Control Considerations of Multi Stage Cold Compression Systems in Large Helium Refrigeration Plants

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The Paper describes a control philosophy for multi stage cold compression systems operating in helium refrigeration systems at sub-atmospheric pressure down to a temperature of 1.8 °K. As an current example, the control philosophy and operational requirements of the four stage cold compression system of the CERN LHC plants are discussed.

INTRODUCTION

CERN, the European Laboratory for Particle Physics, is presently engaged in the construction of the Large Hadron Collider (LHC). This LHC will make use of high field super-conducting magnets operating in super-fluid helium and thus will require a huge refrigeration capacity at several temperature levels. The refrigeration capacity will be generated and distributed by eight refrigerator stations located along the colliders tunnel.

At the lowest temperature level of 1.8 °K, the total specified refrigeration capacity is 19.2 kW, thus 2.4 kW have to be generated by each refrigerator station. In order to reach this low temperature, a process vacuum interface is part of each refrigerator. The approximate process data of this interface are listed in table 1.

	Flow	Intake condition	Return condition
Steady State Modes:			
Nominal Load	126 g/s	14.5 ± 0.5 mbar , 4.0 °K	1.3 bar, < 20 °K
Part load operation down to	43 g/s	14.5 ± 0.5 mbar , 4.7 °K	1.3 bar, < 25 °K
Transient Modes:			
Connecting to the cold experiment	0	14.5– 1000 mbar	
Recovery from a limited quench	>20 g/s	14.5– 200 mbar, 4 – 25 °K	1.3 bar, < 30 °K

Table 1: Specification of the approximate process requirements for the CERN–LHC process vacuum systems. In the transient modes the process can be supported with a make up helium flow of 3 bar and 4.5 °K.

Considering the specified intake pressure and the mass flow, it becomes obvious that only a solution using cold turbo compressors is auspicious. The part load requirements, which will be discussed later in this paper, make further clear that the proposed combination of three to four serially connected cold turbo compressors boosting a warm vacuum screw is the best solution.

After a short overview of the process flow sheet the following pages describe the speed control logic of the four-stage cold compression system, which is supplied to CERN by the IHI-Linde consortium. The description of the final warm vacuum screw, which is also part of the supplied system, is not part of this paper.

NOMENCLATURE

M^* : Mass flow
P_i : Inlet Pressure to stage i
T_i : Inlet Temperature to stage i
N_i : Revolutions Speed of stage i
Π_i : Pressure Ratio stage i
$\Pi_{1\text{-}4}$: Total pressure ratio on cold compressors
x : normalized flow
m_i : Reduced flow $\left(m_i = \frac{M^*}{M^*_{ref}} \cdot \frac{P_{ref,i}}{P_i} \cdot \sqrt{\frac{T_i}{T_{ref,i}}} \right)$
n_i : Reduced speed $\left(n_i = \frac{N_i}{N_{ref,i}} \cdot \sqrt{\frac{T_{ref,i}}{T_i}} \right)$

Indices:
i : stage number
ref : reference values
(=values at design conditions)
S : steady state speed model
T : transient mode speed model

THE PROCESS FLOW SHEET

A simplified flow diagram of the vacuum compression system supplied by the IHI-Linde consortium is shown in figure 1. The approximate gas conditions at nominal load are indicated in the same diagram.

The main components discussed are the four cold turbo compressors. Each machine has its own frequency drive and the shaft of each machine is equipped with active magnetic bearings. The impeller diameters are 250, 165, 115 and 105 mm. As indicated in the figure 1, the total overall pressure ratio is at almost forty five remarkable.

The mixing chamber upstream of the intake of the first stage is especially used during transient modes:

- Before the cold compression system can be connected to the experiment, the suction pressure has to be equalized with the pressure of the experiments. If the pressure in the experiment is lower, the cold compressors have to be started while the connecting valve is still closed. In this case the matching flow conditions for the cold compressors are established by evaporating liquid helium with a heater and mixing the saturated vapor with a very small quantity of warm gas. The separator vessel is continuously refilled up to a constant liquid helium level.
- If the cold compression system has to be disconnected from the experiment, the same flow is established to perform a smooth stop of the system.
- After a limited quench in the magnets of the LHC, the helium temperature at the intake is increasing up to 35 °K. During this operation saturated vapor is added to keep the mixing temperature below 10 °K. This enables the cold compressors to keep running until normal conditions are recovered.

