Design of cold compressor systems in terms of operational and economical aspects

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Abstract. In the past, cooling at 2 K used to be an exotic application in large scale cryogenics. The required sub-atmospheric helium bath was established with the help of one of the following two technical approaches – rough vacuum pumping at ambient temperature or turbo compression at cryogenic temperature – or a combination of both. The aforementioned approaches are still being applied, but the optimum distribution between warm and cold stages is not always obvious. In the last few years, 2 K cooling became a new state-of-the-art in the fields of experimental and applied physics. Standardisation of the machinery and its control significantly reduced commissioning time which has clearly been demonstrated during start-up of refrigeration plants such as Fermilab and DESY. Thus, the technological readiness of cold compressors has successfully been proven. This paper presents criteria for the optimisation of a cold compressor system under operational and economical aspects depending on the required 2 K cooling capacity.

1. Introduction
The need to increase magnetic fields and current densities in superconducting cables, the use of cavities and other applications in a widening field of science asks for cooling at lowest achievable temperature. Carnot’s rule, cavity losses and properties of superfluid Helium define an optimum temperature range between 1.6 K and 2.0 K [1]. The choice of potential cryogenic refrigeration cycles has been investigated [2]. Consistently, the proposed three cycles (warm compression, mixed compression, cold compression) have been examined under the aspect of a cost-to-performance optimization specifically for large systems typically installed at CERN [3]. With progress becoming standard, more and more research projects require 2 K refrigeration. This raises the question, which of the three proposed cycles is optimum choice within the wide range of needed cooling capacity (100 W to 10 kW). Herein, the term 2 K refrigeration shall cover above mentioned temperature range between 1.6 K and 2.0 K. All calculations presented herein are based on 1.9 K except data taken from [3].

2. The 2 K Refrigeration Cycle
For cryogenic refrigeration capacities below 10 K and above a few Watts, helium is used for the refrigerant. To decrease the temperature below 4.2 K, the helium is pumped down to below 31 mbar. This creates a large volumetric flow of deep cold gas that is recompressed to atmospheric pressure prior to joining the main 4.5 K refrigeration cycle. Figure 1 shows the three proposed 2 K refrigeration cycles schematically indicating the main differences between them.
Warm compression: The 4.5 K refrigerator works almost as a liquefier supplying sub-cooled liquid helium to the 2 K load. The exergy of the returning cold sub-atmospheric gas is only recovered to some extent for sub-cooling of the liquid helium supply. The rest is spoiled because of the difficulty of exchanging heat within the coldbox without causing extra pressure drop. The exergy is lost in an ambient heater before the helium gets recompressed in the warm vacuum pump (WVP) (screw compressor, dry claw pump etc.).

Cold compression: The helium returning from the sub-cooler is directly recompressed at cold state to atmospheric pressure. This requires 4 (max. 5) cold turbo compressors (CC) arranged in series. The 2 K return flow can join the low pressure LP line at a same temperature level (20 K to 30 K) supporting the cooling capacity of the 4.5 K refrigeration cycle.

Mixed compression: Compressing the 2 K flow in 2 to 3 cold compressors only to an intermediate, still sub-atmospheric pressure decreases the complexity of the control regime for the cold compressors. The pressure drop along the required extra heat exchanger channel is manageable. At ambient temperature, a small WVP does the final compression to atmospheric pressure.

3. Assumptions for the economical optimization
Applying a 2 K refrigeration system always requires close examination of boundary conditions and operation parameters. Covering all would go beyond the scope of this work. Based on experience with 2 K refrigeration systems and data from [3] the following assumptions have been made.

Valid for all three cycles: The state of helium at the interface to WVP, respectively 1st CC is 23 mbar and 3.4 K.

Warm compression cycle: Prior to compression, the helium is warmed in an ambient heater to 282 K (no electric heating). The warm compressors are either of lubricated screw type with efficiency according to figure 2 or, for smaller systems, dry claw type with an efficiency of 2/3 of screw type.

