetic Losses

To improve the performance of the hydraulic scheme it is necessary to relate the correct amount of exergetic loss to each causing effect:

- (i) temperature difference necessary for heat transfer (subscript HT)

- (ii) pressure drop due to friction in the pipes (subscript Δp)
- (iii) isenthalpic expansion in the control valves
- (iv) non-isentropic compression in the circulator
- (v) mixing of helium in different states in the return header
- (vi) heat transfer losses in the internal heat exchanger (only for warm circulator)
- (vii) heat extraction in the heat exchanger at ambient temperature (only for warm circulator)

The exergy differences of the entering and the exiting mass flows listed from (iii) to (vii) are entirely losses. The exergy losses in heated pipes have to be matched to the heat transfer (i) and the pressure loss (ii). The heat transfer losses arise from the difference in exergy of a heat flux at two temperature levels: The lower temperature of the refrigerant and the higher temperature of the cooled object. In case of the thermal shielding the higher temperature corresponds to the ambient temperature and the heat transfer losses become zero.

$$\dot{B}_{HT} = \dot{Q} \cdot T_a \left(\frac{1}{T_{low}} - \frac{1}{T_{high}} \right) \quad (9)$$

Subtracting the exergetic benefit and exergetic loss due to heat transfer from the total exergy difference yields the pressure drop losses.

$$\dot{B}_{\Delta p} = \Delta \dot{E} - \dot{A} - \dot{B}_{HT} \quad (10)$$

The heat exchanger at ambient temperature is not a necessary component of the warm circulation cycle consuming exergy, hence its benefit deserves a closer look. The temperature of the extracted heat in the water-cooled heat exchanger is above the ambient temperature level. Therefore the heat output is an exergy decrease by definition.

$$\dot{B}_{HXa} = \int_{HXa} \left(1 - \frac{T_a}{T_{HXa}} \right) d\dot{Q}_{wC} \approx \left(1 - \frac{T_a}{T_{HXa}} \right) \dot{Q}_{wC} \quad (11)$$

Although the heat extraction above ambient temperature is an exergy loss, the necessary effort to extract the same amount of heat at cryogenic temperature level is much higher.

$$\Delta \dot{X}_{HX1} = \int_{HX1} \left(\frac{T_a}{T_{HX1}} - 1 \right) d\dot{Q}_{wC} \approx \left(\frac{T_a}{T_{HX1}} - 1 \right) \dot{Q}_{wC} \quad (12)$$

Whereas the exergetic benefit stays the same, the exergetic effort increases by the additional effort to extract the heat at low temperature level. The exergy losses increase by the same amount minus the loss in the heat exchanger at ambient temperature level. Due to the temperature levels of the two heat exchangers, the exergetic efficiency decreases, if the major part of the circulation power is not extracted at ambient temperature level.

8. Results

The pressure loss generated in the BSC cycle is the decisive influence quantity for the necessary power consumption. As the sector dimensions, the beam screen design and the heat loads to extract are determined, the main parameters to reduce the pressure drop are header diameter and the number of magnets in series per magnet string. The diagram in Figure 4 shows the progress of the pressure drop and the exergetic efficiency of the BSC cycle vs. the diameter of the headers for three different lengths of magnet strings (1, 4 and 7 magnets in series). After a strong decrease, the generated pressure drop approaches asymptotically the pressure drop in the last magnet string.

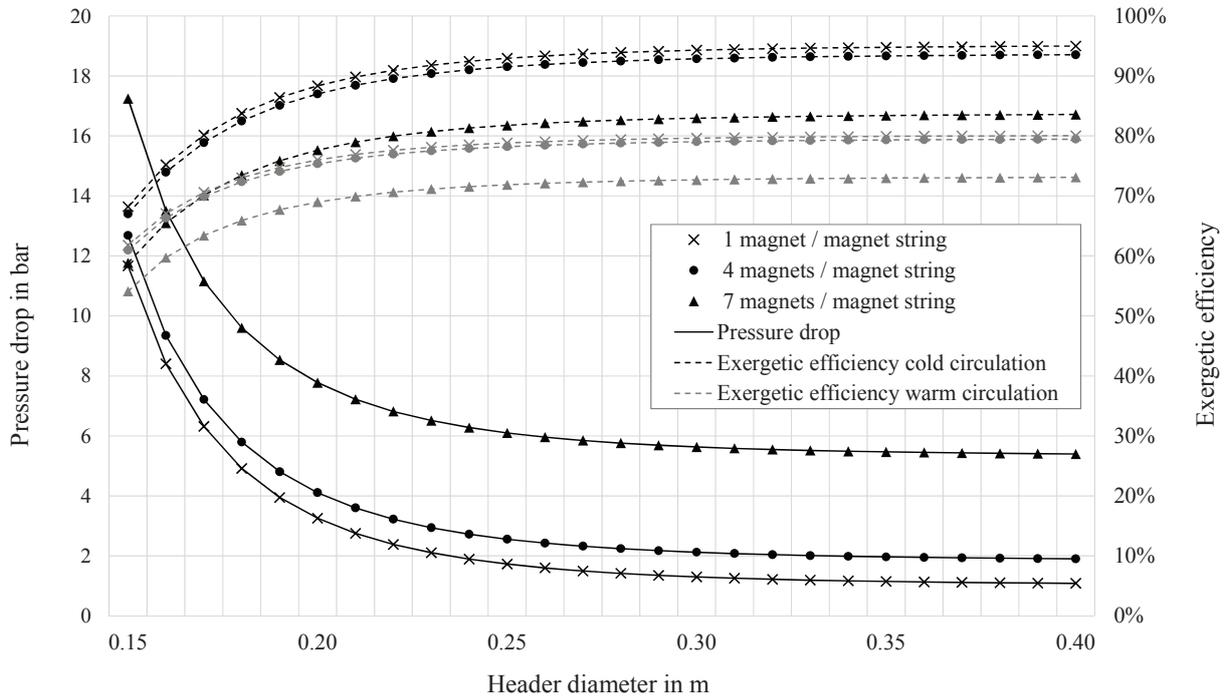


Figure 4. Pressure drop vs. header diameter for different magnet string lengths

The diagram in Figure 5 shows the distribution of exergy losses of the BSC cycle for the cold circulator cycle and the warm circulator cycle and different lengths of magnet strings. For the header diameter, a value of 0.25 m was chosen arbitrarily. Every bar in the diagram represents the sum of the exergy losses.

