# DEVELOPMENT OF A LOW-PRESSURE HELIUM COMPRESSION CONTROL SYSTEM STRATEGY

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## ABSTRACT

Cryogenics is now widely used in large accelerator projects using applied superconductivity. Economic considerations require an increase in the performance of superconducting devices. One way to achieve this is by lowering their operating temperature and cooling with superfluid helium. For this reason, large cryogenic systems operating at 1.8 K and capable of producing refrigeration capacity in the kW range have to be developed and implemented. These cryogenic systems require a large pumping capacity at very low pressure using integral cold compression or mixed cold-warm compression. This paper describes the different cooling methods using cold and hybrid cycles, describes the cycle operational capabilities, and reviews the low-pressure helium compression control strategy for these cycles developed at Fermi National Accelerator Laboratory.

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# 1 ABSTRACT

Cryogenics is now widely used in large accelerator projects using applied superconductivity. Economic considerations require an increase in the performance of superconducting devices. One way to achieve this is by lowering their operating temperature and cooling with superfluid helium. For this reason, large cryogenic systems operating at 1.8 K and capable of producing refrigeration capacity in the kW range have to be developed and implemented. These cryogenic systems require a large pumping capacity at very low pressure using integral cold compression or mixed cold-warm compression. This paper describes the different cooling methods using cold and hybrid cycles, describes the cycle operational capabilities, and reviews the low-pressure helium compression control strategy for these cycles developed at Fermi National Accelerator Laboratory.

# 2 INTRODUCTION

The Fermilab Cryomodule Test Facility (CMTF) provides a test bed to measure the performance of cryomodules and superconducting radiofrequency (SRF) cavities for future accelerators. A superfluid cryogenic refrigerator is being built to support testing of SRF components and a new linear accelerator. These SRF components form the basic building blocks of future accelerators such as Project X, ILC, and a Muon Collider. [1]

Maintaining SRF cavities at extremely low-temperatures is required to keep them in a superconductive state. For this purpose, superfluid helium between 1.8 K and 2.0 K temperatures is used. Helium cooling is provided by a cryogenic system developed for testing of SRF cavities.

For SRF components, dynamic heat load (due to RF power dissipation) on average is an order of magnitude greater than the static heat load (due to conduction and thermal radiation). [2] Therefore, one of the important requirements for the CMTF cryogenic system is the capability to operate efficiently over a wide range of heat loads. From the point of view of satisfying this requirement, the analysis of cold and hybrid cycles was done.

# 3 COLD COMPRESSORS

To maintain a saturation temperature of 1.8 K on the heat sink, low-pressure helium vapor must be compressed up to atmospheric pressure with a pressure ratio of 60 and higher. At moderate mass-flow rates, primary vacuum pumps, i.e. rotary-vane, Roots or liquid-ring pumps appear suitable, in spite of the large volumetric capacity imposed by the low density of helium at ambient temperature.

A practical limit for vapor compression at ambient temperature seems to be reached at around 20,000 $m^3$/h when the size of the machines becomes prohibitive. [3] The need for larger cooling power has triggered the development of cold compressors, which handle helium vapor at higher density and thus are more compact machines. Cold helium compressors must be non-lubricated, non-contaminating and thermodynamically efficient, since the heat of compression is rejected at low temperature.

A class of machines well adapted to these requirements is the hydrodynamic type, i.e. centrifugal and axial-centrifugal compressors. However, the limited pressure ratio achievable per stage requires the use of a multistage arrangement, by coupling of several machines in series. This compounds the problem of limited operating range (operation within surge and choke limits, Fig. 1), loss of efficiency away from the design point and ability to adapt to variable flow-rate and fluid inlet density.

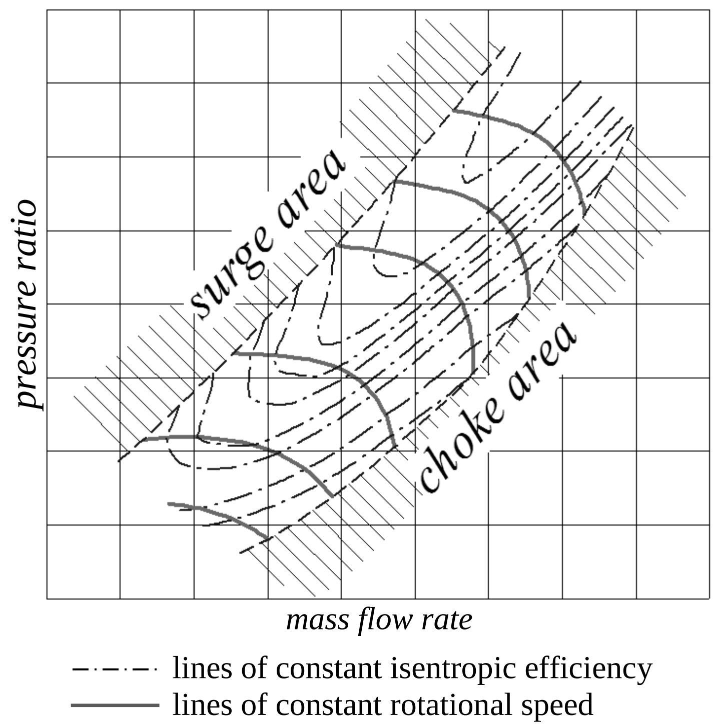



**Fig. 1. Working field of turbine compressor stage**

Besides these intrinsic limitations, cold hydrodynamic compressors also present several technological challenges, involving wheel and housing tolerances, mechanical drive, bearings, and thermal design. In spite of these limitations, hydrodynamic compressors have some advantages over volumetric compressors and it is one of the main reasons hydrodynamic machines are used in low-temperature helium cryogenic systems.

# 4 COLD COMPRESSORS CYCLES

Two of the most common cold compressor cycles for cooling to temperature levels lower than 2.0 K are listed below: [4]

- “cold cycle” based on multistage cold compression using only cold compressors;
- “hybrid cycle” based on multistage cold compression and warm compression at ambient temperature;

Today at the current state of hydrodynamic compressor development it is necessary to use at least four-stage cold compression for the cold cycle and three-stage cold compression for the hybrid cycle. This limitation is due to the maximum achievable limit of modern cold compressor pressure ratio which is within the range of 2.5 to 3.5 depending on the number of stages. For example the overall pressure ratio (excluding hydraulic resistance) for compressing helium vapor at 1.8 K to a pressure of 1 atmosphere is:

$$p_{atm} / p_{sat.\ liq.}|_{T = 1.8\ \mathrm{K}} = 101325\ \mathrm{Pa} / 1638.41\ \mathrm{Pa} = 61.8$$

This pressure ratio is only achievable using four-stage cold compression.

A simplified scheme of the low-temperature region of the cold cycle is shown in Fig. 2, the hybrid cycle low-temperature region is shown in. Fig. 3.