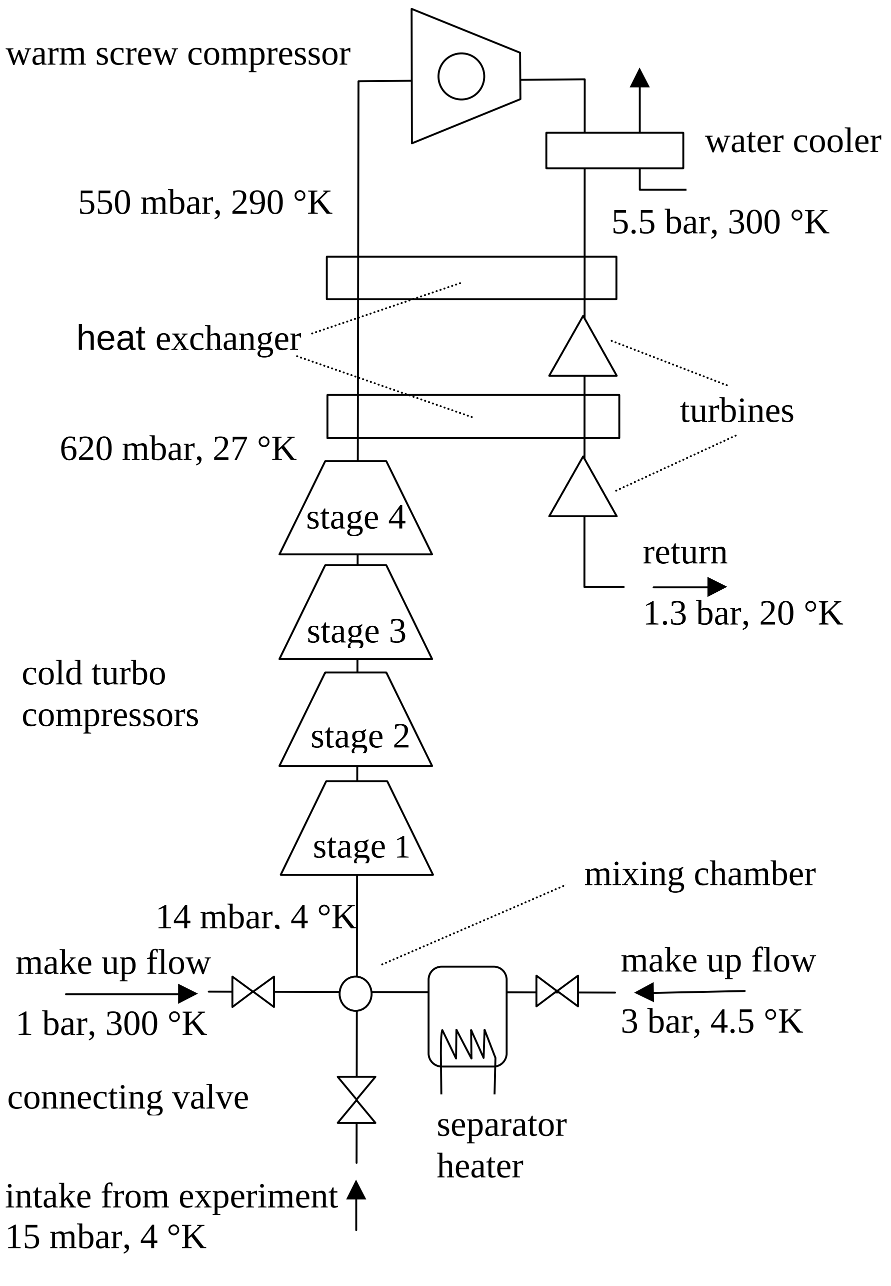

Fig 1: Rough process flow sheet of the LHC cold compressor system supplied by the IHI-Linde consortium

The suction and discharge pressures of the screw compressor as well as the pressure between the two turbines are floating. The return pressure is kept constant on customer side. Only the intake pressure has to be controlled by adjusting the speed of the cold compressors with the frequency drives.

THE WORKING FIELD OF TURBO COMPRESSORS

Unlike piston or screw compressors, turbo compressors are not volumetric type machines. Each turbo compressor has its individual working field indicating the correlation of rotating speed, flow, pressure ratio and suction conditions (Figure 2). Leaving the permitted field across the surge line will separate flow from the blades resulting in a sudden drop of pressure ratio and attended by considerable rotor vibrations.

