# Novel concept of 60-120 kW at 4 K refrigerator

**S Putselyk**

FERCHAU Engineering GmbH, Butzweilerhof-allee 4, 50829 Köln, Germany

E-Mail: sergiy.putselyk@gmail.com

**Abstract.** In the last 50 years several new large cryogenic facilities with capacities up to 150 kW at 4 K were taken in operation. This required development of new generation of refrigerators/liquefiers with capacities ranges 18-25 kW at 4 K. In order to cover the required cryogenic load, several cryogenic groups, e.g. at CERN, ITER, JLab, SLAC, have to install several refrigerators with maximal power capacities each. As an alternative, development of larger refrigerators up to 120 kW power could be an alternative to the installation of several "small" ones. In the present paper, the novel concept of 60-120 kW refrigerator is considered. The discussion of novel concept will be preceded by discussion of basic design of current "state-of-the-art" refrigerators/liquefiers as well as advanced ones using high efficient heat exchangers and turbines with expected overall thermodynamic efficiencies up to 35%. For the "state-of-the-art" design, we tried to use the standard and already proven technologies in as many as possible cases, though resent developments are also considered. In practice, the refrigerators/liquefiers are designed for the specific load, e.g. pure refrigerator, pure liquefier or somehow in between these limits; however, as the cryogenic heat load for future cryogenic facilities could be different, we tried to apply general process schemes, which could be suitable to refrigerator/liquefier working at 4 K temperature level. The general layouts, screw/turbo compressor stations, 80-300 K as well as 4-80 K boxes are considered. For the advanced design, similar components are used; however turbine and heat exchangers efficiencies are in the range of 82-85% and ca. 98%, respectively. For the novel concept, the applications of turbo compressors as well as several new process schemes of 4 K boxes are considered.

## 1. Introduction

In the last 50 years several laboratories faced the challenging situation that in order to cover the whole cryogenic load, several refrigerators/liquefiers of maximal capacities sizes have to be developed and installed, e.g. at Fermi Lab, Ras Laffan, RHIC, HERA, LEP/LHC, LHD, SLAC, ITER. In the future, it could be also the case that even larger cryogenic capacities will be required, e.g. LHC upgrade, FCC, TESLA/ILC.

There are two limitations, which could slow down the development of new generation of refrigerators, i.e.: i) cryogenic heat loads, which are distributed over large distances, and ii) technology development for large refrigerator components. The first one, which implies that it would be more reasonable to install "small" refrigerators distributed over some distance, is difficult to avoid due to distribution of cryostats or cryomodules over large distances, though for future tokamaks or probably for LHC upgrade, Future Circular Collider or TESLA/ILC, "local" large cryogenic consumer will be present and installation of 60-120 kW refrigerator could be justified. In order to overcome the second challenging limitation, the reliable and well-proved components must be applied in order to have high availabilities, otherwise the wish to install several "small" 25 kW refrigerators will dominate (for two 25 kW refrigerators it would be possible to operate the cryogenic system with "part-load" operation, i.e. with one 25 kW refrigerator, for the case if repairing actions on the second one are required). For that reason, we tried to apply already developed and tested components in order to reduce the risks of component malfunctions or required R&D activities.

Advanced design, which is based on high performance turbine (82-85%) and heat exchangers (98%) is considered as the next development step.

For the refrigerator with novel concept, the applications of turbo compressors as well as new schemes with high efficient turbines and heat exchangers are considered.

It is worth to note that practical refrigerator design is somehow different from one presented here and includes the following steps:

 a) Design input parameter is expected heat loads at different temperature levels and operation modes, e.g. refrigerator, liquefier, mixed mode, shield, etc. This is the most important information, which defines further engineering solutions.
 b) Cycle design for different operation modes.
 c) Hardware design based on the cycle design.
 d) If it is necessary, R&D activities for the development of new components are started.

Thereafter, some iteration and optimizations steps are done based on the real available hardware, e.g. screw compressor, turbines, heat exchangers, etc. In the present paper, the most important initial input information on the expected heat loads is not given, so all further steps of refrigerator design do not follow logically, as it could be seen for experienced refrigerator engineers. However, this could be also considered as an advantageous, because one is free to choose different cycle and hardware designs, as well as to start discussion on some R&D activities needed.  For this reasons, the author hopes that present paper brings some new ideas or initiate R&D activities towards new capacity class refrigerators.

## 2.  General consideration

As the cryogenic heat loads, e.g. pure refrigerator, liquefier or any combinations, e.g.  50/50, is not specified and in any case the 60-120 kW refrigerator (for the further discussion it is noted as 60+ kW one) has to be adopted to some specific heat load, so it is worth to consider different process schemes applied by refrigerator manufacturer. Moreover, due to typical limitations on assembling, transportation or installations, the cold box of refrigerator/liquefier is splitted on two parts, i.e. the high temperature one for temperature ranges 80-300 K, and low temperature one for 4-80 K. It is possible to subdivide the boxes further, like it is done for ISABELLE/RHIC, but for the design simplicity reason we tried to avoid it.

In order to be concise and to limit the scope of present work, the following assumptions are applied:

 a) Design concepts from two manufacturing firms, i.e. *Linde Kryotechnik*, *Air Liquide*, and one institution – cryogenic group of JLab, will be considered. Other firms will be also shortly mentioned, but not discussed in details.
 b) The refrigerators with substantial liquefaction fraction in the temperature range 4-30 K, e.g. working with cold compressors, are not discussed. We consider that it is possible to modify the classical refrigerator schemes like it was done for the modifications of old 18 kW refrigerators used for LEP in order to include the 4-20 K temperature load.
 c) As a "backbone" design, we have chosen the refrigerator/liquefier (later called simply as refrigerator), with $LN_2$ precooling, minimal number of turbines, Kundig's turbine arrangement [47, 48] at cold end, splitted cold boxes (80-300 K and 4-80 K boxes) and modular design of screw compressor set. Other options will also be shortly mentioned.

Preference will be given to the standard components available on the market. If R&D activities are needed, it will be explicitly mentioned.

## 3. Compressor stations

### *3.1. General*

Before one starts discussion the design of compressor stations, it is worth to estimate the GHe flow rates of Low Pressure (LP), Medium Pressure (MP), and High Pressure (HP) streams. As a rule of thumb, we choose HP flow of 5 kg/s, e.g. similar to ESABELLE refrigerator, where 4.5 kg/s was achieved. This value is conservative and with some optimization, see chapters 5-6, it is possible to increase the refrigerator power up to 85 $kW_{eq}$ at 4.4 K. The next step will be to define the ratio of MP and LP flows (for the first approximation, we neglect liquefaction load, which in our case could be up to 0.6 kg/s). This ratio depends on the connection of turbines inside the cold box. Several possibilities exist: a) CERN, 4 new refrigerators for LHC, ratio is ca. 50/50, b) HERA – 50% for LP and 50% for MP, c) LHD – 75% for LP and 25% for MP, d) SSC – 57% for LP and 43% for MP, e) SNS – 30% for LP and 70% for MP, f) CEBAF – 58 % for LP and 42% for MP, g) KSTAR – 50% for LP and 50% for

**Table 1.** Summary of screw compressor (SC) parameters for 60+ kW refrigerator.

| Parameter | First case: 50/50 ratio with interm. pressure | | | Second case: 25/75 ratio with interm. pressure | | | Third case: 50/50 ratio without interm. pressure | | Forth case: 25/75 ratio without interm. pressure | |
|---|---|---|---|---|---|---|---|---|---|---|
| Stage [a] | HP | MP | LP | HP | MP | LP | HP | LP | HP | LP |
| Flow rate, kg/s | 5 | 2.5 | 2.5 | 5 | 1.25 | 3.75 | 5 | 2.5 | 5 | 3.75 |
| Pressure ratio [b] | 3.2 | 2.43 | 5.88 | 3.2 | 2.43 | 5.88 | 7.77 | 2.42 | 7.77 | 2.42 |
| Ideal work, MW | 3.78 | 1.42 | 2.83 | 3.78 | 0.71 | 4.25 | 6.67 | 1.41 | 6.67 | 2.12 |
| Isothermal efficiency | 59% | 62% | 51% | 59% | 62% | 51% | 43% | 62% | 43% | 62% |
| Total volumetric flow [c], $m^3$/s | 6.24 | 7.28 | 18.0 | 6.24 | 3.64 | 27.0 | 15.1 | 18.0 | 15.1 | 27.0 |
| Real SC work, MW | 6.38 | 2.29 | 5.60 | 6.38 | 1.14 | 8.40 | 15.6 | 2.28 | 15.6 | 3.42 |
| Min. required number of SC. | 3.5 | 4.6 | 10.1 | 3.5 | 2.3 | 15.2 | 8.5 | 10.1 | 8.5 | 15.2 |
| Chosen number of SC. | 4 | 5 | 10 | 4 | 3 | 16 | 9 | 10 | 9 | 16 |

[a] For the case of LP, MP, and HP screw compressors, the following models are chosen for the comparison: HP – Howden WLVi321/1.93, displacement 1.774 $m^3$/s, motor power 1.864 MW, MP – Howden WLVi321/1.65, displacement 1.577 $m^3$/s, motor power 0.671 MW, LP – Howden WLVi321/1.93, displacement 1.774 $m^3$/s, motor power 0.671 MW, which are similar to ones given in the reference [1-3]. For the case if no intermediate pressure is used, only LP and HP with the same parameters are chosen.
[b] The inlet temperatures for LP and MP is 308 K and for HP – 313 K. Inlet pressure for LP line – 1.02 bar, MP- 2.47 bar. Intermediate pressure is chosen as 6 bar and HP outlet – 19.2 bar.
[c] Volumetric efficiencies are 0.87 for LP and HP stages as well as 0.89 for MP one.


MP. There is a wide range of ratios between the LP and MP streams and designer is free to choose any for each application. For the further discussion, we will chose two limiting case, i.e. 50/50 (CERN, KSTAR, HERA) and 75/25 (LHD).

The next important design issue is the connection scheme of screw compressors. Two schemes are typically applied: i) without intermediate pressure, i.e. LP stream is compressed by low pressure screw compressors, and after that the MP flow is added with following compression of total flow by HP screws (CERN, SSC, KSTAR, SNS), and ii) with intermediate pressure, i.e. LP and MP streams are compressed by their compressors to some intermediate pressure (typically 3-5 bar(a)) and after that total flow is compressed to HP by HP screws (LHD, HERA, CEBAF, Labarge Refrigerator). Though in some cases the first scheme needs fewer compressors, the second one gives several advantages: i) more flexibility at the part load operation mode, ii) higher isothermal efficiency due to operation at lower compression ratio, and iii) somehow lower wear and tear and therefore longer Mean Time between Maintenance due to operation at lower pressure ratio.

Table 1 compares two cases of flow ratios (50/50 and 75/25) for the screw configuration with and without intermediate pressure. The following could be noted:

a) Total ideal electrical power consumption is nearly equal for the cases with and without intermediate pressure; however the real power is lower for the case if intermediated pressure is used. This is related to the fact that isothermal efficiency for HP screws is ca. 16% higher at compression ratio 3.2 than the one at 7.77. For the flow ratio 25/75, the total energy saving is also noticeable, though it is less significant as for the case of flow ratio 50/50.

b) The total number of screw compressor (SC) for the cases with and without intermediate pressure ratio and 50/50 flow ratio is the same, i.e. 19. However, for the case of 25/75 flow ratio, less number of SC is required for operation with intermediate pressure (23 versus 25).

So, operation with intermediate pressure brings advantage due to lower electrical power consumption and in some cases to lower number of installed SC.

During the part load operation modes and assuming *floating pressure control* scheme (see chapter 5) is applied, the HP, MP (and if applicable intermediate pressure) are reduced. In the case of intermediate pressure application, more flexibility is available particular for HP ranges below 10 bars, e.g. switching off HP screw stage, reducing number of operating LP screw compressors.

As an alternative it is possible to operate with lower number of SCs, if one installs SCs with longer shaft, e.g. Howden 321/2.20 (swept volume is increased by ca. 20% in comparison to 312/1.93) or ones of largest size, e.g. Howden 510/1.93. Though in these cases the extensive testing with helium as well as re-design of Bulk Oil Separator (BOS) will be required.