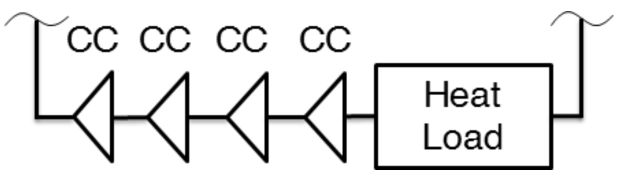



**Fig. 2. Scheme of the cold cycle low-temperature region**

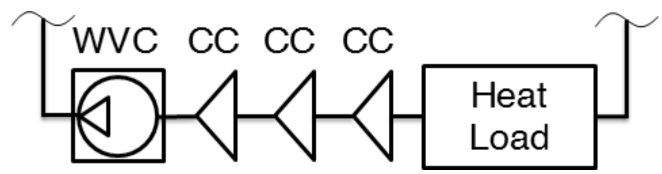



**Fig. 3. Scheme of the hybrid cycle low-temperature region**

In these cycles helium vapor pumping is performed by:

- serial compression in four cold compressor stages to the pressure of one atmosphere for the cold cycle;
- serial compression in three cold compressor stages up to interstage pressure (the suction pressure of warm vacuum compressor) and final compression in the warm vacuum compressor to the pressure of one atmosphere for the hybrid cycle.

The change of helium evaporator heat load caused by the changing mass flow rate through the compression system leads to changing of the cold compressor working region. It can move the working region of the cold compressors out from their respective working fields. Therefore, decreasing the mass flow rate through the compressor at a constant pressure ratio moves the working mode of the compressor closer to the surge area. Increasing the mass flow rate moves the working mode of the compressor closer to the choke area. This in turn results in reduced compressor efficiency or can even result in a compressor emergency stop.

Helium evaporator working modes often have a transient character related to temperature level or heat load. The dependence on the pumping helium vapor mass flow rate from the heat load and temperature level has a specific character due to the complex dependence of liquid helium evaporation heat from the temperature (Fig. 4). This dependence can be found using the following equation:

$$Q = r \times M,$$

*where* $Q$ is the helium evaporator heat load, [W]; $r$ is the evaporation heat of liquid helium, [J/kg]; $M$ is the helium mass flow rate through the cold compression system, [kg/s].

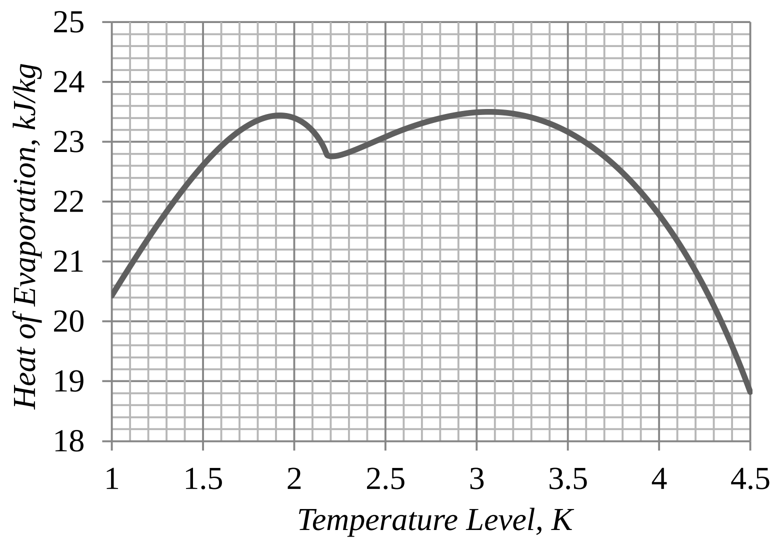



**Fig. 4. Dependence of liquid helium evaporation heat from the temperature level**

The working regimes of SRF cryomodules are typically unstable due to the dynamic character of the heat load caused by RF power dissipation. This requires developing a cryogenic system capable of working efficiently over a wide range of heat loads and at various temperature levels. To satisfy this requirement the helium evaporator heat load must be capable of changing in the proper range. Therefore the pumping helium vapor mass flow rate also must be changed in the appropriate range.

Due to the specific character of the cold compressor working fields (Fig. 1) the changing of helium mass flow rate through the cold compressor stages causes the cold compressor to move away from the optimal working regime with a decrease in efficiency. Further changes in mass flow rate often move the working compressor regime away from the working field. This phenomenon is unavoidable and takes place to a varying degree in all cold compressor cycles.

To control this phenomenon the bypass control method can be used. This method consists of redirecting a portion of the flow around the cold compressors (Fig. 5) so that it becomes possible to move the compressor working mode from the low-efficiency area or surge area by increasing the mass flow rate through the compressor.

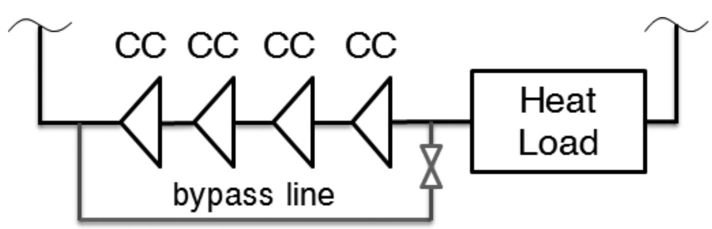
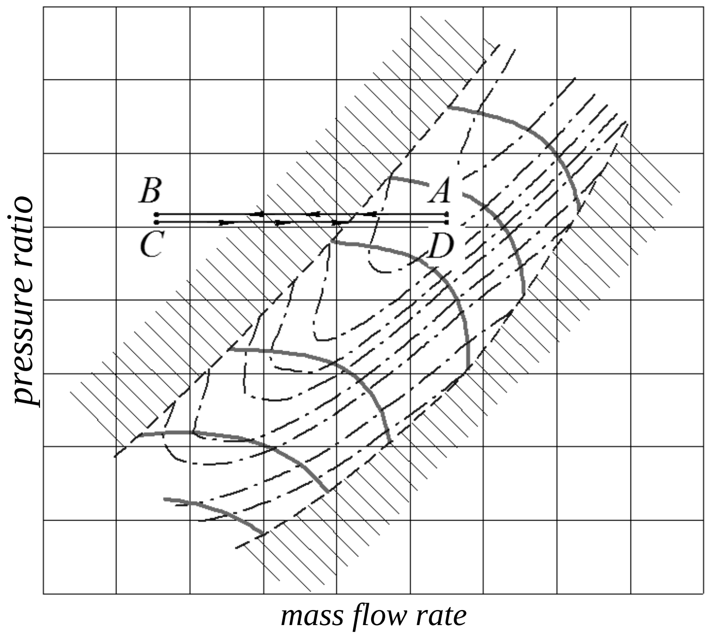



**Fig. 5. The scheme of bypass control method.**

This method is only used to move the compressor working mode from the surge area closer to the design point (Fig. 6), i.e. to avoid decreasing mass flow rate through the compressor. In terms of the helium refrigerator this translates to a decreased heat load of the helium evaporator. Therefore, the nominal working mode for the cold compressors must be designed to maximize the heat load and pumping helium vapor mass flow rate. Such systems are capable of working without additional bypass control limited only by the range of heat load. This decreases the overall efficiency of the cryogenic system in off-design modes (with lower heat load) due to bypass control method inefficiency.