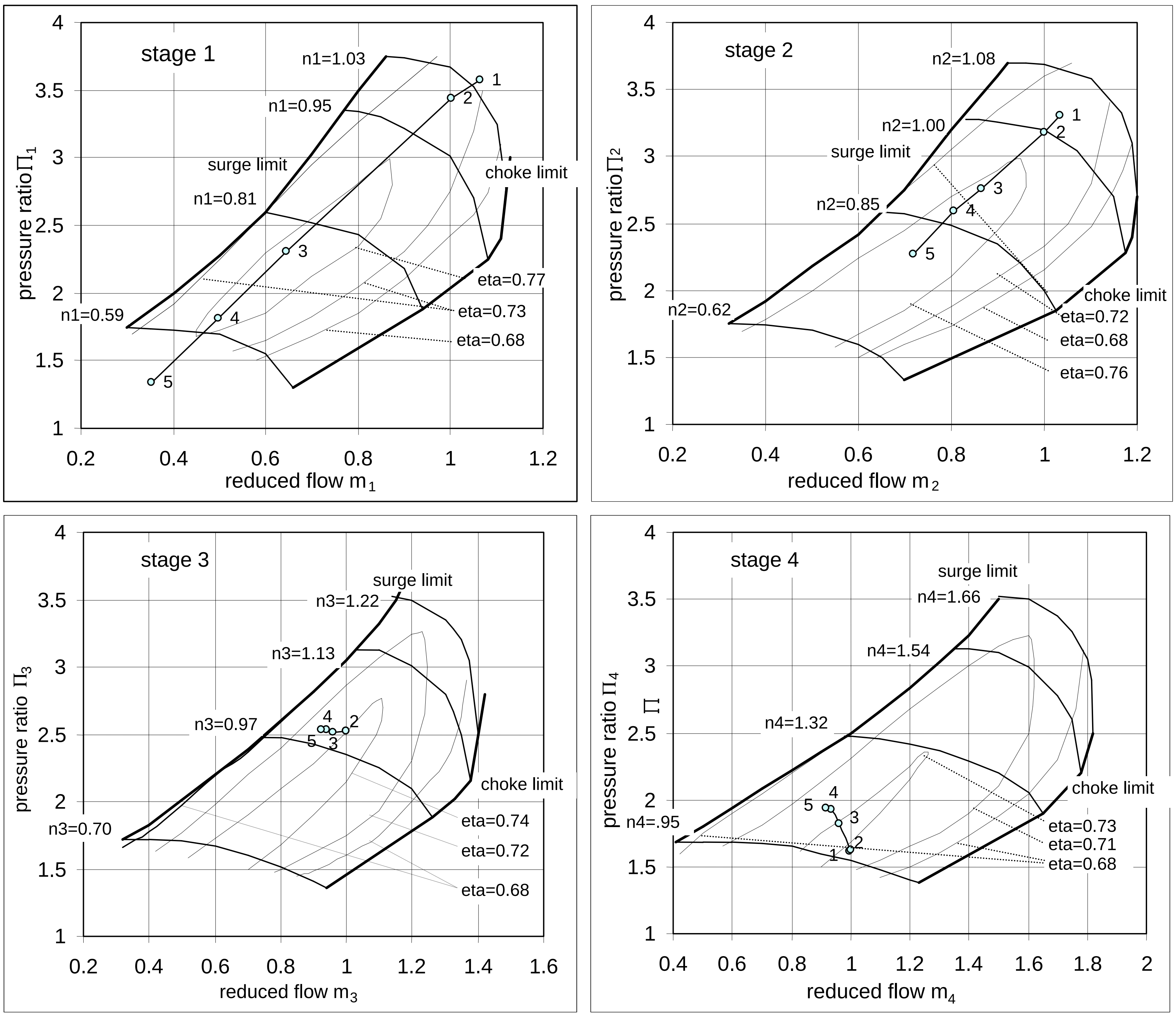

Fig 2: The individual working fields of the four cold turbo compressor stages.

Leaving the permissible field across the choke limit results in a loss of pressure ratio as well.