It is worth to note that two other firms, e.g. Aerzen and Mycon/Mayekawa offer similar products.
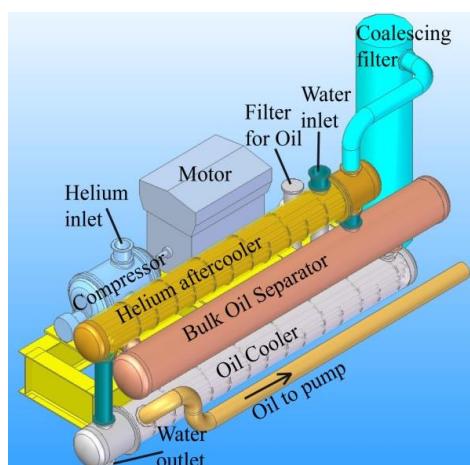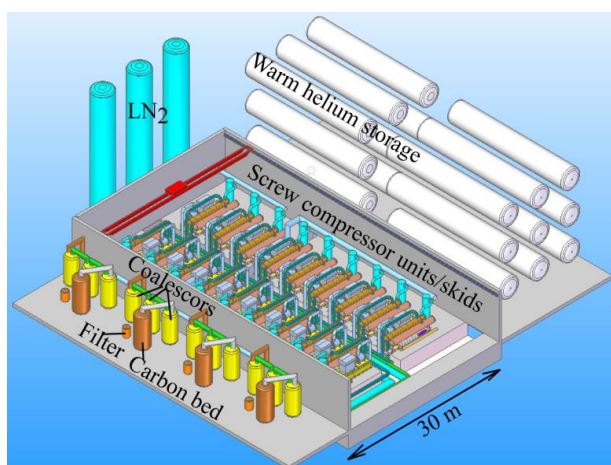
**Figure 1.** The screw compressor unit.



**Figure 2**. The simplified view of screw compressor building.

E.g. Aerzen 536 serie (H, M, or B) were extensively used e.g. at CERN or 436 and 236 series at KIT, Germany. The Mycon/Mayekawa offers 320 series (S, M, L, LL) of similar performance as well as 400S, though larger units of 400 (M, L, LL, XL) series could be also applied in the future.

*3.2. Screw compressor unit*
In order to safely operate the screw compressors with oil injection, it is necessary to have supplementary equipments, which typically includes: i) bulk oil separator (BOS), ii) helium after cooler for cooling down to room temperature, iii) oil cooler, iv) oil pump with corresponding equipment, e.g. electrical cabinets, filter, v) motor and vi) two or three coalescors (typically only for last compression stage), see Figure 1. Two possible realization exists: i) either all components are laid over the floor, or ii) all they are mounted on one mechanical frame together with motor and screw compressor. The first scheme was often applied in the past, e.g. for LHC or HERA and required good access for the components for assembling/disassembling activities. The main idea behind was to reduce the dis- and assembling time due to activities only on few components; however in practice this time is approximately the same for the cases of activities on single components or dis- and mounting of the whole frame. The second scheme brings the following advantages:
 a) Standardized interfaces of the frame to other components and as consequence faster assembling/disassembling.
 b) Lower floor footprint.
 c) Modular design allows faster installation and easiness of further R&D for manufacturing firm.
   The modular design was followed by Mycon/Mayekawa company starting for the small units and nowadays for the maximal one. Unfortunately there is no other company, which offers helium compressor unit in their portfolio, so other firms, e.g. *Air Liquide* or *Linde Kryotechnik*, as well as some institutions, e.g. JLab, have to rely on their own expertise in designing and assembly of compressor units.
   Enormous progress in the last 30 years was done by cryogenic group at JLab, which tried to develop and optimize the compressor units. The compressor unit could be produced either at JLab workshop or according to the JLab's drawings by some manufacturing and assembling firm. Their present experience is based on four generations, starting with one for CHL-1, following for SNS [4], then for NASA and finally for CHL-2 [1-3]. Very low footprint and high compactness allowed installation of compressor units in small buildings. It is worth particular to mention that high compactness was achieved mainly due to two developments:
 a) Very compact BOS: separation of oil/helium mixture occurs in three states, i.e. in opened horizontal tube (which is also applied in other BOS) then in two horizontal tubes, where helium flow through holes to the top of BOS and finally at the top horizontal tube, where helium flows inside through small holes [1-3].
 b) Oil is pumped only for bearings and slide valve control, so total oil flow through pumps is approximately reduced by factor 3 to 5 and other part of oil is circulated by pressure difference across the SC. It is worth to note that in this case the high temperature limitation of oil/helium mixture after compression (T=370 K) has to be tightly controlled.
   The innovation to avoid the any coalescing filter elements inside the BOS also allowed more flexibility on operation with different pressures, though somehow higher oil carry over into the helium

**Table 2**. Main turbo compressor parameters.

| Name [a] | Tinlet/Toutlet | Pinlet/Poutlet | Flow rate |
|---|---|---|---|
| LP comp. [b] | 25/83°C | 10.5/15.5 bar | 84.8 kg/s |
| HP comp. [b] | 25/125°C | 15.4/28.7bar | 84.8 kg/s |
| Main comp. [c] | 826/850°C | 49.5/51.2 bar | 200 kg/s |
| Supp. comp. [c] | 236/258°C | 49/53.5 bar | 56.8 kg/s |
| 1/3 model [d] | 30/57°C | 8.83/10.7 bar | 12.2 kg/s |
| LP comp. [e] | 60/144°C | 8/16.2 bar | 11.7 kg/s |
| HP comp. [e] | 60/144°C | 15.7/32 bar | 11.7 kg/s |
| LP comp. [f] | 23.9/94.3°C | 6.5/10.3 bar | 4.76 kg/s |
| HP comp. [f] | 26.7/97.8°C | 10/15.8 bar | 4.77 kg/s |

[a] Helium blowers (typically of radial type and flow rate 4-13 kg/s) used for helium circulations are not considered.
[b] Oberhausen 2 helium turbine plant (EVO) [7, 8], HP and LP compressors have 15 and 10 stages, respectively, both 100% reaction. Total operation time is ca. 24,000 hours.
[c] High temperature helium test facility (HHV) [9, 10], 8 stage main compressor and cooling radial-type supplementary compressors. Total operation time is ca. 1,100 hours.
[d] 1/3 model for the GTHTR300 [9]
[e] HTR10 [10, 11]
[f] HTR10GT [12, 13]

aftercooler is considered. For the design, the operation of aftercooler with some oil condensed onto the surfaces was not a challenging issue due to slightly increased surface transfer area of this heat exchanger; however, a coalescing filter had to be additionally installed downstream the aftercooler [2]. The complete screw compressor unit is shown in Figure 1.

The width and length of skid with motor and compressor are ca. 3 and 6 m, respectively. The screw compressor sets for LP, MP or HP are nearly identical, main difference is in the dimension of helium aftercooler.

*3.3. Turbo compressor unit*
The novel concept refrigerator will use the turbo compressors as well as advanced process schemes of 4K box. In order to substantially increase an overall refrigerator thermodynamic efficiency, the high efficient compressors are of paramount importance and turbo compressors could be applied [14]. The helium cryogenics could profit from experience gained from the development of the helium turbo machinestry for Generation IV nuclear power plants. In this chapter, the very short overview of helium turbo compressors is considered first, which is followed by basic design of turbo compressor units for 60+ kW refrigerator.

Table 2 lists some helium axial compressors used either for long-term operation or for prototype tests. It is worth also to note that for "small" flow rates of 1-5 kg/s, the radial compressors were also applied as supplementary devices, e.g. for turbine bearing or blade cooling.

The turbo compressor system for the novel concept will have 7 stages with final pressure of ca. 25 and 43 bar(a) after the 6th and 7th stages, respectively and recooling heat exchangers to 300 K after each stage. For rough estimation of the main compressor parameters, the following is assumed: a) pressure ratio is 1.7 for all stages and no pressure losses occur in connecting piping and re-cooling heat exchangers b) polytropic efficiency is 88% for each stage [7-13], c) flow is 5 kg/s and is the same for each stage.

Estimations of compressor operational parameters could be summarized as:
a) Outlet temperature is ca. 382 K (109 °C). If compression is isentropic, this temperature is ca. 371 K.
b) Isentropic efficiency of helium compression – ca. 87%.
c) Compressor work is 2.13 MW and isothermal one – 1.65 MW, so the isothermal efficiency is ca. 77% and energy released to the environment after each compressor stage is ca. 0.48 MW. For the case, if helium compression is ideal (isentropic), the maximal isothermal efficiency could be estimated as ca. 90%.

Based on available prototypes, the compressor dimensions are estimated and presented in Table 3.

As one can note, the estimated helium compressor sizes are well within the present technologies for steam and air ones. The foot-print is small in comparison to the screw compressors and helium purification system is also simpler due to absence of bulk oil applied for screw compressor cooling. It

**Table 3.** Estimated sizes for some turbo-compressor components [a].

| # | Tip diameter (mm) | Rot. Blade Height (mm) | Stat. Blade Height (mm) | Stage number | Net length (mm) [b] |
|---|---|---|---|---|---|
| 1 | 1150-1320 | 40-54 | 31-43 | 4 | 940-1220 |
| 2 | 880-1020 | 30-42 | 24-33 | 5 | 450-590 |
| 3 | 680-780 | 23-32 | 18-25 | 6 | 410-540 |
| 4 | 520-600 | 18-24 | 14-19 | 7 | 370-480 |
| 5 | 400-460 | 8-19 | 11-15 | 8 | 330-420 |
| 6 | 310-350 | 10-14 | 8-11 | 9 | 280-370 |
| 7 | 240-270 | 8-11 | 6-9 | 10 | 240-310 |

[a] Shaft rotation speed is 250 Hz and is the same for all compressors.
[b] Axial length, i.e. all rotor and stator blades.

would be possible to install several turbo compressor stages on the single shaft with one electrical motor; however refrigerator operation will be more flexible, if each turbo compressor is connected with electrical motor, i.e. in this case helium stream addition or removal between stages would be easier. Operation with reduced power, e.g. during refrigerator part load operation, is performed by the variation of pressures and as consequence helium mass flow rates.

*3.4. Screw compressor system and building layout*
Besides the screw compressor unit, the compressor system also includes the final cleaning system, water cooling system, gas handling system (e.g. warm gas storage, piping), high power electrical system, oil preparation system, building, etc. and could be shortly summarized as following:
• The total installed electrical power should be ca. 20 MW,
• Water cooling: the chillers with flow rate of ca. 50 m$^3$/min for ΔT range of 5.5-8.3 K are required.
• The 5 t crane must be installed for lifting the weight of one compressor unit.
• The helium gas and liquid storage will depend on the final users.
• The final oil/water vapor removal system is standard and includes three coalescors, one carbon adsorber and final dust filter.

Figure 2 shows the simplified sketch of screw compressor building. For the present screw compressor design, the most optimal building layout and other components were developed by JLab cryogenic group [1-3] and its main features are also applied for 60+ kW refrigerator, e.g.:
a) Two rows of SC with pipe tranches in between. This allows installation of additional piping or interconnection of SC units between HP, MP or LP lines.
b) The set of helium gas cleaning system includes three coalescors, one carbon bed and dust filter. Three sets are needed for operation and one for reserve. Installation is outside the building; however if weather conditions are not suitable, indoor installation is also possible.
c) The building width is ca. 30 m, which allows installation of 5 tons crane.

For the present design, 18 compressor sets are shown, and with this modular SC design it is possible to add additional ones.

## 4. 80-300 K box (High Temperature Box)

*4.1. General*
As it was mentioned in previous sections, the design of 80-300 K box with LN$_2$ cooling is chosen. Such design allows scaling-up with the present refrigerator technology. For the simplicity, only few cases are considered, i.e. operation with 50/50 and 25/75 mass flow ratios. Different operation modes, i.e. pure liquefaction or refrigeration, or others will not be considered in details, and for theoretical knowledge on refrigerator design one can refer to [5]. The process and layout design of the box in order to apply present state-of-the-art technology components are discussed.

*4.2. Precooling with LN2*
The LN$_2$ precooling scheme was chosen due to following advantages: a) smaller cold box with heat exchangers and screw compressors for the given total refrigerator capacity, b) somehow more stable operation over large operating range and wide turndown capability.