AB – moving compressor working mode away from working field to the surge area
CD – returning compressor working mode from the surge area back to the working field

**Fig. 6. Bypass method of compressor working modes controlling**

There is also another method of compressor stability control. It is possible to decrease helium evaporator heat load fluctuations using additional resistive heat load in the system. But it is obvious that this solution is not economical due to the use of cryogenic power for compensation of additional heat load.

One of the most important requirements for the CMTF cryogenic system is the capability to operate efficiently over a wide range of heat loads. Therefore using the cold cycle for this cryogenic system, results in a higher operating cost.

To resolve the problem of higher operating cost it is necessary to develop a cryogenic cycle that does not require a high operating cost control method or minimizes the need of such a control method. Such a control scheme requires operation in the compressors working fields with a wide range of mass flows. To be able to operate in these modes it is necessary to control the pressure ratio of the compressors when the heat load and mass flow rate are changed (Fig. 7).

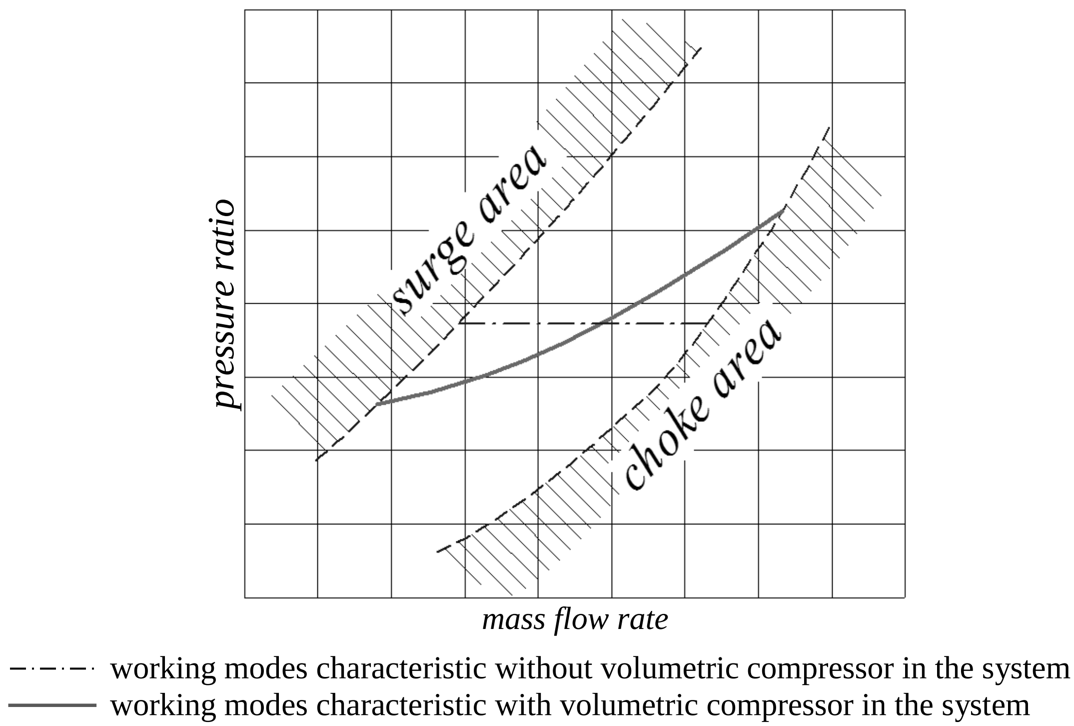



**Fig. 7. Dependence cold compressors working characteristics from volumetric compressor**

To achieve this, the presence of an element with a pressure head characteristic that varies with mass flow rate in the system is necessary. A volumetric compressor satisfies this requirement. A sample characteristic of a volumetric compressor is shown in Fig. 8.

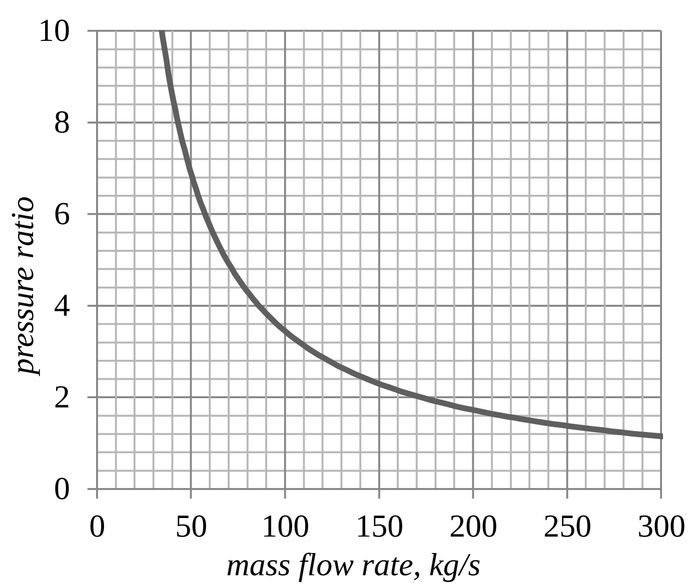



**Fig. 8. Volumetric compressor pressure head theoretical characteristic at fixed discharge pressure (101325 Pa)**

The overall pressure ratio of cold compressor systems can be achieved using different combinations of individual compressor pressure ratios. These combinations are characterized by each compressors individual efficiency and pressure ratio.

The control system strategy for cold or hybrid cycles not only involves operating the cold compressors in working field regions away from the surge and stall areas, it is also involves maintaining of a maximum compressor efficiency at defined helium evaporator parameters (temperature level and heat load). Cold compressor calculation results demonstrate good consistency with these control tasks. Thereby maximization of one of the criteria is enough for cold compressor control system stability. Due to the complexity and inaccuracy of efficiency parameters, using a strategy based on maximizing the distance of the working modes from the surge and choke areas was chosen.

