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Modeling and commissioning of a cold compressor string for the superfluid cryogenic plant at Fermilab's cryo-module test facility

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Abstract

In 2011, Linde Cryogenics, a division of Linde Process Plants, Tulsa, Oklahoma, was awarded the contract to deliver a 500 W at 2 K superfluid cryogenic plant to Fermi National Accelerator Laboratory (FNAL) in Batavia, Illinois, USA. This system includes a cold compressor string with three centrifugal compressors and a vacuum pump skid with five volumetric pumps in parallel used to pump down helium to its saturation pressure corresponding to 2 K. Linde Kryotechnik AG, Pfungen Switzerland engineered and supplied the cold compressor system and commissioned it with its control logic to cover the complete range of system operation. The paper outlines issues regarding compressor design, compressor string modeling, control algorithms, controller performance, and surge protection.

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Keywords: cold compressor; cryogenics; superfluid; helium; refrigeration; centrifugal compressor; logic; control; surge; choke; frequency converter; pump down; pump up; transient; steady state; lambda point; liquid helium; model based control; proportional integral control; mass flow control

1. Introduction

Cold compressors are used in helium refrigeration plants to generate sub-atmospheric boil-off pressures for liquid helium. The reduction of gas pressure above the surface of liquid helium changes the gas-liquid saturation

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conditions. Lower saturation pressures result in lower saturation temperatures. Approximately at 2.17 K liquid helium becomes superfluid. Cold compressors are designed to pump boil-off gas from the surface of liquid helium, until the saturation pressure corresponding to the desired temperature is reached. The reduction of pressure by the cold compressors is one method to achieve the superfluid transition of liquid helium. So far they have been used in few helium refrigeration plants to reach superfluid state by compressing helium from sub-atmospheric pressures to higher discharge pressures [Claudet G. et al (1990)].

The required total pressure ratio is given by the suction pressure of the warm compressor system (typically 1.05 bara) divided by the requested liquid helium bath pressure (typically 24 mbar corresponding to 1.9 K). This pressure ratio can be achieved either using vacuum pumps at ambient temperature, by cold compressors in series at cryogenic temperatures or a mixture of both systems [Kuendig A. et al (2002)]. Cold compressors are first choice. They help to minimize heat exchanger cross sections and reduce the warm vacuum pump unit dimensions or even delete them from the scope and thus minimize capital expenditures. One of the main advantages of cold compressors is the improved heat transfer with the higher density gas [Van Sciver W.S. (1986)]. Beside they also reduce the risk of air in-leak as they also get mounted to the vacuum insulated coldbox. But cold compressors lose efficiency being down-scaled as heat in-leak becomes a major handicap with decreasing throughput. The optimum distribution between number of cold compressors and suction of warm vacuum pump has to be considered under these aspects.

The cryogenic plant at FNAL's CMTF consists of three cold compressors in series being followed by five volumetric vacuum pumps in parallel. The main task of cold compressors is to keep the saturation pressure of superfluid helium on the suction side constant. The control system has to gather all the required information and set the rotational speeds in such a way as to fulfill this pressure stability requirement and keep all the cold compressors in the safe and efficient zones of the compressor maps. The paper covers the control methodology and the compressor string model as applied on the cold compressor system of FNAL's cryo-plant.

Nomenclature

LKT	Linde Kryotechnik AG
FNAL	Fermi National Accelerator Laboratory
CMTF	Cryogenic module test facility
CC	Cold compressor
LN	Natural logarithm
PI	Proportional-integral

1.1. Cold compressor specifications

The plant has two main refrigeration modes, design at 500 W at 2 K and 250 W at 1.8 K. Table 1 shows the design values. Temperature and pressure deviations from the design point influence the density of the flow. To compensate for this variation of the density and the volumetric flow rate the measured mass flow rate and the measured rotational speed need to be converted to unit-less reduced values forming the basis of the characteristic compressor maps [Baines N.C. et al. (1990)].

Table 1. Cold compressor specifications.

Design values		CC 1	CC 2	CC 3
P_{des}	[mbar]	23.7	97.3	221
T_{des}	[K]	4.05	8.25	12.64
M^*_{des}	[g/s]	26.8	26.8	26.8
N_{des}	[Hz]	540	707	707

Equation (1) is used to calculate the reduced mass flow rate. P and T are the measured pressure and temperature at the inlet of each compressor:

$$M^*_{red} = \frac{M^* \cdot P_{des}}{M^*_{des} \cdot P} \cdot \sqrt{\frac{T}{T_{des}}}. \tag{1}$$

The equation (2) is used to calculate the reduced speed:

$$N_{red} = \frac{N}{N_{des}} \cdot \sqrt{\frac{T_{des}}{T}}. \tag{2}$$

1.2. Compressor maps

Compressor maps define the operating boundaries of the cold compressors. Reduced mass flow rate calculated using equation (1) is the x-axis of the compressor map. The pressure ratios π_1, π_2 and π_3 of CC_1, CC_2 and CC_3 are found using equation (3).