**Table 4.** Heat exchanger parameter for 80-300 K box of 60+ kW refrigerator.

| Parameter | HP/LP HX | HP/MP HX | HP/GN$_2$ HX | HP/LN$_2$ HX |
|---|---|---|---|---|
| Flow naming [a] | HP/LP | HP/LP | HP/LP | HP/LP |
| Heat transfer, MW | 2.67 | 2.67 | 0.194 | 0.170 |
| $\Delta T_{hot}/\Delta T_{long}$ [b] | 2.28% | 2.28% | 2.28% | - |
| Capacity ratio | 0.989 | 0.989 | 0.867 | 0 |
| Effectiveness | 97.8% | 97.8% | 97.8% | 95.0% |
| Minimal NTU | 35.6 | 35.6 | 14.4 | 3.0 |
| Minimal UA, W/K | $4.39 \cdot 10^5$ | $4.39 \cdot 10^5$ | $1.3 \cdot 10^4$ | $7.88 \cdot 10^4$ |
| $\Delta T_{log}$ | 6.0 K | 6.0 K | 14.9 K | 2.25 K |
| Core dimension (W·H·L) [c] | 1.45·2.4·7 m | 1.1·2.4·7 m | 1.76·0.25·4 m | 0.4·1.5·1.5 m |
| Appr. heat trans. area (HP/LP), m$^2$ | $8.73 \cdot 10^3/8.73 \cdot 10^3$ | $6.62 \cdot 10^3/6.62 \cdot 10^3$ | $4.25 \cdot 10^2/8.51 \cdot 10^2$ | $3.23 \cdot 10^2/3.23 \cdot 10^2$ |
| Heat transfer coefficient [d] | 67-117 W/m$^2$K | 88-133 W/m$^2$K | 58-69 W/m$^2$K | 660-820 W/m$^2$K |
| Estimated $\Delta P$, mbar | 1.3-3.8/23-40 | 2.1-5.0/15-40 | 0.8-2.2/2.6-5.1 | 7.2-10.1/- |

[a] The temperature, pressure and flow rates are presented in Figure 5.
[b] Ratio of temperature difference between warm and cold flow at the warm side of HX to the temperature difference of the cold flow at inlet and outlet of HX.
[c] Except HP/GN$_2$ HX, where double layer is used for GN$_2$ flow, all HX with single layer are used, i.e. layer pattern is e.g. HP-LP-HP-LP
[d] Heat transfer coefficient strongly depends on which correlation for the calculation is used, see following chapters of present paper for more details.

Other arguments for the choice of LN$_2$ could also be considered [5]; however for the development of new large scale refrigerator, the conservative design is always advantageous and therefore the above mentioned design arguments have to be applied. The schematic view of 80 K box is showed in Figure 3.
The box has the following features:
a) Separate heat exchangers (HX) for HP-MP, HP-LP and HP-GN$_2$ flows. It is very favorable to separate flows in order to avoid possible flow mal-distribution inside the heat exchanger [6]. In addition, due to manufacturing limitation, it is not possible to produce HP-MP-LP heat exchanger as single block.
b) The plate-fin subcooling HX is located inside the LN$_2$ vessel, which is mounted onto the frame together with other HX.
c) The additional LN$_2$ vessel for the operation with thermos-syphon is not installed for two reasons, i.e. it needs additional space, and operation of thermosiphon loop with large flow rates needs to be tested and therefore some R&D activities must be done.
d) Two 80 K carbon absorbers are located outside to the box for two reasons, i.e. space limitation inside the box, and easiness of carbon change.
e) Figure 4 shows very simplified process scheme of 80-300 K box.
It is also worth to note that LN$_2$ logistics is also possible, see e.g. LHC cool-down.

*4.2.1. Assembling and installation.* Three heat exchangers, HX1A, 1B and 1C, should be installed on separate frame, 3.0·3.0·11 m and have to be transported to the final destination place. Other components, e.g. optional floating radiation shield, vacuum vessel top plate with valves, LN$_2$ vessel, could be also partly pre-assemble at manufacturing workshop, if transportation of components with the width of more than 3.5 m is possible (otherwise these components must be installed *on-site*). After final assembling, the box dimensions should be in the range of 15·5.3·3.7 m.

*4.2.2. Heat exchangers.* Four plate-fin heat exchangers are applied for the cooling down gaseous helium (GHe) to 80 K. Two HX are used for the LP and MP streams, one for gaseous nitrogen (GN$_2$) to GHe, and one for boiling nitrogen, i.e. LN$_2$ to GHe (also sometimes called subcooling HX). For the simplicity of discussion, the internal parameters of all HX including ones used inside the 4-80 K box, are the same, i.e.: fin thickness – 0.152 mm, separating plate thickness – 0.150 mm, fin height (sometimes also called as "length") – 6.36 mm, fin pitch – 4.57 mm and distance between successful cuts – 1.50 mm (serrated fins are used). The compactness is in the range 770-850 m$^2$/m$^3$, and
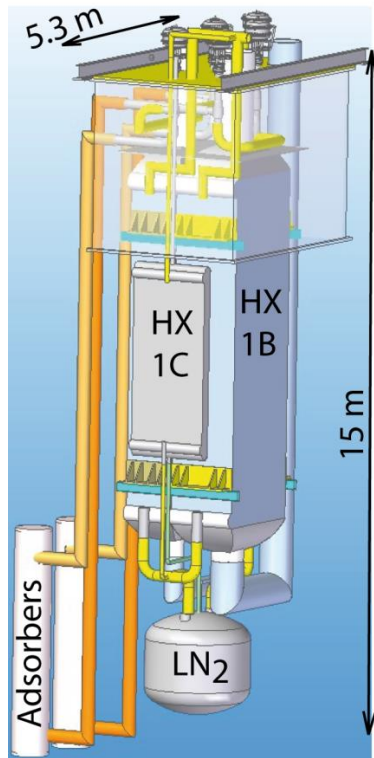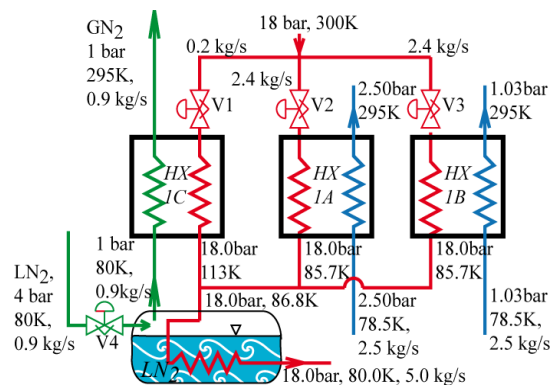
**Figure 3**. The schematic view of 80-300 K box.



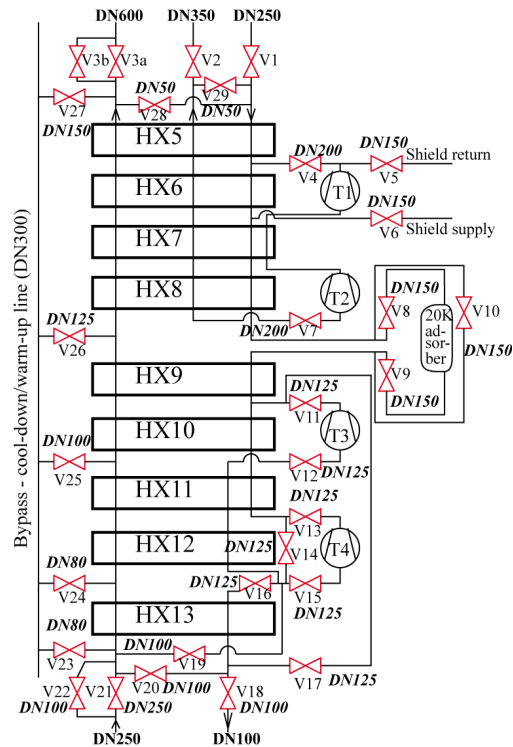**Figure 4**. Simplified process flow diagram for the 4-80 K box.



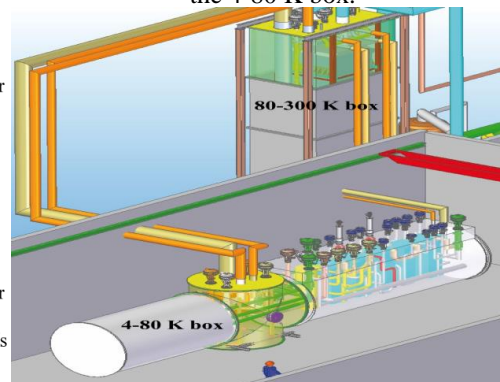**Figure 5.** The simplified process scheme of 80-300 K box.



**Figure 6.** The simplified view of 4-80 K box of 60+ kW refrigerator.

equivalent diameter – 4.71-5.1 mm (depending on which correlation formula for the heat transfer is used). The specific HX parameters are shown in Table 4. With the present HX design it is possible to obtain required heat transfer with sufficient margin. For the calculation, $\Delta T_{log}$=6.0 K is conservatively assumed. In order to keep the HX cross-sectional dimensions within the manufacturing tolerances, e.g. for *Linde* – 1.5·3.0·8.2 m, for *Five Cryo* – 1.3·2.0·7.8 m, *Kobelco* – 1.35·1.35·7.0 m, the single layer scheme for the low and medium pressure lines (except for $GN_2$ flow in HX1C) is chosen. The pressure losses are also within acceptable ranges.

*4.2.3. Other components.* Several large valves of DN150 and/or DN200 sizes must be installed. These valves could be manufactured, e.g. WEKA or Velan firms, and could be considered as proven technology, since several such valves are already at operation at other installations. The two carbon absorbers with minimal volume of 4.5 m³ are installed externally to the box. This installation allows freely choosing the required volume of carbon beds and different adsorbing materials could be applied [5]. $LN_2$ vessel (2.3 m diameter and 2 m height) with incorporated the subcooling plate-fin heat

exchanger is installed at the bottom of the cold box. The volume of the $LN_2$ vessel must be large and should contain at least 1000 liters above the HX surface to guarantee 15 minutes of uninterrupted operation, if $LN_2$ supply is not available. As an option, it is possible to install floating radiation shield, e.g. by mounting the plates on the frame with heat exchanger. Vacuum vessel will consist of several parts. On the upper plate, most of pipe penetrations and valves will be installed. It could be separated on several parts and lowered in order to get access to inner components of the box.

**Table 5**. Some turbine parameters for the case of operation without $LN_2$.

|  | Maximum capacity (50/50) | | Liquefaction | |
|---|---|---|---|---|
| Turbine number | #1 | #2 | #1 | #2 |
| Inlet/outlet temperature [K] | 207/189 | 115/75 | 251/229 | 133/86 |
| Inlet/outlet pressure [bar] | 18/13.1 | 13.1/2.37 | 18/13.1 | 13.1/2.37 |
| Mass flow [kg/s] | 1 | 1 | 1 | 1 |
| $\Delta h$, [kJ/kg] | 95.1 | 211.1 | 115.8 | 247.5 |
| $\Delta h_{is}$, [kJ/kg] | 129.8 | 299.1 | 157.0 | 345.5 |
| $\eta$ | 73.2% | 70.5 | 73.7 | 71.6 |
| Power [kW] | 95.1 | 211.1 | 115.8 | 247.5 |

*4.3. Precooling with turbines*
It is worth to consider an alternative to the $LN_2$ cooling scheme, i.e. usage of turbines. The helium flow through the turbines is typically maximal at two operation mode, i.e. refrigerator cool-down and liquefaction. As the refrigerator cool-down mode is considered to be less critical than the liquefaction one, (it is possible to cool-down the refrigerator in a longer time period) so only maximum capacity and liquefaction modes are further considered. Typical flow through turbines is ca. 20 % of total flow, and outlet temperature of the last turbine is fixed at ca. 80 K temperature level. The HX is subdivided into 4 parts, and turbine flow bypasses the second and the last parts. Table 5 shows some typical turbine operating parameters for two operating modes. The present maximal power turbine produced by Air-Liquide is 180-200 kW range, (for more detailed discussion on turbines, see chapter 5.2.5), so it could be possible to operate the box at the standard operation mode, e.g. 50/50 ratio of refrigerator/liquefier power. In this case, the second turbine should have ca. 210 kW, which is marginally higher (ca. 17%) than present 180 kW ones. However for the case of pure liquefaction mode, the second turbine power is ca. 247 kW and R&D activities on the new turbine class would be necessary. As an alternative to the helium turbine, it could be possible to use $N_2$-refrigerator, so conventional and proved technology could be applied, though the whole cryogenic system would be somehow larger and more complicated.