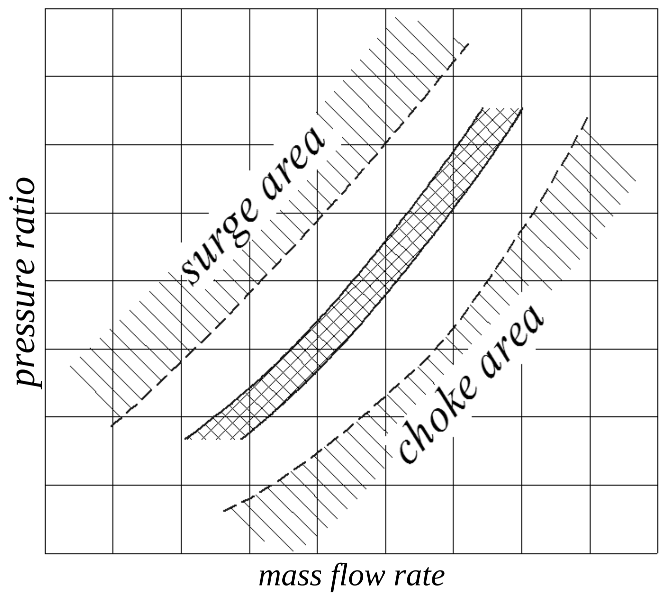



**Fig. 9. Cold compressor working area with maximum stability**

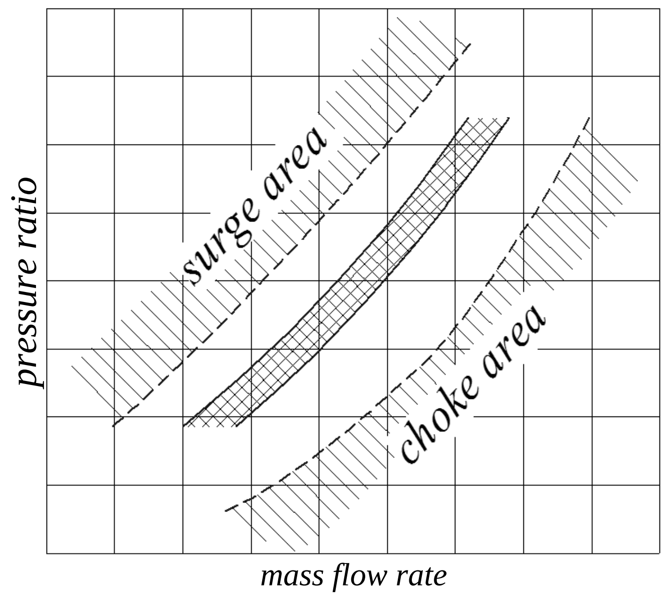



**Fig 10. Cold compressor working area with maximum efficiency**

# 5 LOW-PRESSURE HELIUM COMPRESSION PROCESS SIMULATION

The compression process was simulated for the cycle based on cold compressors and for the hybrid cycle.

## 5.1 Simulation process algorithm for cold compression based cycle

1. Define the first cold compressor stage suction pressure at a given liquid helium temperature in the helium evaporator (pressure of helium saturated vapor).
2. Define the overall pressure ratio of the cold compressors system using the first cold compressor stage suction pressure and the last cold compressor stage discharge pressure (1 atmosphere).
3. Define the cold compressor working field outside of the surge and choke areas using the pressure ratio of each respective cold compressor stage.

4. Define subareas satisfying the condition of achieving the overall pressure ratio defined in step 2 within the cold compressor working areas.
5. Choose the most conducive operating modes inside of the working area defined in step 4 for each working mode characterized by the given helium mass flow rate and heat load.

The working modes of the helium evaporator are defined by the temperature level (step 1) and by the heat load (step 5).

## 5.2 Simulation process algorithm for hybrid compression based cycle

1. Define the first cold compressor stage suction pressure at a given liquid helium temperature in the helium evaporator (pressure of the helium saturated vapor).
2. Define the volumetric compressor suction pressure for the given helium evaporator heat load and respective helium mass flow rate through the system using a “mass flow rate” versus “suction pressure” volumetric compressor pressure head characteristic (at a constant 1 atmosphere discharge pressure).
3. Define the overall pressure ratio of the cold compressor system using the first cold compressor stage suction pressure and the warm vacuum compressor suction pressure.
4. Define the cold compressor working fields outside of the surge and choke areas with the pressure ratio of each respective cold compressor stage.
5. Define the subareas satisfying the condition of achieving the overall pressure ratio defined in step 3 within the cold compressor working areas.
6. Choose the most conducive operating modes inside of the working area defined in step 5 for each working mode characterized by the given helium mass flow rate and heat load.

The working modes of the helium evaporator are defined by the temperature level (step 1) and by the heat load (steps 2, 6). The cryogenic systems under consideration are designed for the maximum efficiency at 2.0 K temperature level.