$$\pi_1 = \frac{P_{out\_1}}{P_{in\_1}} = \frac{P_1}{P_0},\ \pi_2 = \frac{P_{out\_2}}{P_{in\_2}} = \frac{P_2}{P_1},\ \pi_3 = \frac{P_{out\_3}}{P_{in\_3}} = \frac{P_3}{P_2}. \tag{3}$$

The pressure ratio is the y-axis on each compressor map. In order to reach high pressure ratios the reduced mass flow rate passing all compressors has to be increased. Fig. (1) shows the compressor maps of each compressor. The location of the five operation curves is determined considering the total reaction of the system. A detailed explanation is given in chapter 2 "Cold compressor string model". Each compressor map is stretched by 5 so-called capacity (X) curves. The curves lying on the surge boundary are called X=0, the curves stretching through high mass flow zone X=1. The other 3 in the middle are X=0.25, X=0.5 and X=0.75 and their distribution is inhomogeneous and different from one compressor to another.

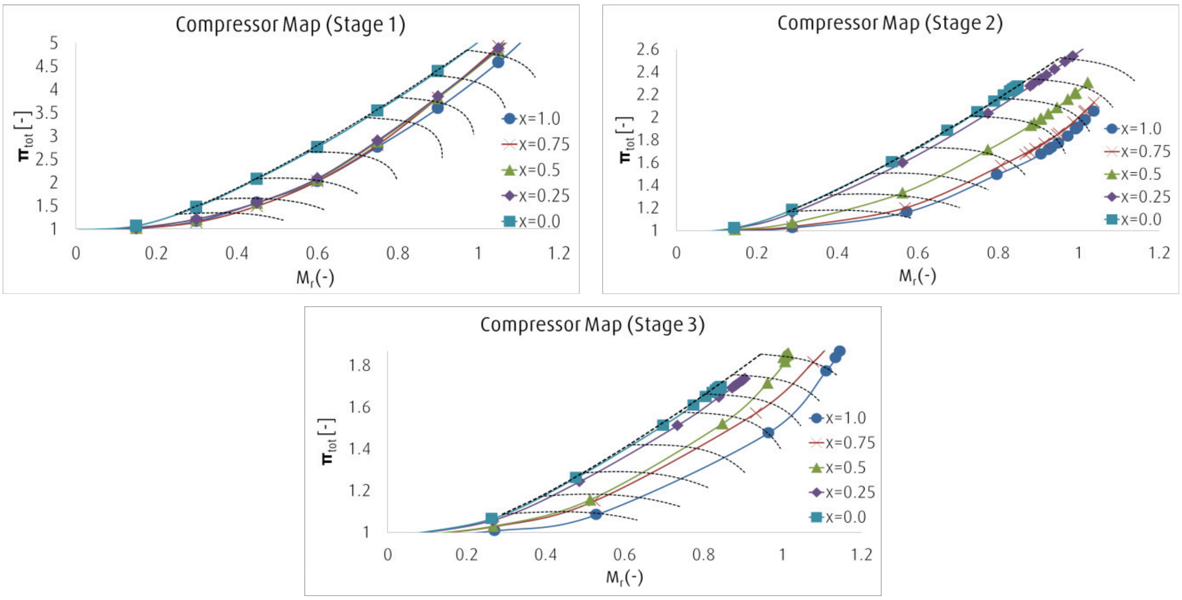

Fig. 1. Compressor maps with operation curves.

2. Cold compressor string model

The compressor string is modeled by defining five operating curves between the surge and choke zones of each compressor map. The inlet temperature and pressure of CC_1 are the boundary conditions of the model and they are assumed to be equal to design values. The region of all possible reduced mass flow rates for CC_1 is defined to be between 0 and 1.2. Based on the compressor map at each value of the reduced flow rate five reduced speed values between surge and choke are selected. The speed value on the surge boundary is defined to be on the X=0 curve, which is called the low capacity curve, meaning that it corresponds to the lowest reduced flow rate possible. The value on the choke boundary is defined to be on the X=1 curve, meaning that it is the highest possible reduced flow rate.

The distribution of the curves can be non-homogenous, i.e. the numbering is just a description of the order. By defining five curves stretching along the compressor map of CC_1, at each reduced flow rate between 0 and 1.2, 5 reduced speeds between the surge and the choke boundary can be selected. Equation (2) is used to convert the obtained reduced speed value into a real value. The same process is repeated for CC_2 and CC_3. The distribution of the capacity (X) curves is determined in such a way in order to obtain steadily increasing reduced speeds for increasing total pressure ratios. Fig. 2 shows the reduced speed curves of all cold compressors. X-axis is the natural logarithm of the total pressure ratio and the y-axis is the reduced speed to be used as the set speed.