**5. 4-80K box**

*5.1. General layout*
As it was shortly mentioned in the previous chapters, the general layout of 60+ kW refrigerator generally follows the optimized design of CHL-2 at JLab. This design is typical for the large refrigerator installations in the North America, e.g. 80-300 K box is installed outside the building, piping is mainly on the ground level (not underground in trenches), gas cleaning system outdoor, $LN_2$ usage for precooling. The 4-80 K box is installed in the adjacent building to the compressor hall, and 50 ton crane is needed for the safe operation with this box, see Figure 6. The horizontal LHe vessel of 80 m$^3$ volume with the subcooling heat exchanger is installed outdoors. Depending on the operation requirements, e.g. large liquefaction rates, the installation of one or two additional LHe vessels of up to 150 m$^3$ each could be also possible.

*5.2. Process design*
*5.2.1. General.* Before one starts discussion of the process flow schemes, see Figure 4, T-S diagrams and box components, it is worth to say few words on the basics of helium refrigeration technology and last development trends.

The ultimate goal of any refrigeration system is to design an optimal system. The "optimal system" could be very broad interpreted, e.g. minimal operational, capital or maintenance costs, maximum capacity or availability and as a rule of thumb, it could be also possible to define efficient system as ones, which results in low manufacturing and operational costs, which also implies minimization of the supply helium flow to the cold box. As the heat loads are not explicitly defined in the present paper it

would be very difficult to follow the classical approach, e.g. choice of cycle according to the heat loads, high efficiency components, etc., so an alternative methods should be chosen. According to the author opinion, it would not be wrong if one starts discussion the system, which is operated in automatic regime with smooth transitions between different operational modes, e.g. pure refrigeration/liquefaction, part load operation, etc. So, during development of automatic procedures some optimal operational paths are chosen as well as some savings on the costs for operating personal are also possible. For that reason, the good starting point on refrigerator design could be the choice of operational parameters, which will allow maximal automatic operation at different modes.

Therefore, it is necessary to find the indicating parameter for the heat load, followed by choosing the main parameter suitable for indication and control of refrigerator performance between maximal and minimal heat loads.

As an example of indicating parameter for the heat load, one can mention:
a)   For refrigeration cycle – the liquid helium (LHe) level on one common dewar or in separate LHe cryostats. Sometimes heater, which is used to control the LHe level, is used for indicator.
b)   For liquefaction cycle – the LHe level in dewar (or heater used for LHe control).
c)   For system used to supply supercritical helium to customer – LHe level in the subcooling dewar with heat exchanger (HX) for supercritical helium (or again, heater used for LHe control).
d)   For system used to supply supercritical helium to customer but without subcooling dewar with HX (e.g. this dewar is located far away from refrigerator) – the supply temperature or return temperature.
e)   For non-isothermal refrigeration cycle, i.e. operating between two temperature levels – the return temperature.
f)   For refrigerator/liquefier operating with cold compressors – the cold compressor outlet temperature or function calculating the non-isothermal heat load, i.e. GHe mass flow multiplied by enthalpies difference at inlet and outlet of cold compressor. It could be also possible to apply a combination of these two methods or more complicated function.
g)   For mixed refrigerator/liquefier cycle – more complicated function could be applied; however, the LHe level in dewar could be also used for most of the cases.

The parameters used to describe refrigerator power and directly related (e.g. by control loops) to the one used for the cryogenic heat load measurements are supply and return pressure to/from the cold box (if medium pressure is used – it could be also applied). In this case the return pressure from the cold box could be varied in the range 10-40 mbar, while the supply pressure is varied in the range of few bars for operation around one "single" point of the heat load, or 6 to 20 bars if part-load to nominal operation is foreseen at daily basis. So, in the last case, the supply pressure is automatically stabilized somewhere between 6 and 20 bars depending on the required heat load and after that it is slightly varied around the operating point for stabilization of small fluctuation of the customer heat load. The adaptation of the cold box to the new power is performed by the changing of the turbine power by compressor brake circuit, while the helium flow and supply pressure is varied according to the rotational speed of screw compressors (excess amount of GHe is return to storage tanks and *vice versa*)

This is the basic idea of *Floating Pressure Capacity Control* or also known as *Ganni Cycle*. This control method is widely used in gas & steam turbine technologies, and the first application in cryogenic refrigeration is dated in 1964 [15]. Several small liquefiers (or refrigerators) were built e.g. by Sulzer for CERN in 60 and 70ies, which had similar features at least in the turbine circuit. Application to the "large" refrigerator also started approximately at that time, see e.g. 1.5kW refrigerator for FNAL [16, 17], four large Helix-Sulzer liquefiers in USA [18, 19], 1.5 kW refrigerators for LLBL [17] and ORNL [20] and 3 kW for LLBL [21], though not all these refrigerators/liquefiers have been developed to operate in floating pressure control in all circuits (typically, only turbine circuits were designed for that).

One of the good examples of application of floating pressure controls for small refrigerators/liquefier is one at Tacoma Substation for cooling of superconducting magnetic energy storage unit with 2800 refrigerator produced by Koch Process Industries (former CTI-Cryogenics) with gas bearing turbines and maximum capacity of ca. 220 W [22, 23]. Inlet pressure to screw compressors was variable in the range of 0-200 mbar(g) for heat load variation 0-40 W. Another example is a fully automatic "Linde- turbo-refrigerator for MR-tomographs" (more than 50 were built by July 1989) [24]. Similar concept was also applied for some helium refrigerators used for hydrogen target cooling, though references in the opened literature are scarce.

In the end of 80ies and 90ies, this concept was further applied for the large refrigerators, e.g. HERA, LEP, SSC (e.g. ASST-A is re-installed as End Stage Refrigerator at JLab). Though these refrigerators were successfully tested at different operation modes including the part load operation,

they were seldom used in the "classical" way mentioned above, i.e. for fully automatic transition between one operation mode, e.g. from liquefier to refrigerator and *vice versa*, or to part load operation. Two reasons contributed to fact that these refrigerators were operated with floating pressure control around some operation point, and transition to other modes was performed by some additional program (or in some cases also "per hand"):

a) Cleaning of helium gas after compression for large refrigerator imposed the limitation on minimal operational supply pressure. This was related to the fact that Balk Oil Separator was not operated efficiently at low pressure and oil carry over to coalescors occurs, which potentially endangers the whole cleaning system

b) Typically large refrigerators are designed for fixed cryogenic loads, and the variation around the operating point is small, e.g. ±10-15%. Therefore, automatic control for daily heat load operation down to ca. 20-40% of maximal refrigerator load was not required. So, only rarely the refrigerator was settled at heat load operating point of 20-40% for continuous operation. For that reason, the control around one operation point was completely sufficient and there was no need for automatic smooth transition between different modes.

As an additional comment to the last argument, it is worth to note that typically for European market and in many cases for Japaneese one, the large refrigerators, i.e. refrigerators with capacities above ca. 1 kW, are built only for one user and after experiment termination they are also decommissioned (please note that for "small" refrigerators, i.e. refrigerators, which are standard "from-the-shelf" and have relative small "dimension/weight" parameters, situation is completely different: many refrigerators around the labs or institutions were relocated to other experiments!). Only two exceptions are known to the authors' knowledge – HERA/XFEL and LEP/LHC refrigerators. The situation is completely different at the North America market, where several very large-scale refrigerators have been reused for applications with significantly different heat loads (and many of them were re-installed at different institutions). This was one of the main reasons, why cryogenic group at JLab tried to apply floating pressure control for large-scale refrigerators.

In practice these two above mentioned reasons led to some design and operation differences in comparison to the originally developed floating pressure control:

a) Turbine flow coefficients were optimized for limited operation range ca. 80-110%.

b) Supply high pressure (HP) was limited to ca. 14-20 bar. Operation with lower pressures was possible but not for extended time period.

c) Pressure control for suction/discharge pressures at screw compressor was different. For the operation around some point, it was sufficient to control the suction pressure with by-pass valve. The transition from one operation point to another one, i.e. from one box capacity to another one, is performed by changing of HP level (low pressure and intermediate pressure must be also changed), and after that these new pressure levels are kept constant with by-pass valves (one can also imagine it as "step-wise" transition). For classical floating pressure control, the by-pass valves are not used for continuous regulations but are activated at some limiting cases, e.g. avoiding Low Pressure (LP) line get sub atmospheric, avoiding Medium Pressure (MP) line to be too low to protect high pressure screw compressor from too high compression, etc.

Further limitation has been introduced by 40-80 K radiation shield. Very often this heat load and temperature ranges are fixed independently for any operation mode, e.g. full/part load or refrigerator/liquefier operation mode. For the floating pressure control, this implies limitation during part load operation because cooling power of corresponding turbine used for covering of shield load is fixed. However it is possible to cancel this limitation if one accepts larger temperature difference during part load operation. (In many cases due to expansion of shield flow over the turbine, the higher outlet temperature from the shield leads to higher inlet temperature of turbine). In this case the following situations are possible:

a) Larger temperature difference over turbine inlet and outlet, could be covered with smaller GHe flow.

b) If shield is operated at ca. medium pressures, it is possible to expand flow from shield directly into MP line at higher temperature. This will not disturb the box (at least not significantly), because part of this load will be covered by $LN_2$ or turbine string operating in the range 80-300 K. This scheme is realized e.g. at CHL-2 at JLab [25, 26].
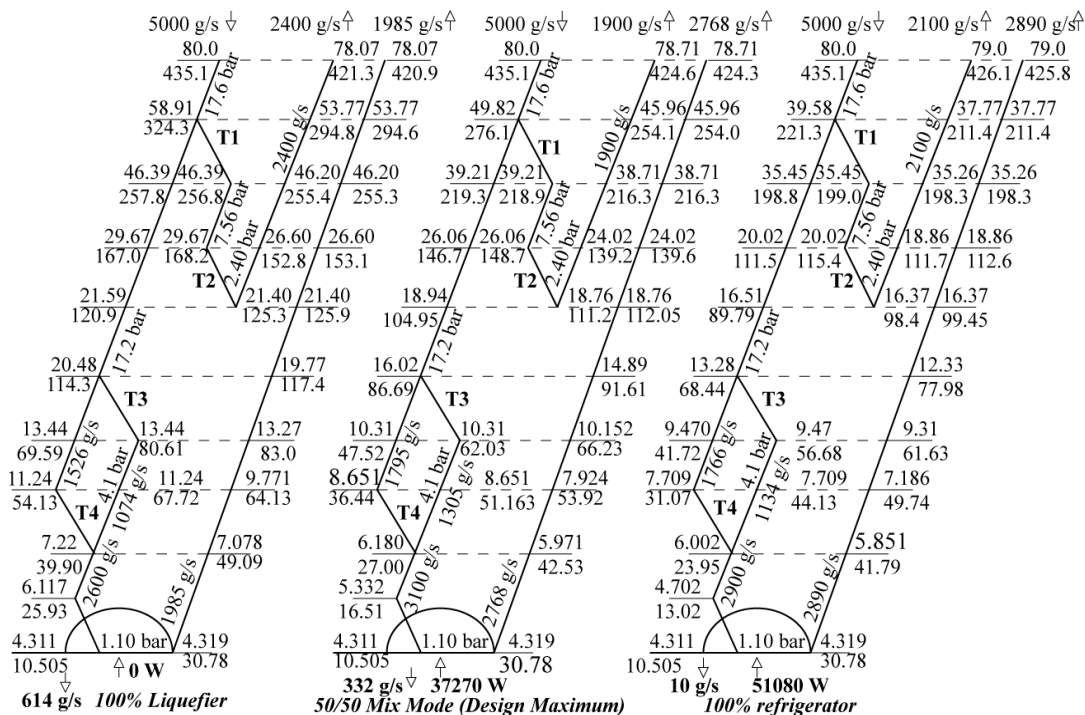
**Figure 7**. T-S (temperature versus entropy) diagrams for 100% liquefaction, 50/50 Mix. (also called Design Maximum) and 100% refrigeration modes.

CHL-2 at JLab, National Superconducting Cyclotron at MSU, SNS at Oak Ridge, RHIC at BNL, NASA at Houston.

For the successful design and operation of 60+ kW refrigerator, it is important to implement floating pressure control scheme at all design stages starting with basic design and finishing with detailed engineering stage. In the further chapters, several comments on its application will be given.

*5.2.2. Process flow diagram for 4-80 K box.* The process flow diagram for typical refrigerator or liquefier is similar for many refrigerators, see Figure 4 (please note that for the simplicity of discussion, unless other explicitly mentioned, any refrigerator working as 100% refrigerator, 100% liquefier, at any combination of refrigeration and liquefaction loads, or part load operation mode, is called as "refrigerator").

