16

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SAFE AND EFFICIENT OPERATION OF MULTISTAGE COLD COMPRESSOR SYSTEMS

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ABSTRACT

Large refrigeration rates in the temperature range of super fluid helium can only be obtained with the help of centrifugal cold compressors. For the large 2 K systems, four compression stages are necessary to reach atmospheric pressure. Centrifugal cold compressors are quite sensitive to mass flow and suction temperature variations; but these have to be expected in a real system. The first step in the systems design is to find safe and efficient quasi-stationary modes of operation.

The system which is being proposed for the TESLA refrigerators relies on two features. The first is to allow the room temperature screw compressor, downstream of the cold compressors to work occasionally with a subatmospheric suction pressure. The second is to stabilise the suction temperature of the third stage of compression at about 10 K. With these features it is possible, that in all modes of operation all four compressor stages operate exactly at their design point.

INTRODUCTION

In the linear collider project TESLA electrons will be accelerated up to a collision energy of 500 GeV. TESLA will work with superconducting cavities operating at 1.3 GHz. This e^+e^- -collider will be built from nine-cell cavities with an acceleration gradient of about 25 MV/m. For a final energy of 500 GeV a total accelerator length of 32 km is required. The liquid helium for the linac will be made available by 16 refrigerators in 8 refrigerator stations. This means that each refrigerator supplies a length of 1.8 km of the linac. The refrigerator specification is mainly given by the cooling demands of the cavities and the quadrupoles. The superconducting cavities have to be cooled below 2 K [1]. The quadrupole magnets for focusing the beam are designed for a working temperature of 4.4 K. An 80 K cooling shield is proposed in order to reduce radiation losses. The provisional specifications for one refrigerator are given in Table 1.

Table 1: Heat loads of TESLA at the three temperature levels per refrigerator

	static load	total load
2 K	500 W	4.2 kW
4.4 K	2 kW	4.4 kW
80 K	6.5 kW	22 kW

Figure 1: TESLA refrigeration scheme

It is important to note that additionally to the static load, dynamic loads are induced by the 1.3 GHz excitation of the cavities. So the refrigerators will have to cope with strongly and rapidly changing loads. Nevertheless the refrigerators should have a very good efficiency, because the running costs of the collider are a major cost factor and they contribute strongly to the viability of the total project.

So the challenge is to invent a refrigerator scheme, which is thermodynamically efficient, but also flexible to cope with rapidly changing loads.

COLD COMPRESSORS

The temperature of 2 K corresponds to a vapour pressure of helium of 31 mbar. The large refrigeration capacity required for TESLA can only be obtained with a multi-stage cold compression system [2]. The total pressure ratio up to atmospheric pressure is about 36, so using a four stage system an average pressure ratio of 2.5 per stage is required. Actually the colder stages will have a somewhat larger pressure ratio.

The main difficulty of the task comes from the fact, that such turbocompressors have a very narrow safe operational band, quite different from the load changes, which one has to expect. The cold compressors are very sensitive to changes in the suction temperature, because the suction temperature influences strongly the necessary enthalpy head which has to be covered by the single compressor.

Figure 2: Required duties and safe operational regime of cold compressors

Figure 2 demonstrates the dilemma caused by the ranges of mass flow and suction temperature which have to be covered. Figure 2 shows even only half of the problem, because it does not yet describe the dynamics, i.e. the speed at which the changes may occur.

QUASI-STATIC SOLUTIONS

The task has to be solved in two steps:
First a solution has to be found for the quasi-static operation, i.e. for the required operational range shown in figure 2, where the speed of change is not yet taken into account.

Figure 3a

Figure 3b

Figure 3c

Figure 3d

Figure 3: Diverse approaches of controlling safe operation of cold compressors

In figure 3, a number of previous approaches are shown [3]. Figure 3a shows the starting system with the characteristic of figure 2. In figure 3b the suction temperature of the first stage of compression is stabilised by a thermal anchoring to a 4.4 K bath. By this method the disturbing influence on the required enthalpy head is removed, but

- the changes in the mass flow rate are not addressed
- in the heat exchanger thermodynamic losses due to heat transfer from 4.4 K to 3.5 K are produced
- and this is a very expensive heat exchanger causing additional pressure drop for the low pressure gas.

Figure 3c shows the proposal for a bypass valve from the vapour space of the 4.4 K bath to the suction side of the first compression stage. This way the fluctuations of the mass flow rate are reduced, to a certain degree also the suction temperature is stabilised. But still this operation is thermodynamically not acceptable over longer periods of operation.

Figure 3d shows a bypass valve over all four stages of compression. This may stabilise the suction mass flow, but it has a detrimental effect on the suction temperature and efficiency.

Our approach was led by the search for a solution with a minimum of loss producing features at least for the quasi-static operation. To cope with the mass flow variation, we intend to work with the speed control of the compressors only: If less flow comes the speed is reduced, if more flow comes the speed is increased. And if the suction temperature rises or falls we live with that pressure ratio, which the compressor is able to handle under such flow and speed conditions.

With this control method, the total pressure ratio cannot anymore be reached under all mass flow conditions, the exhaust pressure of the fourth stage is being reduced at smaller than design flow rates. But this fits well with the volumetric suction characteristic of the main refrigerator screw compressor. At lower mass flow rates it may go to subatmospheric suction pressure. At higher than design flow rates, the cold turbo compressors provide a higher than design pressure ratio and the screw compressor may operate at an elevated suction pressure.

To make this scheme possible, the flow between the cold compressors and the associated screw compressor needs its own channels in the plate-fin heat exchangers.

DAMPENING HEAT EXCHANGER

The pressure ratio which can be reached with this scheme can be stabilised by anchoring the suction side of the third compression stage to the 10 K level of the main refrigeration cycle. This is the temperature range, where the high pressure stream has its peak specific heat. Therefore it is a quite stable range - similar to a condenser in a vapour compression system. At low flow rates, the first two stages of compression provide a relatively low pressure rise and therefore also a small temperature rise to the low pressure stream. Thus the low pressure stream will be warmed up in the 10 K anchor exchanger. So the third stage gets a larger suction volumetric flow rate and can produce a higher pressure ratio.

At high flow rates, the low pressure stream comes at a higher pressure and temperature to this heat exchanger. It is cooled down to 10 K and the duty of the third and fourth stage of compression is reduced.

For the design of the anchoring heat exchanger, we propose to use a plate fin exchanger in cross flow configuration. Since no internal distributors are needed in this design, pressure drop is minimised. Because this heat exchanger is situated at a relatively high pressure and temperature level, the thermodynamic losses are minimal.

FLOW DIAGRAM

Figure 4: Proposed flow diagram

In figure 4 the flow diagram of one of the refrigerators is shown. The screw compressor downstream of the cold compressors is allowed to work with a suction pressure below atmospheric pressure.

Figure 5: Means for handling of dynamic loads

HANDLING OF DYNAMIC LOAD CHANGES

Having established a scheme, which allows efficient and safe operation at all foreseeable quasi- static modes of operation, further dampening means have to be provided which give the refrigerator enough time to adjust to quick load changes. The operation of 3 valves is provisionally foreseen for this purpose:

- V1- a HP bypass valve around the final heat exchanger
- V2- an injection of liquid helium into the cold compressor suction line
- V3- the throttling of 4.4 K vapour into the cold compressor suction line

The arrangement of the valves is shown in Figure 5. The questions, which of these valves will be operated under what conditions will be studied in a dynamic numerical simulation investigation.

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