The purpose of building a model is to generate a compressor string map including all three compressors. This map will use the total pressure ratio of the compressor string and the capacity value X to determine at which point of the compressor map the compressors are rotating. If the curves for CC_1 are determined, then the compressor can be simulated for all possible reduced flow rates using the reduced speed values corresponding to the X curves. This in return generates varying inlet conditions for the downstream compressor. Based on equation (1), varying inlet conditions result in a different reduced mass flow rate for the downstream stage. The downstream compressors are subsequently simulated using the same X value to obtain numerous total pressure ratios, which are functions of the X value and the reduced flow rate. Fig.2 depicts the output curves of the model, which are to be uploaded on a programmable logic controller in form of look-up tables to control the compressor speeds.

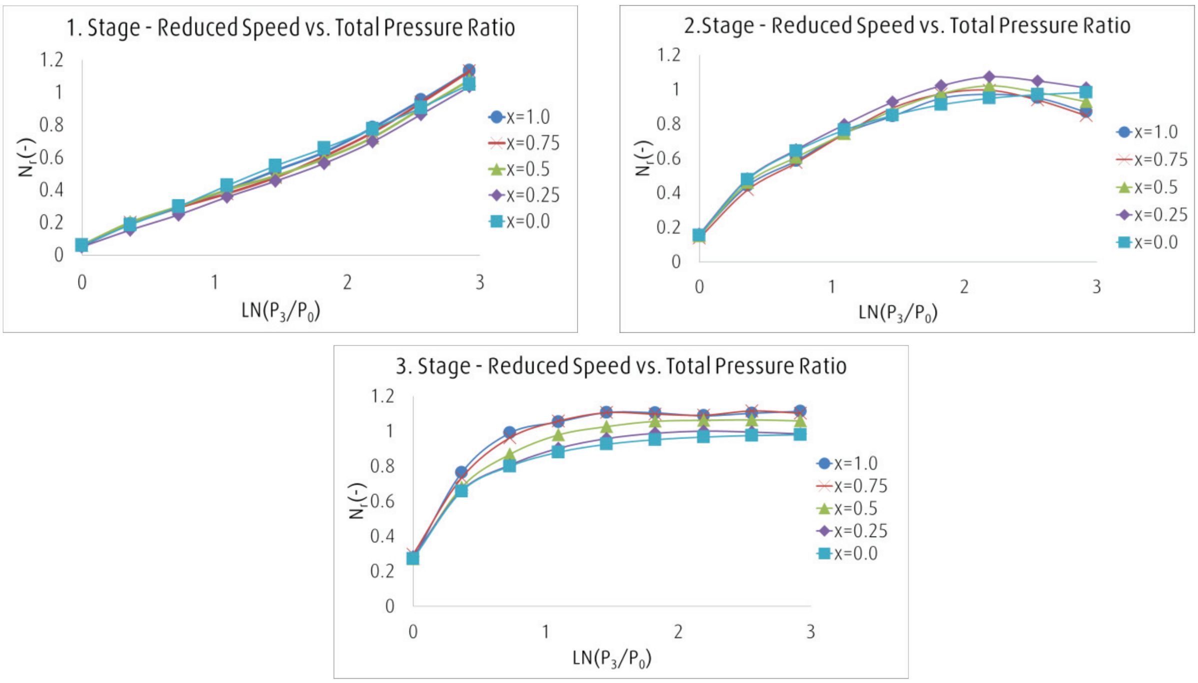

Fig. 2. Reduced speed curves.

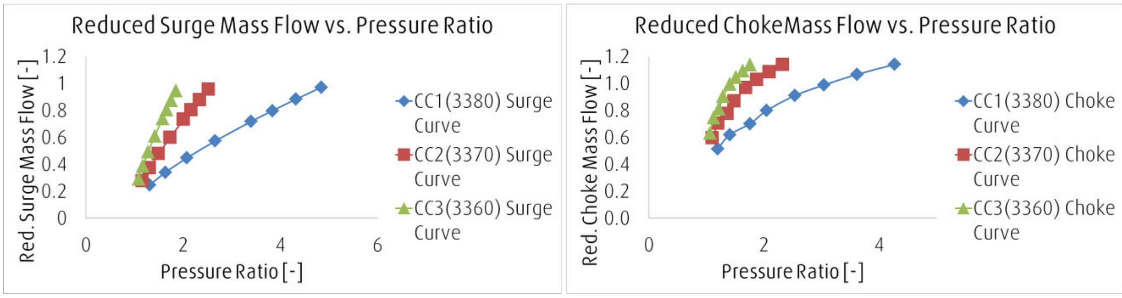

Fig. 3. Reduced mass flows on surge and choke curves.