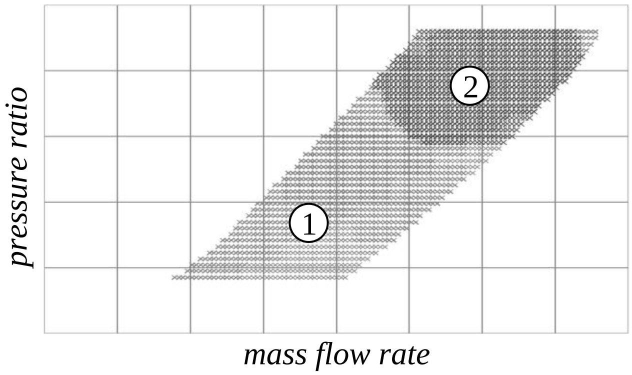



**Fig. 11a. Cold cycle; 2.0 К;**
**1st cold turbine compressor stage**

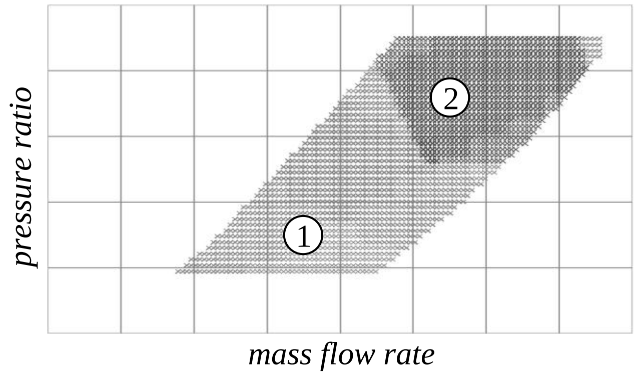



**Fig. 11b. Cold cycle; 2.0 К;**
**2nd cold turbine compressor stage**

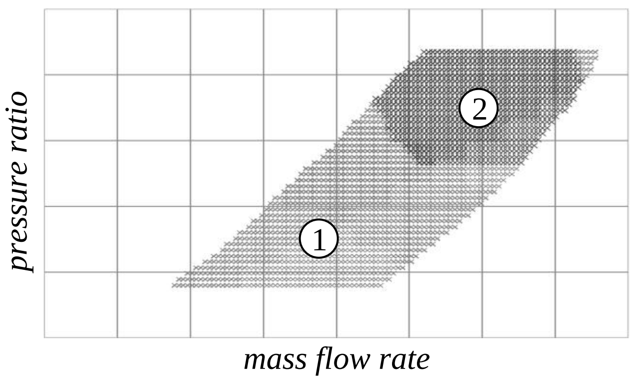



**Fig. 11c. Cold cycle; 2.0 К;**
**3rd cold turbine compressor stage**

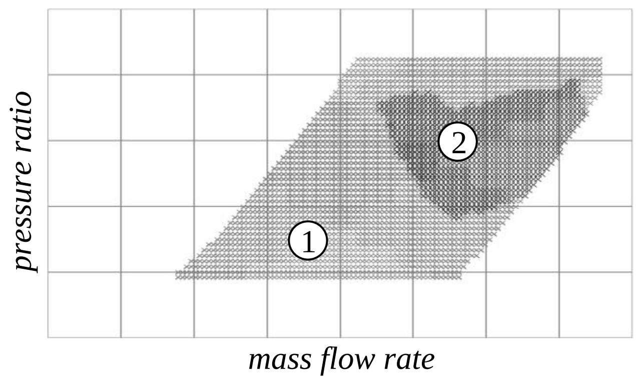



**Fig. 11d. Cold cycle; 2.0 К;**
**4th cold turbine compressor stage**

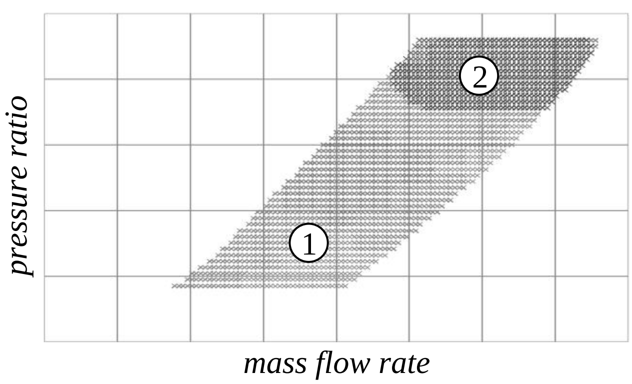



**Fig. 12a. Cold cycle; 1.96 К;**
**1st cold turbine compressor stage**

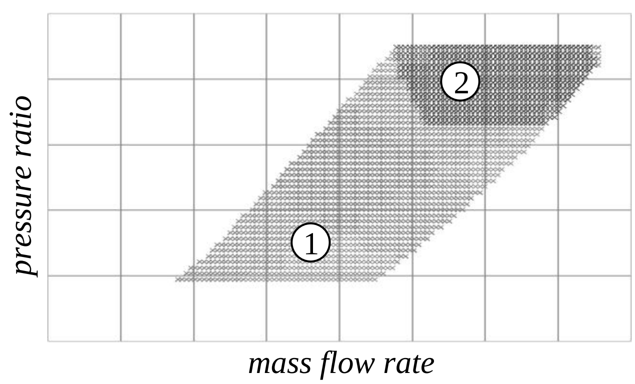



**Fig. 12b. Cold cycle; 1.96 К;**
**2nd cold turbine compressor stage**

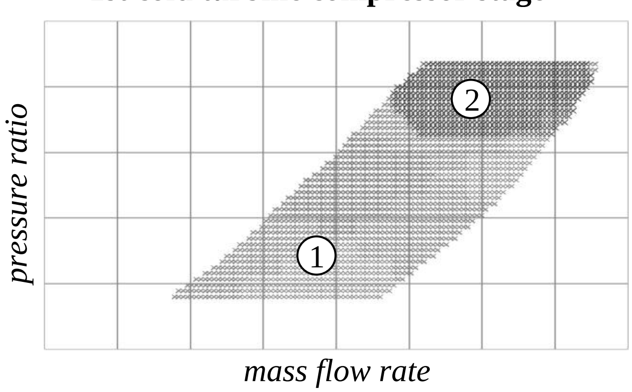



**Fig. 12c. Cold cycle; 1.96 К;**
**3rd cold turbine compressor stage**

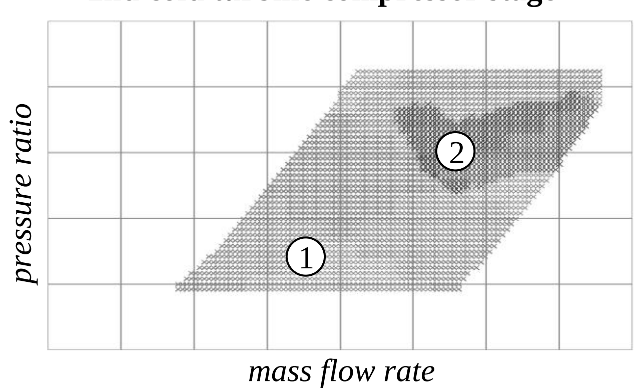



**Fig. 12d. Cold cycle; 1.96 К;**
**4th cold turbine compressor stage**

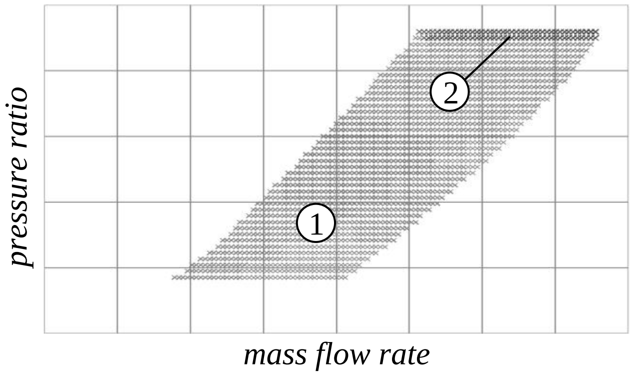


**Fig. 13a. Cold cycle; 1.91 K;**
**1st cold turbine compressor stage**

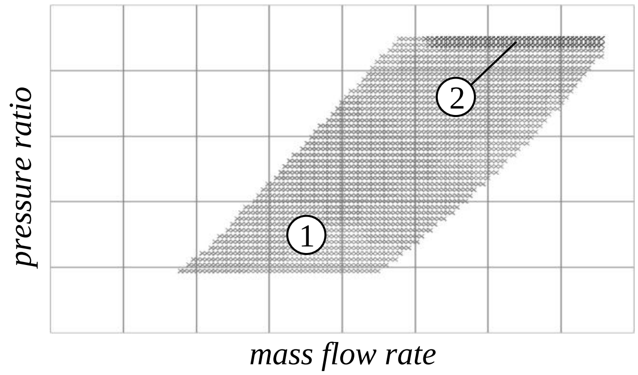


**Fig. 13b. Cold cycle; 1.91 K;**
**2nd cold turbine compressor stage**

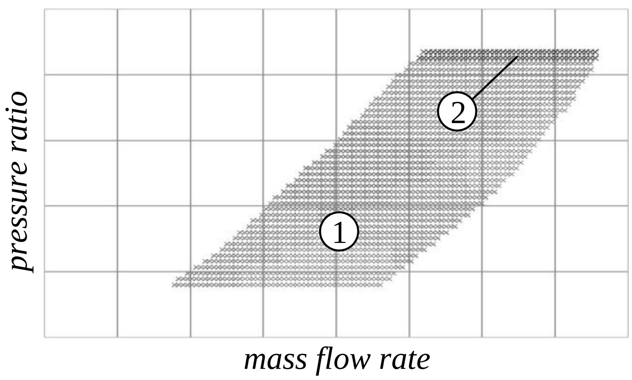


**Fig. 13c. Cold cycle; 1.91 K;**
**3rd cold turbine compressor stage**

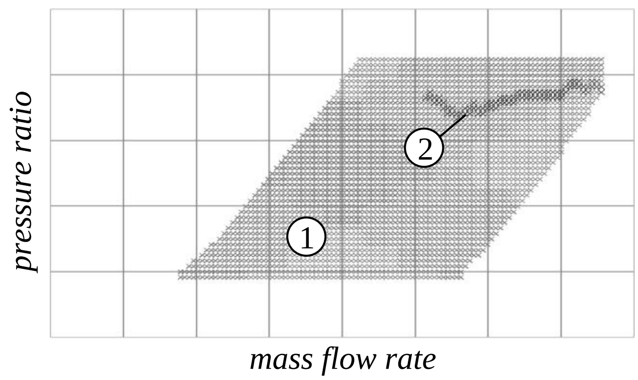


**Fig. 13d. Cold cycle; 1.91 K;**
**4th cold turbine compressor stage**

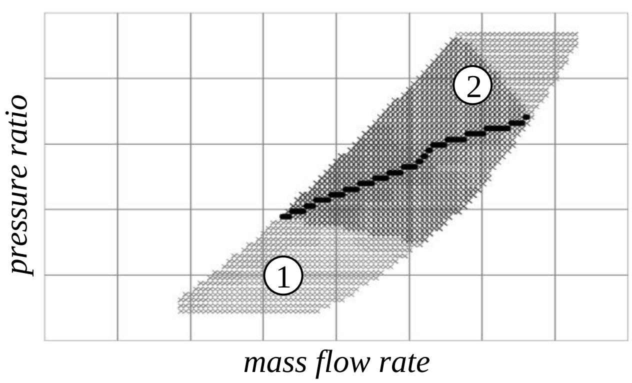


**Fig. 14a. Hybrid cycle; 2.0 K;**
**1st cold turbine compressor stage**

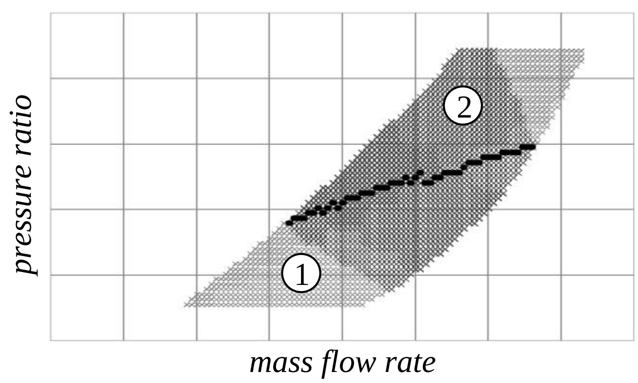