Cold compression: All cold compressors and related cryogenic equipment are mounted directly into the 4.5 K refrigerator. This has - at least for small to medium capacities - the advantage of saving...
cost by reducing the number of coldboxes. The adiabatic efficiency of the radial turbo compressors is 75%, and the heat leak into each stage is determined by the outer wheel diameter. The cold compression cycle has 4 CC stages compressing to 1.1 bar. The interstage pressures are set to 98 mbar, 320 mbar and 620 mbar.

Mixed compression: The pressure drop between discharge of the last CC and the suction of the WVP is 10% of the CC discharge pressure. The mixed compression has 2 stages, respectively 3 stages. The interstage pressures are same as for cold compression.

![Isothermal efficiency of warm compressors (WC) [3].](image)

### 4. Cost analysis

Each resulting cost value ($V_{ci}$) for a refrigeration capacity of $x$ kW at 1.9 K has been divided by the equivalent cost value for the refrigeration capacity of the identical $x$ kW at 4.5 K. This gives a factor that allows estimating the cost of a 2 K refrigeration system based on much better known numbers of reference 4.5 K refrigerators without disclosing any proprietary information. The reference cost value $V_{ci, ref}$ is defined in equation 1, the resulting cost value $V_{ci}$ in equation 2.

$$V_{ci, ref} = V_{ci, 4.5K} / V_{ci, 4.5K} \quad [-] \quad (1)$$

$$V_{ci} = V_{ci, 1.9K} / V_{ci, 4.5K} \quad [-] \quad (2)$$

with

- $V_{ci, 4.5K}$ [MEUR*] cost value for a 4.5 K refrigerator
- $V_{ci, 1.9K}$ [MEUR] cost value for a 1.9 K refrigerator at the same load as $V_{ci, 4.5K}$

* MEUR: million euros

#### 4.1. CAPEX (capital expenditure)

The main CAPEX cost drivers for 2 K refrigeration systems can be split into the following cost values: 4.5 K refrigerator (C1), warm vacuum pump (C2) and cold compressors (C3).

Using the same approach as in [3], C1 (equation 3) can be evaluated using Linde’s database of installed 4.5 K refrigerators and from $r$, the isothermal refrigeration capacity at 4.5 K exergetically equivalent to the capacity at the requested temperature (figure 3).

$$C1 = \frac{V_{ci, 4.5K}}{r} \quad [MEUR] \quad (3)$$

C2 (equation 4) for WVP considers only the machine with its related cost. It is adapted from [3] applying current prices for oil lubricated screw compressors and dry claw pumps adding extra for interfaces etc. Frequency converters are not considered.
C3 (equation 5) per single cold compressor includes the machine, its drive and the complex control regime. The motor power is included. This equation derives from Linde’s standard cold compressors, estimating prices for larger machines.

\[ C_1 = A \cdot r \cdot Q^{*1.9} \cdot n + B \quad \text{[MEUR (million euro)]} \quad (3) \]
\[ C_2 = D \cdot \left( \frac{Q^{*1.9}}{p_s} \right)^m + E \quad \text{[MEUR]} \quad (4) \]
\[ C_3 = F \cdot \left( \frac{m^{*1.9} \cdot T_i^{0.5}}{p_i} \right)^k \quad \text{[MEUR]} \quad (5) \]

with
- \( m^{*1.9} \) [kg/s]: 1.9 K massflow
- \( p_s \) [bar]: suction pressure of WVP
- \( p_i \) [mbar]: CC inlet pressure
- \( Q^{*1.9} \) [kW]: 1.9 K load
- \( r \) according to [3]
- \( T_i \) [K]: CC inlet temperature
- A, B, D, E, F, k, n, m: proprietary data

Figure 3. Specific equivalent 4.5 K load induced by a 1.8 K refrigeration unit [3].