The HX with changing flow capacities ratio (HP is smaller than LP one and vice versa) are connected on one after another, where e.g. HX5 has LP flow with smaller heat capacity, while HX6 has smaller HP flow. In the 15-30 K temperature range, the "20 K adsorber" with the by-pass valve is located. For temperatures below 20 K, two possible arrangements of turbines are possible:

a)  Parallel flow (also called Kundig's [47, 48]) arrangement: it was used for HERA, SSC, LHD, CERN (Linde company).

b)  Serial flow through two (or more) turbines: this arrangement is used mainly by Air Liquide company, e.g. at CERN. At the present time this arrangement is rarely used, because due to development of high power and high pressure ratio expansion turbines, it is possible to use only single turbine, e.g. Las Raffan, KSTAR, ITER. For the Las Raffan, KSTAR and ITER refrigerators, one turbine expands flow directly to LP (or MP) line, while the coldest one expands from high pressure to ca. 3-4 bar(a)

For the future discussion, the parallel flow arrangement is chosen, because it seems that during the part load operation it brings a bit more operation flexibility. Additionally for the Kundig's arrangement, it is possible to use the subsonic turbines, which are simpler than their supersonic alternatives.

In the past, for the large scale refrigerators, also serial (2 or 3) arrangement of JT-valves was sometimes applied. However, due to development of high power turbines, the high temperature JT-
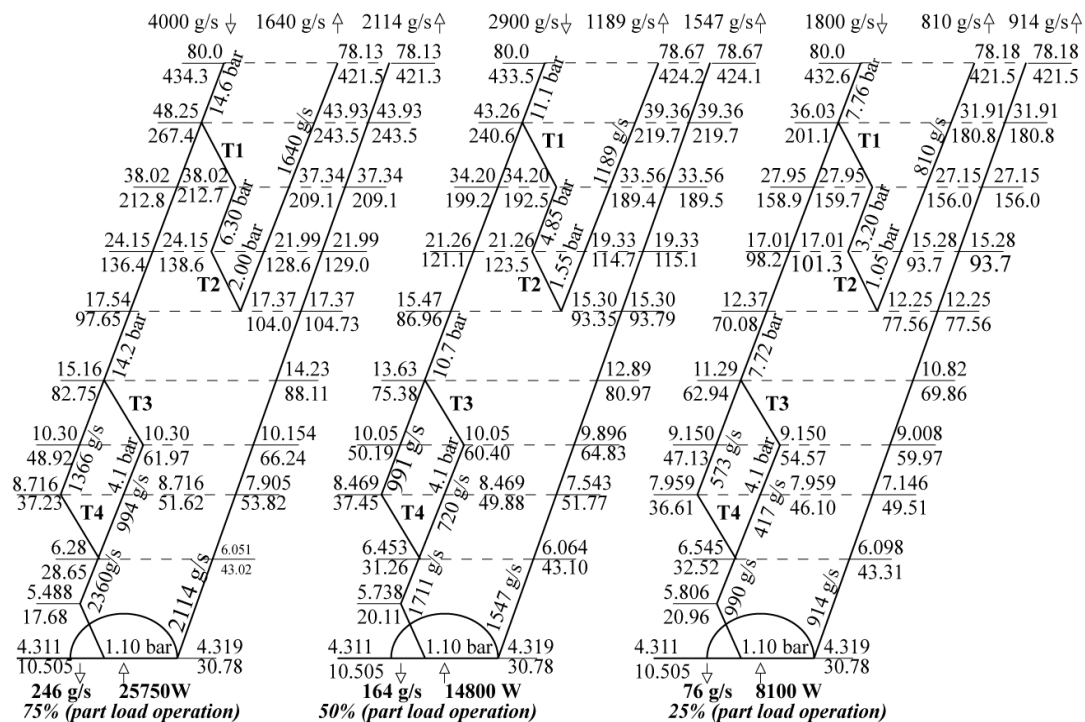
**Figure 8.** T-S diagrams for part load operation modes: 75% (left), 50% (center) and 25% (right).

valve(s) were replaced by turbine, see e.g. modification of Isabelle/CBA/RHIC refrigerator. It is worth also to mention that Linde company also uses scheme with single turbine, if the refrigerator works as a pure liquefier, e.g. refrigerator in Algeria or in Labarge, USA.

And last but not the least, it should be also noted that parallel flow could be easily arranged in the serial flow, if one of the turbines is switched off. In this case, low temperature turbine is typically operated, and the operating temperature and pressure ranges should be chosen according to the operation mode, i.e. pure liquefier or refrigerator, part load, etc.

*5.2.3. T-S diagrams.* A good starting point for the refrigerator design could be T-S (temperature versus entropy) diagrams, see Figure 7 and 8. These diagrams give information on the cryogenic heat load requirements as well as some limitations on "hardware" design, e.g. on turbines and heat exchangers for each particular design mode. It should be particularly stressed that T-S-diagrams are only informative and refrigerator operation must not be "fixed" to them. The reasons are: i) actual "hardware" (e.g. turbine, heat exchangers, etc.) are always somehow different from the ideal design values, and ii) real heat load is either never measured accurately or could change during operation. In practice the T-S diagrams just limit extremities (or it could be called as "operational field"), e.g. limits "left" and "right" as pure refrigerator and liquefier, "top" and "down" as maximal capacity and part load operation (25%). Further limitations could be also given by other modes, e.g. 80 K stand-by, maximal liquefaction with some refrigeration load, etc. The final goal is to find operational parameters for the heat exchangers and turbines for all operational modes.

From T-S diagrams it is worth to note one important parameter – turbine *flow coefficient* (or for the case, if normalization to the flow cross-sectional area is not needed, than *normalized flow* coefficient). The product $m \cdot K^{0.5}/P$ (*normalized flow*), where $m$ is mass flow, $K$ inlet temperature and $P$ is inlet pressure, has to be constant at all operation modes for the turbines with fixed nozzles. For the present *state-of-the-art*, the turbines with fixed nozzles are applied; however for the future, more advantageous would be to use turbines with variable nozzles, because it makes no limitations on the mass flows and more optimal usage of refrigerator at different operation modes is possible. For that reasons, the T-S diagrams in Figures 7 and 8 show the case of application of turbine with variable nozzles, and could be used as guidance for the specification of range for turbine flow coefficient. It is also possible to

**Table 6**. Heat exchanger parameter for 4-80 K box of 60+ kW refrigerator (Mix Mode – Maximum Capacity).

| Heat exchanger [a] | HX5 | HX6 | HX7 | HX8 | HX9 | HX10 | HX11 | HX12 | HX13 |
|---|---|---|---|---|---|---|---|---|---|
| Heat transfer, kW | 795 | 176 | 358 | 129 | 56.6 | 70.2 | 34.0 | 31.5 | 32.5 |
| $\Delta T_{hot}/\Delta T_{long}$ [b] | 4.3%, | 36.4% | 3.8% | 28.7% | 6.2% | 19.8% | 9.5% | 29.4% | 24.6% |
| Capacity ratio | 0.92 | 0.683 | 0.895 | 0.738 | 0.754 | 0.830 | 0.744 | 0.793 | 0.513 |
| Effectiveness | 96.2% | 95.5% | 96.7% | 97.5% | 95.6% | 97.2% | 93.6% | 87.1% | 90.5% |
| Minimal NTU | 13.9 | 6.45 | 13.4 | 9.3 | 7.5 | 11.3 | 6.1 | 4.2 | 3.5 |
| Minimal UA, W/K | $33.9\cdot10^4$ | $10.7\cdot10^4$ | $32.7\cdot10^4$ | $16.8\cdot10^4$ | $10.9\cdot10^4$ | $13.9\cdot10^4$ | $9.29\cdot10^4$ | $5.39\cdot10^4$ | $6.99\cdot10^4$ |
| $\Delta T_{log}$, K | 2.345 | 1.644 | 1.095 | 0.766 | 0.517 | 0.494 | 0.373 | 0.416 | 0.509 |
| Core dims. (W·H·L)[c], m | 2.2·0.65·5.3 | 2.2·0.37·5.3 | 2.2·0.55·5.3 | 2.2·0.37·5.3 | 0.9·0.9·3.0 | 0.9·0.9·3.0 | 0.9·0.9·3.0 | 0.7·0.7·3.0 | 0.9·0.9·3.0 |
| Approx. heat transfer area (HP/LP), m$^2$ | $2.71\cdot10^3$ $2.71\cdot10^3$ | $1.52\cdot10^3$ $1.52\cdot10^3$ | $2.27\cdot10^3$ $2.27\cdot10^3$ | $1.52\cdot10^2$ $1.52\cdot10^2$ | 870 870 | 870 870 | 870 870 | 530 530 | 580 1160 |
| Heat transfer coef. [d], W/m$^2$K | 180-230 | 190-220 | 170-220 | 120-180 | 100-190 | 80-130 | 90-130 | 80-120 | 70-110 |
| Estimated pressure losses [d] (HP/LP), mbar | 5-12 60-90 | 2-5 50-160 | 2-6 20-60 | 1-3 20-70 | 0.5-2 2-10 | 0.1-0.4 2-10 | 0.1-0.6 2-6 | 0.1-0.7 2-8 | 0.2-0.7 1-4 |

[a] The temperature, pressure and flow rates are presented in Figure 7.
[b] Ratio of temperature difference between warm and cold flow at the warm side of HX to the temperature difference of the cold flow at inlet and outlet of HX.
[c] Except HX13, all HX with single layer are used, i.e. layer pattern is e.g. HP-LP-HP-LP. If two forward flows are applied, e.g. HX7 and HX11, their surface areas are added, and the same is valid for LP and MP flows (HX5-8)
[d] Heat transfer coefficient and pressure losses strongly depend on which correlation for the calculations is used. The following correlations were applied: Grigorjev and Epifanova [27, 28], Cryogenmash [27], Kays and London [29], Wieting [30-32], Manglik and Bergles [32], Joshi and Webb [33], Maiti and Sarangi [31].

establish other T-S diagrams, which are optimized for operation at other ranges, or for application of turbines with fixed nozzles.

Pure liquefier:
It is possible to liquefy up to 600 g/s LHe into storage vessel. All turbines should operate at maximum design values. During liquefier modes, the refrigerator will operate at higher inlet temperatures and maximal pressure at the turbine inlets with the goal to maximize turbine output power. The turbines with fixed nozzles are selected according to this mode.

Pure refrigerator:
It is possible to obtain refrigeration of ca. 50 kW; however it is worth to remember that T-S diagram for pure refrigerator is not a good performance indicator due to the following reasons:
  a)  Turbine flows are very small, near on the lowest limit. In many cases it is possible to switch off one or two turbines, e.g. T1 and T3 (typically high temperature turbines are switched off).
  b)  Operational parameters, e.g. temperature and pressure, depends on the parasitic heat loads (which is in many cases difficult to precisely predict) and some smaller (but in many cases present) amount of liquefaction load, e.g. shown as 10 g/s in Figure 7. It is also possible to reduce the supply pressure to obtain larger volumetric flow for stable operation of turbines.
  c)  If heat exchangers are chosen with large NTU number, it is possible to reduce $LN_2$ consumption as well as to reduce the helium supply flow to the cold box.
     Refrigerator operating with the fixed nozzle turbines typically has low temperature at turbine inlets, lower HP supply (please note that for the consistency reasons, supply pressure is fixed to ca. 17 bar(a) for all operational modes, which implies that at this mode the turbines are working at the lower limit, see Table 7) and maximum volumetric displacement capacity for low pressure screw compressors. In order to have refrigerator with high thermodynamic efficiency, it is necessary to have high performance HX, e.g. 95-98% range, so HX are designed according to this mode.

50/50 Mix Mode (Design Maximum):
Maximum refrigerator capacity is achieved at this mode. With some optimization, i.e. operating the turbines at maximum power as well as choosing HX with large NTU, it is possible to increase total refrigerator capacity up to 70 kW at 4 K. This mode is further used for the comparison of "state-of-the-art", advanced and novel concepts.
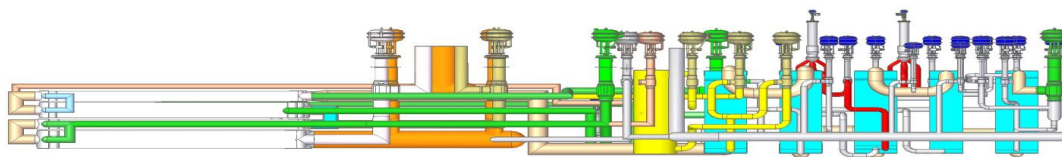
**Figure 9.** Positioning of HX inside the 4-80 K box.