**Fig. 14b. Hybrid cycle; 2.0 K;**
**2nd cold turbine compressor stage**

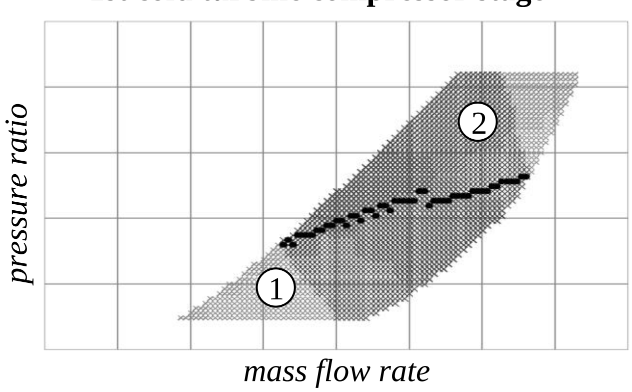


**Fig. 14c. Hybrid cycle; 2.0 K;**
**3rd cold turbine compressor stage**

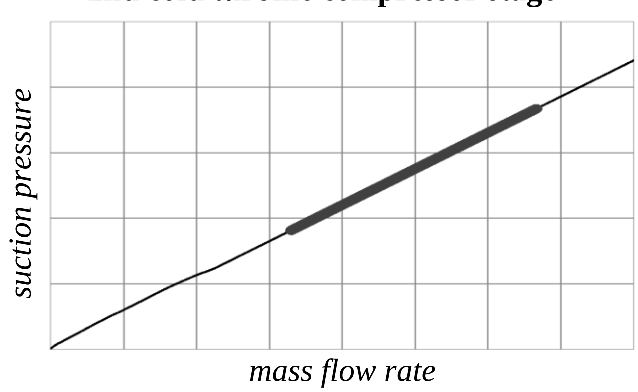


**Fig. 14d. Hybrid cycle; 2.0 K;**
**volumetric warm compressor**

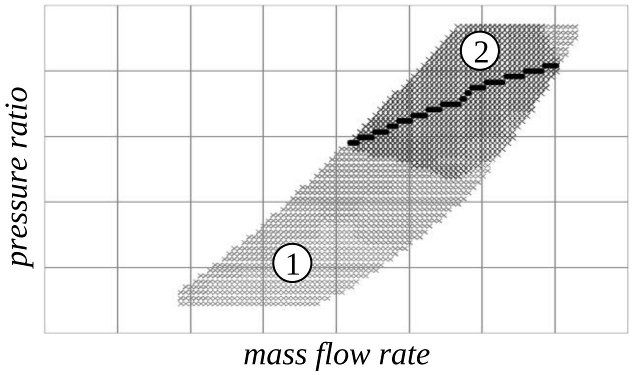


**Fig. 15a. Hybrid cycle; 1.9 K;**
**1st cold turbine compressor stage**

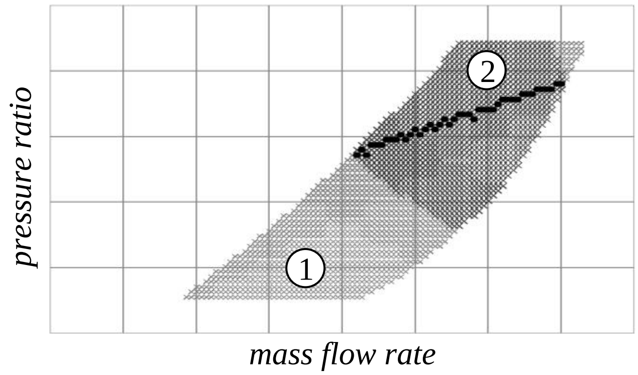


**Fig. 15b. Hybrid cycle; 1.9 K;**
**2nd cold turbine compressor stage**

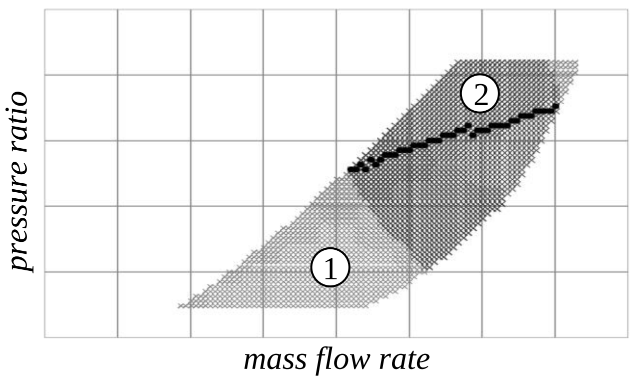


**Fig. 15c. Hybrid cycle; 1.9 K;**
**3rd cold turbine compressor stage**

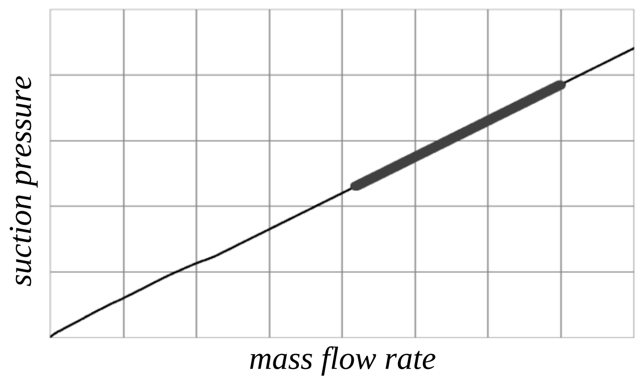


**Fig. 15d. Hybrid cycle; 1.9 K;**
**volumetric warm compressor**

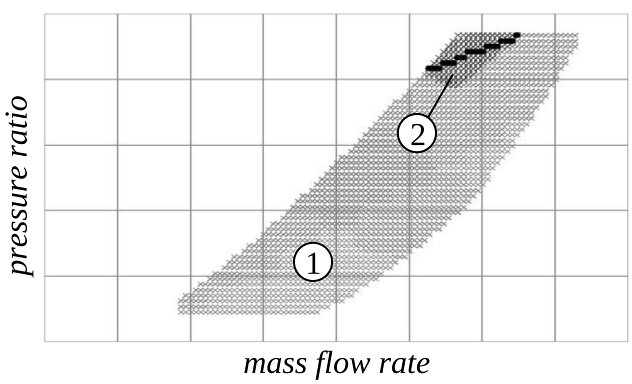


**Fig. 16a. Hybrid cycle; 1.8 K;**
**1st cold turbine compressor stage**

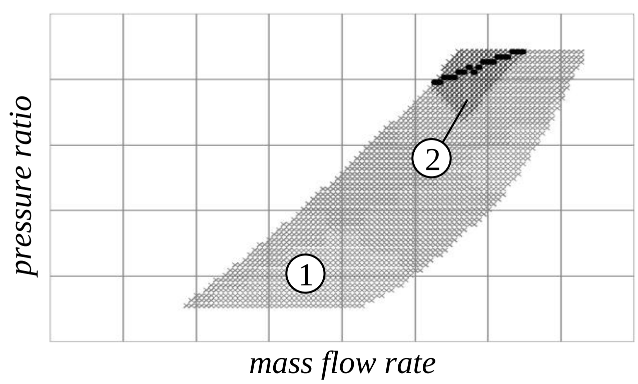


**Fig. 16b. Hybrid cycle; 1.8 K;**
**2nd cold turbine compressor stage**

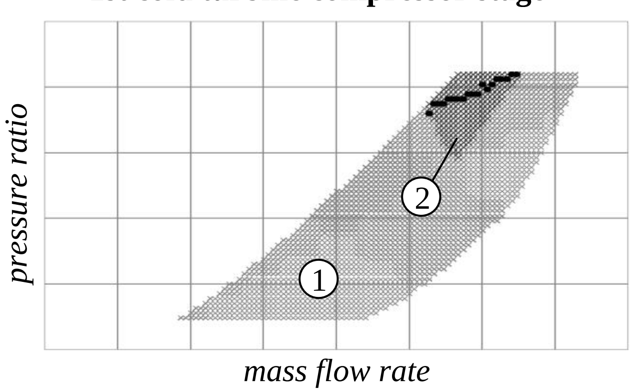


**Fig. 16c. Hybrid cycle; 1.8 K;**
**3rd cold turbine compressor stage**

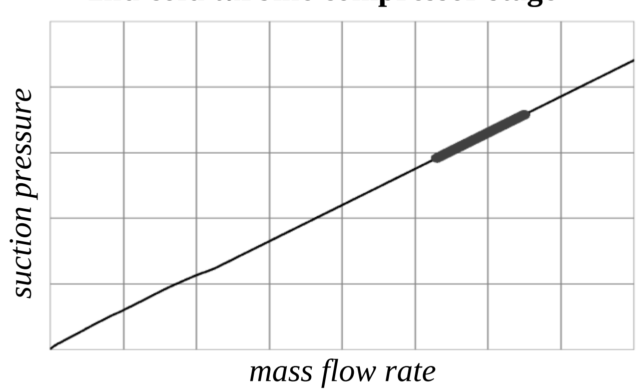


**Fig. 16d. Hybrid cycle; 1.8 K;**
**volumetric warm compressor**

# 6 COMPARISON OF COLD AND HYBRID CYCLE CALCULATION RESULTS

Figure 11a-d, 12a-d, 13a-d and 14a-d, 15a-d, 16a-d show the results of the cold and hybrid cycle calculations at the different temperature levels. Area “1” represents the cold compressor working fields outside of the surge and choke area with a pressure ratio supported by the respective cold compressor stage. Area “2” represents the working fields which achieve an overall pressure ratio by the cold compressor system. For hybrid cycles the most optimal working modes (from a working stability point of view) for all helium mass flow rates through the system are shown. Also the working areas for the volumetric warm compressor are shown in the “suction pressure” versus “mass flow rate” characteristic.

These plots show a cold compression based cycle with low flexibility in relation to the heat load and temperature level as compared to the hybrid cycle. Due to the absence of a volumetric compressor in the cold compression system the working characteristics of the cold compressors operate within a limited range of pressure ratio values. This limits the cold compressor working areas and thus the heat load values.

The helium evaporator heat load reduction capability is mainly defined by the cold compressor ability to operate at the reduced mass flow rate relative to the design characteristics. Increasing the heat load relative to the design characteristic is possible only in a small range defined by the heat exchangers perfomance. Therefore the capability of the cryogenic system to increase the heat load is not considered. Fig. 17 shows a comparison chart of the heat load ranges for the cold and hybrid cycles without use of additional control methods (bypass or resistive heating).