The order of the accumulated exergy losses in every bar is the same. Starting with the bottommost, the black part represents the exergy losses due to the non-isentropic compression. The second, dark grey part of the bar represents the exergy losses due to the pressure losses in the cooling channels. The third, medium grey part of the bar represents the exergy losses in the valves. The fourth, light grey part represents the exergetic losses in the internal heat exchanger (only in warm circulator cycles). The fifth, white part of the bar represents the residual exergy losses due to pressure losses in the headers and the thermal shielding piping, the heat transfer, the mixing in the return header and the heat extraction in the heat exchanger at ambient temperature (only in warm circulator cycles).

The table below the diagram in Figure 5 illustrates the exergetic efforts for the refrigeration of one long FCC sector. Line 3 shows the necessary exergy of the BSC cycle consisting of the exergy gained in the heat exchangers connected with the Nelium cycle and the circulation power. Line 4 shows the necessary exergy of the Nelium cycle for extracting the heat from the BSC cycle based on an assumed thermal efficiency of the Nelium cycle of 42%. Line 5 shows the total exergy consumption of the Nelium cycle and the circulator in the BSC cooling cycle. The total exergy consumption also is shown graphically by the black solid line with rectangular markers and the secondary vertical axis. The significantly stronger increase of the cold circulation cycle is caused by the additional circulation power, which has to be extracted at cryogenic temperature level.

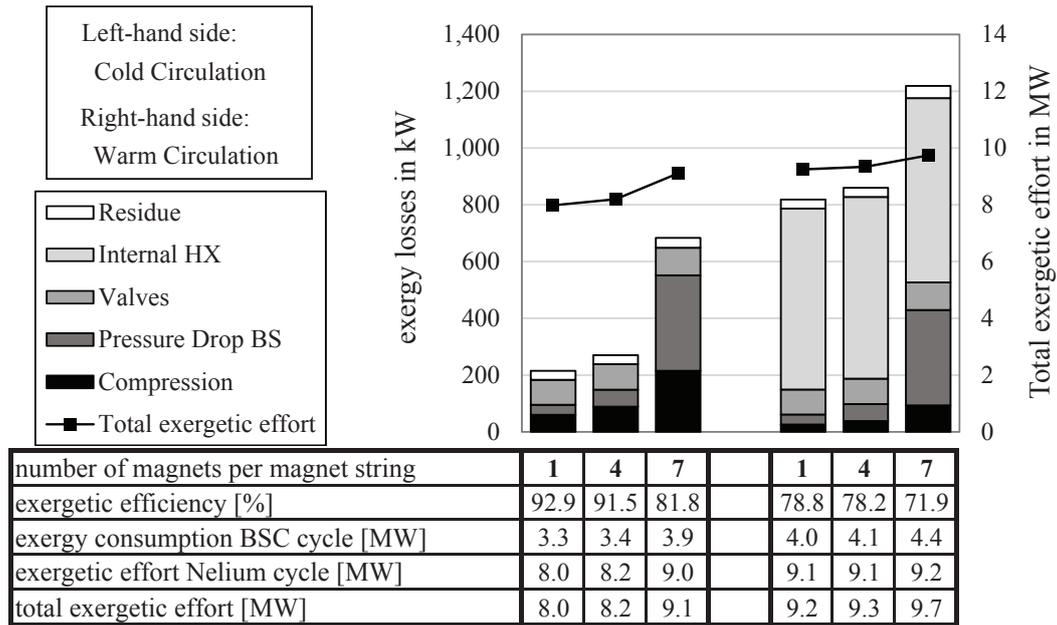


Figure 5. Distribution of exergy losses for different cooling schemes

9. Summary and Outlook

The pressure drop in the BSC cycle dictates the necessary total power consumption of the cryogenic system. Two main parameters are crucial: the header diameters and the number of magnets in series per magnet string. From a certain header diameter on (≈ 0.25 m), the pressure drop almost only depends on the length of the magnet strings.

Following the original idea to minimize the number of control valves, magnet strings containing several magnets in series are necessary. Other applications and cryogenic infrastructure already are supposed to follow the pattern of half-cells of a length corresponding to seven magnets in series. Adjusting the BSC cycle units to this pattern simplifies the assembly, the maintenance and the organisation, paying for these conveniences with higher operation costs.

Circulators at two different working temperatures have been investigated. A cold circulator scheme has advantages, if the pressure drop in the BSC cycle is low (short magnet strings); with increasing length of magnet strings, a warm circulator scheme becomes more interesting. Despite of the higher operation costs additional advantages could make the use of a warm circulator preferable, for example easier handling, less error proneness and the possibility of multi-purpose use (e.g. during cool-down and warm-up).

The investigation of the performances of different circulator concepts including the impact on the Nelium cycle is the next step in the development of a reliable and efficient BSC system.

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