*5.2.4. Heat exchangers.* The crucial parts of any cold box are HX and special attention has to be paid to the HX design for all operation modes. Table 6 shows some HX parameters calculated for the maximum capacity mode and internal design of HX, e.g. fin height and pitch, wall thickness, etc., is kept the same for all HX (see chapter 4.2 for the details of HX internal design). For the practical applications the following NTU should be chosen: HX5-16, HX6-10, HX7-15, HX8-12, HX9-10, HX10-15, HX11-10, HX12-7, HX13-9 and total NTU=109. In order to avoid maldistribution of helium flow in parallel channels, the aspect ratio is always more than factor 3 [6]. Total heat transfer surface area related to the HP flow is ca. 11700 m$^2$.

One of the challenges is the design of the last HX (HX13). Though for the maximum capacity mode the heat transfer area is sufficient, for the pure refrigerator it could be necessary to increase the heat transfer area, e.g. by increasing the total dimensions or making more compact internal geometry.

The next challenge is the pressure drop for LP stream in HX5 till HX8. It would be possible to decrease it by increasing the free cross-sectional area, however free space limitation in the horizontal part of the box, see Figure 5 and 9 allows only moderate HX volume increase. Space limitation due to diameter of this cylindrical part of vacuum vessel is 3760 m (with flanges 3975 mm) and it could be possible to increase it up to ca. 4200 mm, if transportation of this vessel on roads could be possible. It is also possible to perform optimization of HX surface area, e.g. by combining HX5 and 6 as well as HX6 and 7 in single cores, so only two HX blocks in the left side, see Figure 9, are present (this option is typical for HX design of present large refrigerators). In this case the length of HX blocks could be increased up to maximum and additionally it is possible to optimize the internal geometry, e.g. by increase the cross-sectional area for LP (and MP) streams and decreasing for HP one. Another option could be to relocate some of HX to another box. Three possible solutions could be possible:

a) To relocate HX5 to the high temperature box, see chapter 4 for details on 80-300 K box design. This solution is similar to the one applied on new 18 kW refrigerator at JLab [6].
b) It is possible to relocate HX5 and 6 to additional vertical cylindrical part of the present box, see Figure 5. This vertical part could be easily located beneath the middle crosspiece of present box. It could be also possible to slightly modify the crosspiece to include the side flange and to connect this cylindrical part horizontally at the ground level.
c) To design separate box, similar to the CBA/ISABELLE/RHIC refrigerator.

It is worth to note that in many refrigerators, the cross-sectional area of LP return line is decreased in order to have higher pressure losses and increased heat transfer and as consequence smaller HX dimensions. However, decreasing the pressure losses and lowering the 4K temperature level will have other advantages, e.g. high efficiency, low temperature for some specific experiments, or larger ΔT in subcooling HX for supercritical helium, etc.

*5.2.5. Turbines.* Table 7 summarizes some turbine operational parameters. As an encouraging results it worth to mention that T3 and T4 turbines could be taken as standard ones, produced by *Air Liquide* or *Linde Kryotechnik*, while for T1 and T2 some modifications or developments could be needed, see [34-44, 49] for the present state and turbine development history at *Air Liquide* and *Linde Kryotechnik*.

Air Liquide developed high power turbines [34, 36, 30, 43], which are able to cover the cooling power for T1 and T2 one. However, the required flow coefficients are large, e.g. 0.251-0.426 and 0.306-0.442 for T1 and T2 respectively, which seems to be beyond the values applied for turbines at Las Raffan refrigerator. Unfortunately due to not sufficient information given in the opened literature, it is not possible to give precise answer on the used flow coefficient for the Las Raffan refrigerator, but just very rough estimations based on typical flow rates and temperatures show that required normalized flow is by factor 4-5 larger for this 60+ W plant than for the Las Raffan one. Similar rough estimations of normalized flow for high power prototype turbines (ca. 120 kW) tested at *Air Liquide* test benches again give approximately the same factor [36, 39]. At the present, 150 kW class turbines are developed for Turbo-Brayton cooling at 77 K and most probably it is possible to apply them with

some minor modifications [49]. Therefore, R&D activities for T1 and T2 turbine, e.g. dimensional increase of turbine impeller and diffusor, could be required in order to cope with higher mass flows. It is worth to mention that application of several turbines connected in series, e.g. by *Linde Kryotechnik*, seems to be quite challenging due maximal power limitation of 50 kW per single turbine.

**Table 7**. Some turbine parameters for different operational modes.

| Turbine [a] | Maximum capacity | | | | Part load (25%) | | | | 100% Liquefier | | | | 100% Refrigerator | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | T1 | T2 | T3 | T4 | T1 | T2 | T3 | T4 | T1 | T2 | T3 | T4 | T1 | T2 | T3 | T4 |
| Inlet Temp. [K] | 49.82 | 26.06 | 16.02 | 8.651 | 36.03 | 17.01 | 11.29 | 7.959 | 58.91 | 29.67 | 20.48 | 11.24 | 39.58 | 20.02 | 13.28 | 7.709 |
| Outlet Temp. [K] | 39.21 | 18.76 | 10.31 | 6.180 | 37.95 | 12.25 | 9.15 | 6.545 | 46.39 | 21.40 | 13.44 | 7.22 | 35.45 | 16.37 | 9.47 | 6.002 |
| Inlet Press. [bar] | 17.5 | 7.56 | 17.1 | 17.0 | 7.75 | 3.20 | 7.71 | 7.70 | 17.5 | 7.56 | 17.1 | 17.0 | 17.5 | 7.56 | 17.1 | 17.0 |
| Outlet Press. [bar] | 7.56 | 2.40 | 4.10 | 4.10 | 3.20 | 1.04 | 4.10 | 4.10 | 7.56 | 2.40 | 4.10 | 4.10 | 7.56 | 2.40 | 4.10 | 4.10 |
| Mass flow [kg/s] | 1.9 | 1.9 | 1.305 | 1.795 | 0.810 | 0.810 | 0.417 | 0.573 | 2.40 | 2.40 | 1.075 | 1.526 | 2.1 | 2.1 | 1.134 | 1.766 |
| $\Delta h$ [b], [kJ/kg] | 57.14 | 37.42 | 24.66 | 9.440 | 42.37 | 23.70 | 8.37 | 4.09 | 67.43 | 42.85 | 33.68 | 14.23 | 22.2 | 17.0 | 11.7 | 7.12 |
| $\Delta h_{is}$ [b], [kJ/kg] | 76.12 | 49.89 | 32.85 | 12.58 | 56.46 | 31.41 | 11.13 | 5.45 | 89.83 | 57.08 | 44.91 | 18.96 | 60.5 | 37.6 | 24.9 | 11.06 |
| Efficiency [c], [%] | 75.1 | 75.0 | 75.1 | 75.0 | 75.0 | 75.4 | 75.2 | 75.1 | 75.1 | 75.0 | 75.0 | 75.0 | 36.7 | 45.2 | 47.1 | 64.3 |
| Power, [kW] | 108.6 | 71.1 | 32.2 | 16.9 | 34.3 | 19.2 | 3.49 | 2.35 | 161.8 | 102.8 | 36.2 | 18.9 | 46.7 | 35.7 | 13.1 | 12.6 |
| Jet velocity [d], [m/s] | 390.2 | 315.9 | 256.3 | 158.6 | 336.0 | 250.6 | 149 | 104.4 | 423.8 | 337.9 | 299.7 | 194.7 | 347 | 274 | 223 | 148 |
| Sound velocity [e] [m/s] | 436 | 310 | 263 | 260 | 363 | 245 | 202 | 173 | 471 | 330 | 292 | 241 | 392 | 272 | 247 | 276 |
| Wheel diameter [f], [m] | 0.094 | 0.094 | 0.050 | 0.050 | 0.094 | 0.094 | 0.050 | 0.050 | 0.094 | 0.094 | 0.050 | 0.050 | 0.094 | 0.094 | 0.050 | 0.050 |
| Tip velocity [g] [m/s] | 273 | 221 | 179 | 111 | 235 | 175 | 104 | 73 | 296 | 235 | 209 | 136 | 243 | 192 | 156 | 104 |
| Revolution [Hz] | 925 | 749 | 1142 | 707 | 797 | 594 | 465 | 465 | 1005 | 801 | 1336 | 868 | 824 | 650 | 996 | 663 |
| Norm. speed [Hz/K$^{0.5}$] | 131 | 146 | 285 | 240 | 132 | 144 | 197 | 165 | 131 | 147 | 295 | 258 | 131 | 145 | 273 | 238 |
| Nor. flow [kg/sK$^{0.5}$/bar] | 0.766 | 1.283 | 0.305 | 0.311 | 0.627 | 1.044 | 0.181 | 0.209 | 1.05 | 1.729 | 0.284 | 0.300 | 0.755 | 1.242 | 0.241 | 0.288 |
| Flow coefficient [h] | 0.254 | 0.370 | 0.153 | 0.155 | 0.201 | 0.299 | 0.118 | 0.119 | 0.348 | 0.499 | 0.144 | 0.147 | 0.251 | 0.356 | 0.120 | 0.147 |
| | 0.043 | 0.063 | 0.049 | 0.049 | 0.034 | 0.051 | 0.038 | 0.038 | 0.059 | 0.085 | 0.046 | 0.047 | 0.042 | 0.061 | 0.038 | 0.047 |
| Blade Mach  number | 0.626 | 0.712 | 0.680 | 0.426 | 0.647 | 0.713 | 0.517 | 0.420 | 0.630 | 0.715 | 0.718 | 0.565 | 0.619 | 0.703 | 0.631 | 0.376 |

[a] The temperature, pressure and flow rates are also presented in Figures 7 and 8.
[b] Enthalpy difference between inlet and outlet for real and isentropic cases.
[c] Turbine efficiency was limited to maximal value of ca. 75%.
[d] Jet (spouting) velocity, $c_{jet}=(2\cdot\Delta h)^{0.5}$.
[e] Sound velocities at the inlet were taken from HePak values, because these values calculated from ideal gas law had large errors for temperatures below 20K.
[f] Wheel diameter of 94 mm and height 4 mm for T1 and T2 is taken from the prototype turbine developed by Air Liquide [36, 39] and also used at Ras Laffan plant, while for T3 and T4, wheel diameter and height were 50 mm and 4 mm respectively in order to choose maximal turbine dimension produced by Linde Kryotechnik [40].
[g] tip velocity for all turbines was calculated assuming entrance velocity ratio 0.7 (entrance velocity ratio is ratio of tip velocity to jet/spouting one).
[h] There are different methods to define the flow coefficient: i) $m\cdot(R_{He}\cdot T_{inlet}/\gamma)^{0.5}/(A\cdot P_{inlet})$ and ii) $m/(\rho_{inlet}\cdot A\cdot(2\cdot\Delta h)^{0.5})$, where $m$ is mass flow, $R_{He}$ is helium specific gas constant, $A$ is cross-sectional area, $T_{inlet}$, $P_{inlet}$ and $\rho_{inlet}$ are inlet temperature, pressure and density, respectively. In order to avoid correction factors due to helium compressibility in the first formula, the second one was used. As a cross-sectional area, the formula $\pi\cdot D_{out}\cdot h_{out}$ was used, where $D_{out}$ is impeller diameter at the flow inlet, and $h_{out}$ is blade height at inlet of impeller (or at diffusor outlet). It is worth to mention that in some cases, other formula for the cross-sectional area, i.e. $\pi\cdot D_{out}^{2}/4$, is used in the opened literature [35]. Two values presented in the table are calculated for the cross-sectional areas $\pi\cdot D_{out}\cdot h_{out}$ (upper) and $\pi\cdot D_{out}^{2}/4$ (lower), respectively.

Turbines T3 and T4 have relative low power, up to 36 kW and normalized flow is in the range of 0.178-0.312. For these two turbines it is possible to use standard TED45 (old naming TGL45 according to Linde notation), e.g. turbines 3 from HERA or LHD refrigerators have similar operation ranges. Air Liquide has also similar turbines at several plants, e.g. second turbine for new 18 kW refrigerator for CERN, or second one for the Ras Laffan plant.

And last but not the least; it is worth to mention that for the future refrigerators, it would be very advantageous to have turbines with variable flow nozzles. First application of turbines with variable nozzle was at Isabelle/RHIC refrigerator as well as CHL at Fermilab, and other developments were done at Japan [45] and USA [46].