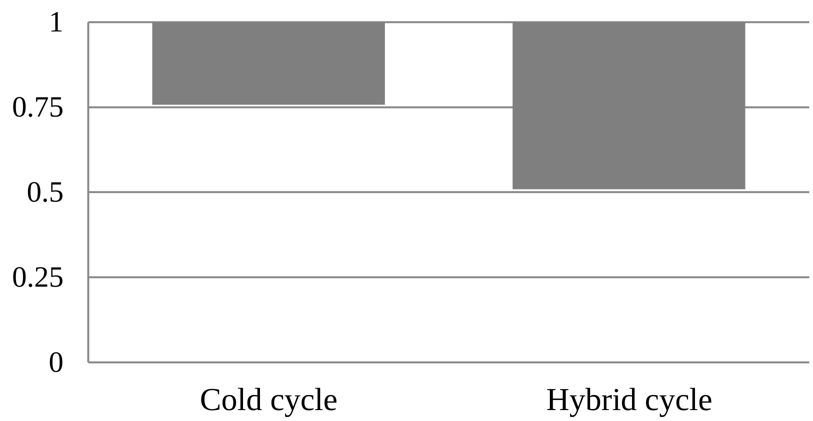



**Fig. 17. Heat load reduction capabilities for the cold and hybrid cycles without using of additional control methods**

A two-fold excess of the hybrid cycle mass flow range over the cold cycle mass flow range means the hybrid cycle has a higher efficiency in the working region with reduced mass flow rate because decreasing the heat load below the minimum value (51% of the nominal for hybrid cycle and 76% of the nominal for cold cycle) causes moving working parameters inside of the surge area. To return cold compressors working parameters back to the working field from the surge area using one of considered control methods (bypass or resistive heating) is needed. Thereby the hybrid cycle is a more efficient control method and increases the overall cryogenic system efficiency in off-design modes with reduced helium evaporator heat load.

On the other hand, analysis of the plots shown in Fig. 11a-d, 12a-d, 13a-d and 14a-d, 15a-d, 16a-d shows the cold cycle inability to decrease the helium evaporator temperature level. Even at the 1.91 K temperature level the cold compressor working areas are reduced as you move to a maximum pressure ratio limit. Further temperature reduction in the helium evaporator is not possible because it causes an increase in overall cold compressor system pressure ratio to an unachievable value.

Use of a warm volumetric compressor in the system with a pressure head characteristic dependent on the mass flow rate enables a change in the cold compressor working fields towards a region with a lower pressure ratio. Thus the presence of a volumetric compressor in a low-pressure helium compression system enables an improvement in the cold compressor working conditions and allows a lower pressure value at the suction of the first cold compressor stage. This in turn enables a decrease in temperature level at the helium evaporator increasing helium saturated vapor pumping.

# 7 COMPRESSION CONTROL STRATEGY OF A HYBRID CYCLE

The hybrid cycle compression control strategy consists of:

- the cold and warm compressor system startup process strategy;
- regulation of the helium evaporator temperature level;
- regulation with changing helium evaporator heat load;

Cold compressor system control is possible by changing only one working parameter of the cold compressors, the rotational speed. The other working mode characteristics of all the compressors are uniquely defined by the following external parameters:

- helium evaporator heat load;
- required helium evaporator temperature level.

The helium evaporator heat load uniquely defines the required pumping helium vapor mass flow rate. Using this parameter the suction pressure of the volumetric warm compressor (the discharge pressure of the last cold compressor stage) is defined. The required helium evaporator temperature level defines the helium saturated vapor pressure above the liquid helium. This pressure is provided at the suction of the first cold compressor stage. Using the defined cold compressor system overall pressure ratio, the optimization of every cold compressor working regimes is made (Fig. 18).

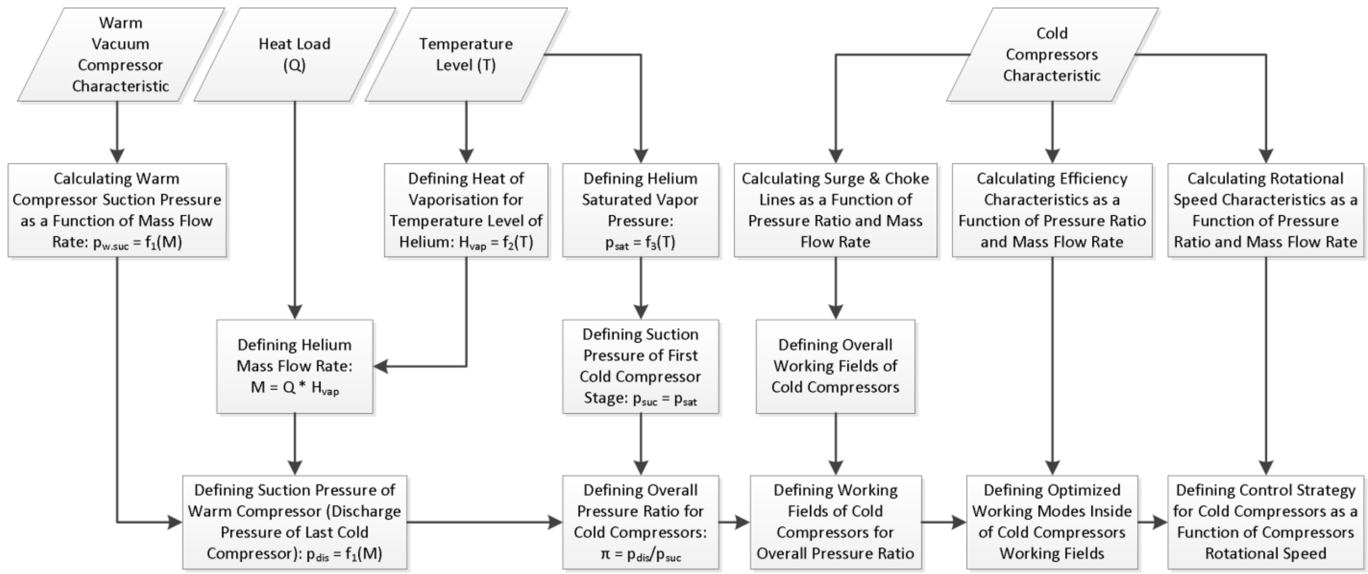



**Fig. 18. Cold compressors control system strategy calculation scheme**

# 8 CONCLUSION

In this paper the different cryogenic cycles for low temperature helium production was observed. Cold compression is needed in these cycles due to the large volumetric capacity imposed by the low density of helium at ambient temperature. A low-pressure helium compression common control strategy was developed. Using this strategy the low-pressure helium compression processes with variable parameters was simulated. The results of the simulation show that the hybrid cycle is more adapted for the Fermilab CMTF system than the cold cycle. Using a hybrid cycle has certain advantages such as higher efficiency and a lower operating cost than the cold cycle.

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