The resulting CAPEX is given as the sum of C1, C2 and C3 (figure 4).
4.2. **OPEX (operational expenditure)**

Operating costs include electricity, cooling water, and man power during all operation modes and periods such as full load, part load and maintenance. Part load is very important not only for OPEX reasons but also for flexibility in operation. An efficient part load scheme also helps to match the installed 2 K refrigeration system to the final cooling requirements of the experiment. Both systems tend to be specified with sufficient massflow margin resulting in overdesigned cold compressors. The final operating point of the 2 K system equals part load in the operating field of the cold compressors.

In this first analysis only electricity cost at full load shall be considered. Water consumption could be directly derived from electricity in a second step. Part load needs further investigation.

Pricing for electricity varies widely with location and consumption and may differ significantly from values stated in [4]. The following values are used as an assumed average:

- $c_e$: specific cost of electricity = 0.15 EUR/kWh
- $t_y$: yearly time of operation = 6000 h
- $t_p$: payback time = 10 years

OPEX is treated same as CAPEX, i.e.: 4.5 K refrigerator (O1), warm vacuum pump (O2) and cold compressors (O3).

O1 (equation 6) uses the same equivalent load factor $r$ from equation (3) and an additional specific load factor $s$ of Watts needed for warm compression per Watt load at 4.5 K. It is an empirical function extracted from state-of-the-art installed systems. This factor $s$ is given in figure 5 for loads at 4.5 K, which equals $Q_{1.9}^* \cdot r$.

O2 (equation 7) is directly calculated from ideal isothermal compression power and the isothermal efficiency of machine (figure 2).

O3 for the cold compressor stages is negligible (typically 1% of O1, due to the much lower suction temperature at the cold compressors and the high efficiency of cold compression versus warm compression).

\[
O1 = (Q_{1.9}^* \cdot r \cdot s) \cdot t_y \cdot t_p \cdot c_e \quad \text{[MEUR]} \quad (6)
\]
\[ O_2 = \frac{P_T}{\eta_T} \cdot t_y \cdot t_p \cdot c_e \cdot 1e^6 \]  \[ \text{[MEUR]} \]  \[ (7) \]

with \( \eta_T \) isothermal efficiency

\( P_T \) ideal isothermal compression power

\( s \) specific compression power per load at 4.5 K proprietary data

**Figure 5.** Specific compression power per load at 4.5 K [W/W].

The resulting OPEX is given as the sum of O1, O2 and O3 (figure 6).

**Figure 6.** Specific OPEX for 1.9 K loads.

### 4.3. TOTEX (total expenditure)

The total cost of the 2 K refrigeration system is then given by the sum of CAPEX and OPEX (figure 7).
5. Discussion
Warm compression cycles, although they could simply be arranged by adding an ambient heat and a warm vacuum pump to a standard helium liquefier, should be avoided whenever possible because their efficiency is very poor. They should only be chosen for 2 K refrigeration loads below 200 W. In this range state-of-the-art cold compressors are of disadvantage because they become small and their static heat inleak gets dominant (see I in figure 8).

For 2 K refrigeration loads between 200 W to 500 W, 2 cold compressors in series with a WVP are best choice. Again stage 3 or 4 would be too small (see II in figure 8).

Above 500 W (see III in figure 8), 2, 3 and 4 stages seem to be competing with each other with advantages on side of 2 stage cycles. This differs from the results presented in [3]. An explanation could be found in the assumptions for CAPEX C2 taking only the WVP into account, whereas [3] installs one additional 2 K coldbox hosting the cold compressors. Both assumptions are legitimate: the one for C2 for small and medium size plants, and the one in [3] for large sizes. Further investigation for an optimized scope split vs. 2 K load is needed here.
6. Conclusion
Many assumptions and simplifications had to be made in order to derive the results. They shall not be understood as exact science but seen as a good tool for the right choice of the 2 K refrigeration cycle. In addition they give a first rough cost estimates in relation to state-of-the-art 4.5 K refrigerators. Further work might give answers to the sensitivity of the data on requested temperature of the 2 K refrigeration cycle (1.6 K and 2.0 K), the degree of exergy recovered in the sub-cooler and different part load scenarios, offering not only advantages but also carrying their own specific handicaps.

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