*5.2.6. Valves, piping and 20K absorber.* Figure 6 shows estimated dimensions of valves and piping. Though the valves up to DN100 are standard solution for several firms, e.g. WEKA, Velan and Flowserve, installation of the larger valves requires special engineering solutions. For application at helium temperatures, WEKA and Velan could produce up to DN300 (or in some case DN400), however due to very limited number of such valves produced so far, extensive testing at LN$_2$ (or LHe) temperatures would be necessary. For temperatures above 80 K, it is possible to apply valves from other manufacturing firms which have experience on valve application for air separation or natural gas

liquefaction plants; however large valve sizes, e.g. DN400-600, or leak tightness over the seat are still challenging issue for many valves.

It worth also to mention that due to scaling-up, it could be necessary to install small parallel control valves to the larger ones. This could facilitate flow controlling at small flow rates, e.g. during start-up of refrigerator, see e.g. Figure 6, where V22 (DN100) is installed in parallel to the larger valve V21 (DN250). It is possible to install the 20 K absorber within the box. It depends on type of absorbing material used, but it seems to be possible to use the vessel of 3 m height and 0.9 m diameter. If it is necessary, the diameter and length of 20 K absorber could be increased.

*5.2.7. Vacuum and LHe storage vessels.* The vacuum vessel consists of three parts: i) right side, where HX9-13 and 20 K absorver are located, ii) left side, where HX5-8 are mounted and ii) middle crosspiece, where two parts with HX are connected, see Figures 5 and 9. It is possible to dismantle the vacuum vessel and radiation shield for the left side of the box in order to have access to HX5-8. It is possible to install vertical or horizontal extension parts, which could be connected to the crosspiece, if some additional equipment is needed, or relocation of some HX is required.

The critical design parameter is the diameter of cold box. This value is different, e.g. 3.8 m (upper and lower temperatures boxes) for LHD, 4.2 m for Ras Laffan, 3.5 m for Air Liquide 12 kW LEP refrigerator, 4 m (vertical part) and 3.5 m (horizontal part) for Air Liquide 18 kW LHC one, and depends on transportation possibilities for final installation on-site. So, the final design of vessel diameter could be different, and for the present design, the 3.960 m (flange outside diameter) is arbitrary chosen.

The number and total volume of LHe storage vessels of 80-140 $m^3$ volumes could be chosen according to the actual customer needs. For the case if customer required supercritical helium at ca. 4 bar, 5 K, an optional sub-cooling HX could be installed inside the LHe storage vessel.

## 6. Refrigerator with high performance turbines and heat exchangers (advanced concept)

In this chapter, the influence of high performing turbines and HX on thermodynamic efficiency is roughly estimated. In relative short term perspective, it is possible to apply high performing HX with efficiencies of ca. 98% and turbines with 82-85% ones. This is a realistic assumptions for the near future R&D activities, though it is worth to mentioned that HX with 98% efficiency is current state-of-the-art (however they are not always applied in the refrigerator technology), and for turbines, some R&D activities are still needed.

For the simplicity of the estimations and discussion, the following assumptions are further taken: i) only Mix (50/50) mode is considered, ii) pressure losses in HP and MP lines are neglected (though for LP line, the losses are considered) and supply pressure to the cold boxes 19 bar(a), iii) 4 bar supply pressure to the LHe vessel (or customer), iv) for the last expansion from 4 bar(a) to 1.1 bar(a), the JT-valve is taken and neglecting some possible efficiency increase in case of turbine application instead of JT-valve, v) screw compressor set is used with intermediate pressure of 6 bar(a). As detailed design of single components is not given, the calculations with enthalpies are preferred over detailed exergy analysis of single components. Figure 10 shows the T-S diagram of refrigerator with high efficiency HX and turbines. For the estimations of overall thermodynamic efficiency, the 35% efficiency of $LN_2$ cooling circuit is assumed [5], and primary energy related to the $LN_2$ liquefaction is 770·844/0.35= 1.86 MW, where 770 is energy required for the liquefaction of 1 g/s, and 844 g/s is $LN_2$ flow. The ideal and real compressor work for the HP, MP and LP compressor stages are 3.78, 1.03, 3.34, and 6.38, 1.65, 6.61 MW respectively, which makes thermodynamic efficiency of the compressor sets as 55.7%.

For the rough estimations, it is assumed that 1 g/s liquefaction rate is equivalent of ca. 100 W of refrigeration power, and inversed coefficient of performance 68.7 W/W is used. The refrigeration and total electrical powers are $(55.4 \cdot 10^3 + 300 \cdot 100) = 85.4$ kW and $(1.86+6.38+1.65+6.61) \cdot 10^3$ kW=16.5 MW, respectively and estimated efficiency is ca. 35.6 %.

The manufacturing of HX is challenging, particular for the high temperature (300-80 K) one. The NTU>60 and e≈99% requires special attention to HX design in order to limit parasitic longitudinal heat conduction and to have the high efficient heat transfer. Due to reduced logarithmic temperature difference ($\Delta T_{log}$ is in the range of 0.12 till 0.29 K) and increased energy transfer at low temperatures, the NTU and UA are increased by factor 2 till 3 for HX10-13 in comparison to the data presented in Table 6. With the present available free space inside the box and/or optimization of HX internal geometry it is possible to install larger and efficient HX10-13 in order to obtain high performance operation at the refrigerator cold end.
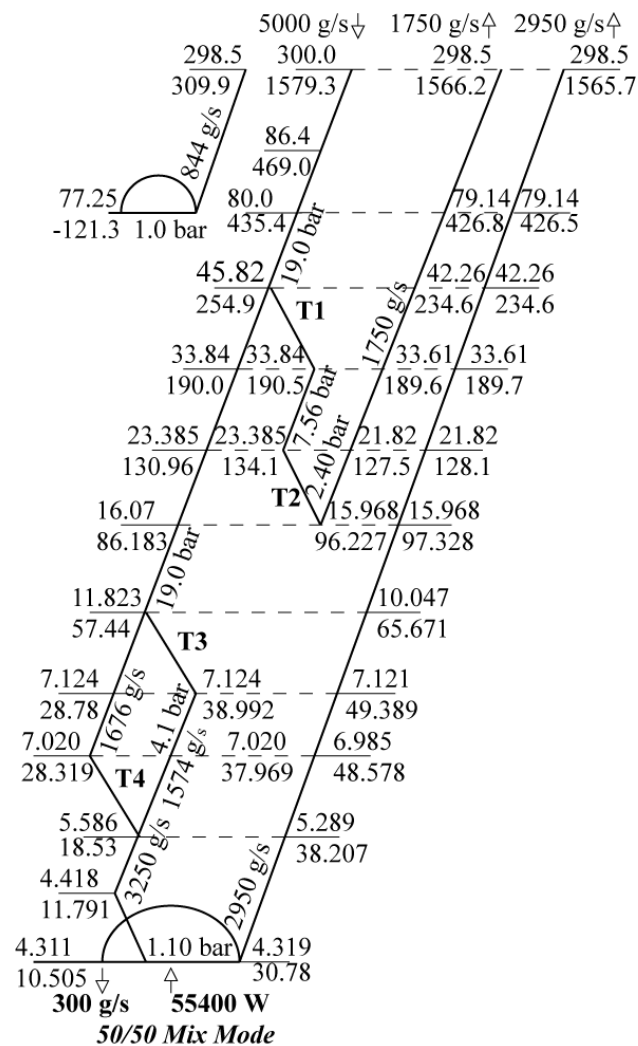
**Figure 10.** T-S diagram for 50/50 Mix Mode assuming high efficiencies of HX and turbines.

So, for the near term perspectives, it is worth to invest into R&D of high efficiency turbines and to install high performance HX in order to improve overall thermodynamic efficiency of cryogenic plants.

It is also possible to apply 6-stage turbo-compressors, e.g. 2 stages for LP stream with compression ratios 1.512 and 4 ones for HP stream with ratios 1.678, see chapter 3.3. The total electrical power is $(2 \cdot 1.05 + 4 \cdot 2.07) \cdot 10^3$ kW= 10.38 MW and estimated overall efficiency is ca. 56.5 %. So, due to low compression ratio per stage, the polytropic and isothermal efficiencies are high and results of advanced and novel concepts (see also chapter 7) could be comparable.

## 7. Novel concept

In this section, several novel process schemes of 4 K box operation are considered. The goal is to design refrigerator with overall thermodynamic efficiency of more than 45%. Figure 11 shows several proposed process schemes of 4-80 K box. The high efficient HX (up to 98.5%) and turbines (82-85%) were used as well as 43 bar(a) supply pressure. To limit the scope of present paper, only four cases are considered, i.e. pure serial connection of turbines, see *Case A* in Figure 11, parallel-serial turbine connection, see *Case B*, parallel-serial turbine connection either with supply line or return line, see *Cases C* and *D* respectively. The key idea is to precool HP stream with turbines, which is similar to cooling with Kundig's arrangement or proposal of Quack and Ziegler [47]. Table 8 summarizes some refrigerator parameters.

*Case A*: HP helium stream is serially expanded in 4 turbines. The advantage is low number of HX and turbines.
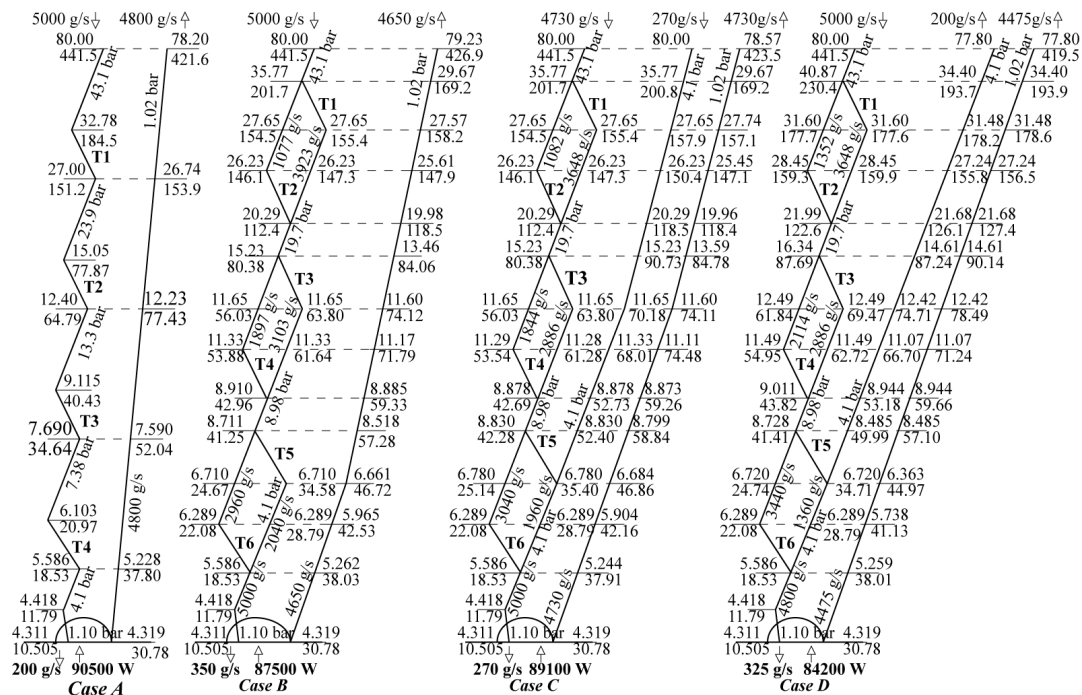
**Figure 11.** TS diagrams for some proposed process schemes of novel 60+ kW refrigerator.

*Case B*: HP helium stream is expanded in 6 turbines. Turbines are parallel arranged, e.g. T5/T6, T3/T4, T1/T2, which are then connected in series and intermediate pressures are 19.7 and 8.98 bar(a). It is worth to note that instead of LN2 pre-cooling it is possible to install two additional turbines with intlet/outlet temperatures 114/96 K and 95/80 K as well as power 24 and 420 kW, respectively. In this case the HP has to be increased to 73.2 bar, which is beyond the limit for many refrigerator components.

*Case C*: it is variation of Case B with additional supply flow at 4.1 bar and 270 g/s for liquefaction. In some practical cases, e.g. helium recovery from natural gas, it is advantageous to have separate line for helium flow from other facilities, which is then liquefied and LHe supplied to the customer. Advantage is to separate clean refrigerator flow from potentially "unclean" low pressure flow to be liquefied. In some cases there is an additional advantage to use already available flow at low pressure (in our case 4.1 bar) and further compressions to high pressures is not required.