3. Controller

The control system uses a PI controller. The process variable is the boil-off pressure of the liquid helium used as coolant. Depending on the operation mode the set pressure is either the saturation pressure corresponding to 2 K or the saturation pressure corresponding to 1.8 K. The control variable, i.e. the output of the controller, is the capacity (X) value and is limited to 0.05 and 0.9 to avoid surge and choke. If the process variable is higher than the set pressure, the value of X is increased towards 0.9.

The natural logarithm of the total pressure ratio across the compressor string is described as LN(P_3/P_0). Using the value of X calculated by the PI controller and the value of LN(P_3/P_0) the model-based control system can calculate all three reduced speeds. The reduced speeds are converted to real speeds using equation (2) and sent to the corresponding frequency converter as set speed. Due to reciprocal influences between the compressor stages and the non-ideal behaviour of the compressors away from their design points it may be the case that the value of X gets saturated before the set pressure on the suction side is reached. In that case the measured total pressure ratio LN(P_3/P_0) is sligthly increased, which results in the model sending higher reduced set speeds corresponding to higher total pressure ratios.

4. Surge protection

The model continuously calculates reduced speeds lying on the surge boundary as a function of the reduced flow rate. As shown in equation (4) the reduced speed value on the surge curve is a function of the reduced flow rate found using equation (1).

$$N_{red\_surge} = f(M^{*}_{red}). \tag{4}$$

The model-based control determines at each instant the reduced set speeds using the curves on Fig. 2 as look-up tables. The reduced set speeds are always limited to the value calculated using equation (4) with an offset. Beside the set speed, the measured real speed is continuously converted to the reduced form using equation (2) and compared to the reduced speed on the surge curve. If the surge speed is crossed, a counter is started, which counts the number of crossing events. In case the event has a tendency to repeat, the compressor string may be either switched off or a gas supply valve will be opened to send more mass flow rate.

The compressor string has also its own by-pass valve sending gas from the outlet of the string to the inlet. At steady state condition, if enough mass flow rate flows through the compressors, the valve is closed. During pump-down at high suction pressure the reduced flow rate is low and in order to set higher speeds to increase the pressure ratio the compressors need more mass flow. Fig. 3 shows the minimum and maximum reduced flow rates for each compressor as a function of pressure ratio. The minimum mass flow rate corresponds to the surge curve and the maximum flow rate corresponds to the choke curve. The choke curve is defined using the highest reduced mass flow rates at which each reduced speed curve is defined. Surge protection algorithms in the controller continuously calculate the minimum required mass flow rates. The highest of the minimum required mass flow rates from 3 compressors is increased by an offset and set as mass flow set point. At steady state the mass flow rate of the boil-off gas is high enough and the bypass valve is closed. During pump-down mass flow controller opens the by-pass valve.

4. Conclusion

Centrifugal compressors can either be simulated using the compressor map as a look-up table or using the Euler turbo-machinery equations. The compressor map of a centrifugal compressor can be crossed by different operation curves stretching from low pressure ratios to the maximum pressure ratio of the compressor. The operation curve with the lowest capacity of reduced flow rate lies on the surge curve.

On the other hand the highest capacity curve stretches through the choke zone. If one PI controller controls the suction pressure of a single compressor stage it has to react such as to set higher speeds, if the suction pressure is higher than the set pressure. Higher reduced speeds can be set by increasing the capacity value from the lower boundary X=0 towards the higher boundary X=1 at the measured pressure ratio.

However in a compressor string the compressors influence each other in a reciprocal way. Away from compressor design points and especially during transients the increase of the capacity value X from 0 to 1 does not always result in the desired collective system performance. This effect can be compensated using the saturation of the capacity value X as a trigger variable to generate a slightly higher total pressure ratio for the model-based control. In this way one single controller can be used to control the suction pressure of a compressor string with different number of compressors covering both transient and steady-state operation.

The plant engineered by LKT for FNAL uses this control logic. The application of the control logic based on capacity curves significantly reduced the pump-down duration. At steady-state conditions the controller was very robust and successfully reacted to disturbances to keep the suction pressure constant.

The control system is also tested with a cold compressor string including 4 cold compressors in series at a different plant. The model generation method is repeatable for different number of compressors. The X value can be increased very fast until saturation by using high gain values in the PI controller. Since a saturated X value on the high capacity curve increases the total pressure ratio sent to the model based control, the control system can react very fast and continuously increase the rotational speeds in the allowed region of the compressor map. In such a way the pump-down duration of experiments with large volumes can be significantly reduced.

The thorough formulation of these control algorithms and a diligent process control simulation helped to fine tune control variables and settings and to eliminate bugs prior to hardware commissioning. As a result the three cold compressors in series for the FNAL CMTF could be started up successfully and they reached the design suction pressure of 24 mbar within a short time of just a few days after pushing the start button.

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