*Case D*: it is variation of *Case B* but with some low pressure return flow, e.g. 0.2 kg/s and 4.1 bar after expansion in turbine 5. This scheme applies novel cooling by turbine expansion as well as by classical cold return flow. The return flow could be taken at different temperature level and turbine outlets (or installation of additional turbines for some required outlet temperature is also possible). This corresponds to the practical case, when some additional cooling of the radiation shield, e.g. at 5-10 K or 40-80 K temperature level, is needed. For example, for the 40-80 K shield cooling it would be possible to take some flow from outlet of T1, or to install one (or two) additional turbine, which expands 80 K flow to the required shield inlet temperature.

As one can note, the estimated efficiencies are within 4% range for the proposed process schemes. It is worth to note that this value is low in comparison to the typical uncertainties related to R&D on high pressure components, e.g. up to 50 bar, as was as ones on manufacturing tolerances for high efficient turbines and HX. For this reasons, it seems to be more reasonable to manufacture novel 60+ kW refrigerator based on the practical considerations, e.g. lower number of turbines and HX, reasonable assumptions on HX efficiencies, etc.

It is worth to note that it is possible to further increase the thermodynamic efficiency, e.g.: a) to reduces the LN$_2$ consumption (LN$_2$ cycle has only 35% thermodynamic efficiency), b) to increase the polytropic efficiency of turbo compressor stages up to 92%, (for estimations, 88% is assumed and 90% is achieved on several prototypes to be applied for future helium cooled nuclear reactors) and c) to

**Table 8.** Short summary of key parameters of novel refrigerator.

| | Case A | Case B | Case C | Case D |
|---|---|---|---|---|
| Turbo-compr. power [a] | 14.94MW | 14.94MW | 14.42MW | 14,72MW |
| LN$_2$ for precooling [b] | 495 g/s | 1000 g/s | 1450 g/s | 857 g/s |
| T1 | 167 kW | 182 kW | 169 kW | 193 kW |
| T2 | 65.3 kW | 36.3 kW | 36.5 kW | 49.6 kW |
| T3 | 28.9 kW | 51.5 kW | 47.8kW | 52.6 kW |
| T4 | 12.2 kW | 20.7 kW | 20.0 kW | 23.5 kW |
| T5 | - | 13.6 kW | 9.77 kW | 9.11 kW |
| T6 | - | 10.5 kW | 10.8 kW | 12.2 kW |
| Liquefaction | 200 g/s | 350 g/s | 270 g/s | 325 g/s |
| Refrig. power | 90.5 kW | 87.5 kW | 89.1 kW | 84.2 kW |
| Equiv. power at 4.3K [c] | 110.5 kW | 122.5 kW | 116,1 kW | 116,7 kW |
| Estim. therm. effic. [d] | 47.3 % | 49.1 % | 45.3 % | 48.2 % |

[a] seven turbo compressor stages are considered with isothermal efficiency of 77-79%, see chapter 3.3 for more details on main parameters of turbo compressors.

[b] 35% thermodynamic efficiency and 770W/g/s primary energy are taken for LN$_2$ circuit.

[c] for estimations, 1 g/s liquefaction is taken as 100W refrigeration power.

[d] estimated thermodynamic efficiency doesn't include any parasitic or other heat loads

increase the number of turbo compressor stages, i.e. with lower compression ratio, the isentropic efficiency reaches the polytropic one.

## 8. Conclusion

In the present paper the basic design of 60 kW+ refrigerator is given. Except some R&D activities on high power turbines needed, most of the components could be designed and manufactured using present *state-of-the-art* technology of helium refrigerators. With high efficiency turbines and HX it is possible to reach the refrigerator thermodynamic efficiency of ca. 35%. Several novel process schemes for 4 K box are discussed. Application of turbo-compressors, turbine cooling of supply flow, as well as high efficient HX and turbines allows increasing of thermodynamic efficiency up to ca. 50%.

## 9. References

[1]　Ganni V at al. "Compressor system for the 12 GeV upgrade at JLab" *Proc. of 23rd Int. Cryo. Eng. Conf.* **23** (2010) p859

[2]　Knudsen P at al. "Commissioning of helium compression system for the 12 GeV refrigerator" *Adv. Cry. Eng. Conf.* **59** (2013) p962-967

[3]　Knudsen P at al. "Performance testing of Jefferson Lab 12 GeV helium screw compressors", *9th Int. Conf. on compressors and their systems, Material Science and Engineering* **90** (2015) 012072

[4]　Ganni V at al. "Screw compressor characteristics for helium refrigeration systems", *Adv. Cry. Eng. Conf.* **53** (2007) p309

[5]　Cryogenics seminars at JLab, January 25 to April 26, (2011), Cryogenics at JLab, Cycle design/Carnot Analysis, System Optimization/Ganni Cycle, 2K system/LN2/Typical Components, Instrumentation/Controls/Design Overview, Small 2K system design, Helium purification, 12 GeV Cryo System Planning

[6]　Knudsen P at al. "Commissioning and operational results of helium refrigeration system at JLab for the 12 GeV upgrade" *Adv. Cry. Eng. Conf.* **61**, *Material Science and Engineering* **101** (2015) 012125

[7]　Weisbrodt I A, "Summary report on technical experiences from high-temperature helium turbomachinery testing in Germany", *Report XA9642789*

[8]　McDonald C F at al. "Helium turbomachnine design for GT-MHR power plant", *GA-A21720*, July (1994)

[9]　Takada S at al. "Research and development programs on helium gas compressor and magnetic bearing suspended rotor for the Gas Turbine High Temperature Reactor (GTHTR300)", *Int. Conf. on global environment and advanced nuclear power plants* (2003) paper 10696

[10]　LaBar M P at al. "The Gas-turbine modular helium reactor", *Nuclear Energy Materials and*

*Reactors*, Vol. II, Encyclopedia of Life Support Systems (2010) p263-272

[11] Rodwell Ed, "Helium Gas Turbine Reactor Technical Challenges", *Technical Progress Report, TP-114690* (2000)

[12] No H C at al. "A review of helium gas turbine technology for high-temperature gas-cooled reactors", *Nuclear Engineering and Technology*, **39** (2007) p21-30

[13] Wu Z at al. "HTGR projects in China", *Nuclear Engineering and Technology*, **39** (2007) p103-110

[14] Gessner R L, "Multistage, gas-bearing, helium compressor development", *Adv. Cry. Eng. Conf.* **12** (1967) p631-639

[15] Ergenc, Giger, Trepp, "Betriebscharakteristik einfacher Helium-Kaeltekreislaufe mit Kaltgasturbinen" *Kaeltetechnik* **16** (1964) 4 p.916-931

[16] "Operator's manual 1500 Watt Helium refrigerator for FERMI/LBL", March 1978, *Helix Process System*

[17] Byrns R A, "Large Helium Refrigerators and Liquefiers", *LBL-15167 Report*, January 1984

[18] Hubbel R, "A 1400 L/hr Helium Liquefaction Cycle" *71st Annual Meeting American Institute Chemical Engineer*, 1978

[19] Trepp, "A large Scale Helium Liquefier", *Adv. Cry. Eng. Conf.* **12** (1966) p645-651

[20] Powell R, "A unique Helium refrigeration system for the Large Coil Task at Oak Ridge National Laboratory" *Adv. Cry. Eng. Conf.* **27** (1981) p565-571

[21] Slac D and Nelson R "Cryogenic systems for the Mirror fusion test facility", *Adv. Cry. Eng. Conf.* **31** (1985) p583-594

[22] Schermer R I at al. "Design and operation of the 30 MJ superconducting magnetic storage system on the Bonneville power administration bus", *Adv. Cry. Eng. Conf.* **29** (1984) p123-132 , see also J. D. Rogers "Superconducting Magnetic Energy Storage Program" *LA-9208-PR report* (1982)

[23] Dean J, "Liquid helium refrigerator control", *Cryogenics* **25** (1985) p87-91

[24] Clausen J at al. "The Linde-turborefrigerator for MR-tomographs", *Adv. Cry. Eng. Conf.* **35** (1989) p949-955

[25] Knudsen P and Ganni V "Simplified helium refrigerator cycle analysis using the "Carnot step", *Adv. Cry. Eng. Conf.* **51** (2005) p1977-1986

[26] Ganni V and Knudsen P "Optimal design and operation of helium refrigeration systems using the Ganni cycle", *Adv. Cry. Eng. Conf.* **55** (2010) p1057-1071

[27] Grigoriev V A and Krokhin Yu I, "Heat and Mass Transfer Equipment of Cryogenic Heat and mass transfer apparatus of cryogenic engineering", *Energoizdat*, Moscow (1982) p312 (in Russian)

[28] Epifanova V I and Akselrod L S "Air separation by deep cooling, first band: Thermodynamic basics of air separation, systems and apparatus of air separation plants", (1973) Vol. I, p472, and second band: "Engineering and equipment", Ed., *Mashinostroiyenije Pub.* Moscow, (1973) Vol. II, p568 (in Russian)

[29] Kays W M and London A L "Compact Heat Exchanger", *McGraw-Hill*, New York (1984)

[30] Wieting A R "Empirical correlations for heat transfer and flow friction characteristics of rectangular offset-fin plate-fin heat exchangers", *ASME Journal of Heat Transfer* **97** (1975) p.488-490

[31] Alur S Experimental Studies on Plate Fin Heat Exchangers, *Ph.D. work,* Mechanical Engineering Department, National Institute of Technology, Rourkela (2012)

[32] Manglik R and Bergles A, "Heat Transfer and Pressure Drop Correlation for the Rectangular Offset strip Fin Compact Heat Exchanger" *Experimental Thermal and Fluid Science* **10** (1995) p.171-180

[33] Joshi H and Webb R, "Heat transfer and friction in the offset stripfin heat exchanger", *Int. J. of Heat and Mass Transfer* **30** (1987) p.69-84

[34] Marot G at al. "Recent developmments of Air Liquide cryogenic expanders" *Adv. Cry. Eng. Conf.* **45** (1999) p1493

[35] Gretegny D at al. "Efficiency improvement of small gas bearing turbines – impact on standard helium liquefier performance" *Adv. Cry. Eng. Conf.* **49** (2003) p272-278

[36] Marot G at al. "Development of a new Air Liquide cryogenic expander" *Adv. Cry. Eng. Conf.* **51** (2005) p1845-1851

[37]   Bischoff S at al. First operating results of a dynamic gas bearing turbine in an industrial hydrogen liquefier *Adv. Cry. Eng. Conf.* **55** (2009) p887-894

[38]   Ohlig K at al. Dynamic gas bearing turbine technology in hydrogen plants *Adv. Cry. Eng. Conf.* **57** (2011) p814-819

[39]   Delcayre F at al. "Recent developments on Air Liquide advanced technologies turbines" *Adv. Cry. Eng. Conf.* **57** (2011) p820-827

[40]   Eber N at al. "Gas bearing turbines with dynamic gas bearing and their application in helium refrigerators" *Cryogenics* **18** (1978) p585-588

[41]   Gass J at al. "Gas bearing cryogenic expansion turbines: application to a refrigeration plant delivering 10 kW at 25 K" *Proc. of 4th Int. Cryo. Eng. Conf.* **4** (1972) p74-76

[42]   Giger U at al. Combined helium refrigerator and liquefier with gas bearing turbines *Proc. of 5th Int. Cryo. Eng. Conf.* **5** (1974) p380-383

[43]   Gistau G at al. "High power industrial gas bearing cryogenic expansion turbines" *Proc. of 14th Int. Cryo. Eng. Conf.* **14** (1992) p87-90

[44]   Schoenfeld H at al. "Standard liquefier-test results with improved turbines" *Proc. of 20th Int. Cryo. Eng. Conf.* **20** (2004) p119-122

[45]   Kato T at al. "Development on a large helium turbo-expander with variable capacity", *Adv. Cry. Eng. Conf.* **37** (1991) p827-834

[46]   Agahi R R at al. "High performance turboexpanders", *Adv. Cry. Eng. Conf.* **41** (1995) p941-947

[47]   Ziegler B at al. "Helium refrigeration at 40 percent efficiency?" *Adv. Cry. Eng. Conf.* **37** (1991) p645-651

[48]   Kundig A "Method and apparatus for liquefying a low boiling gas" *US Paten 4.606.744*

[49]   Gondrand C at al. "ALAT lates magnetic bearings turbomachines commissioning", *this conference*