In multistage configurations, the different compressor stages are interacting. During startup it can happen that the first stage drops into surge flow while the last stage is choked. Multi stage, single shaft radial turbo compressors used for air service, at ambient temperature of course, solve this problem with blow off valves between the stages. Axial compressors used for air service on the other hand have adjustable stator blade rows, which considerably enlarge the permitted working field.

The cold compressor configuration at CERN has neither blow off valves nor adjustable stator blade rows. Instead, there are different measures, which can be taken in order to enable a safe operation of the cold compressor stages within their permissible working fields.

Firstly there is the warm screw compressor unit as the last link in the compression chain, which is matching well to the requirements of the cold compressor within the working fields. Screw compressors have an almost constant volumetric suction flow. The helium inlet temperature to the screw compressor is always near to the cooling water temperature and therefore constant. The pressure between outlet of the final cold compressor stage and the inlet of the warm screw compressor must be almost proportional to the circulated mass flow. It can be easily shown, that the proportionality between mass flow and the discharge pressure of the cold compressors leads to a proportionality between the reduced flow to the first cold compressor stage and the overall pressure ratio of the cold compressors. The proportionality factor is only depending on the inlet temperature, but not on the pressure of the first cold compressor stage.

Secondly there is the possibility of a makeup flow to adjust temperature. Of course, using make up gas is dissipating energy and a permanent use of make up gas is therefore not allowed. The make up gas is used to perform a minimal flow during startup before connecting the compressors to the experiment and after a limited quench to generate a suction temperature of approximately 8 °K.

As the last, there are the variable frequency drives of each single stage, which together with a intelligent control model, care for optimal performance.

MODELS OF SPEED CONTROL

The IHI-Linde consortium decided to develop and incorporate two different structural models for the speed control. One of them performs minimal control fluctuations and the highest adiabatic efficiency in all the steady state conditions. The other one performs the largest overall working field. It facilitates the operation in a safe distance from the surge and the choke limit during transient modes. The PLC-program is switching automatically between the two models.

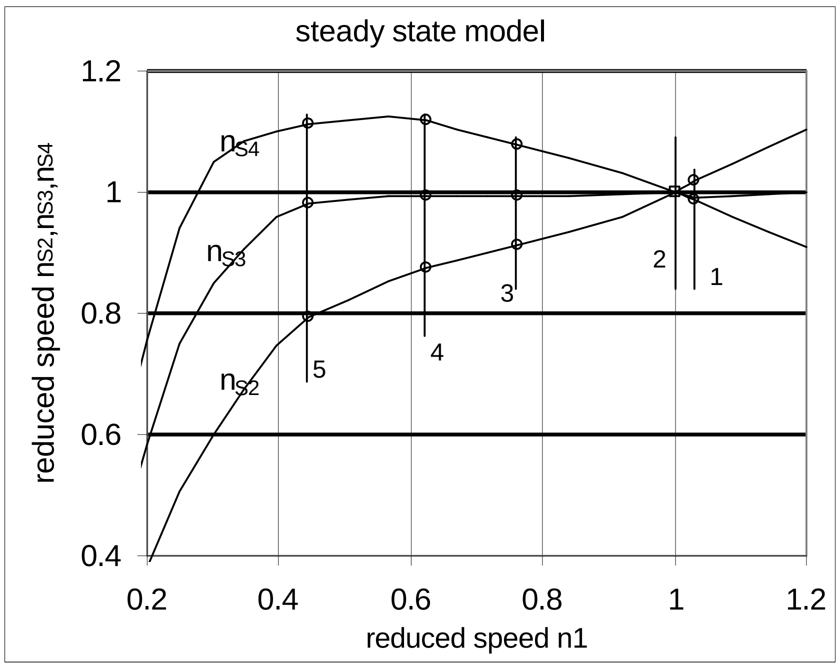

Fig 3: The reduced speed model for steady state operations. The five discrete steady state operating points from figure 2 are indicated.

For the steady state model, reduced speed values n_i have been optimized for five discrete cases in range of the steady conditions from the series specified in table 1. These reduced speed values are completed by interpolation to continuous functions $n_{S2}(n_1)$, $n_{S3}(n_1)$ and $n_{S4}(n_1)$, which are shown in figure 3. There is one free parameter n_1, which is set as an output of a PI-controller performing the required intake pressure.

The speed functions n_{Ti} of the transient model use two independent parameters, the measured overall

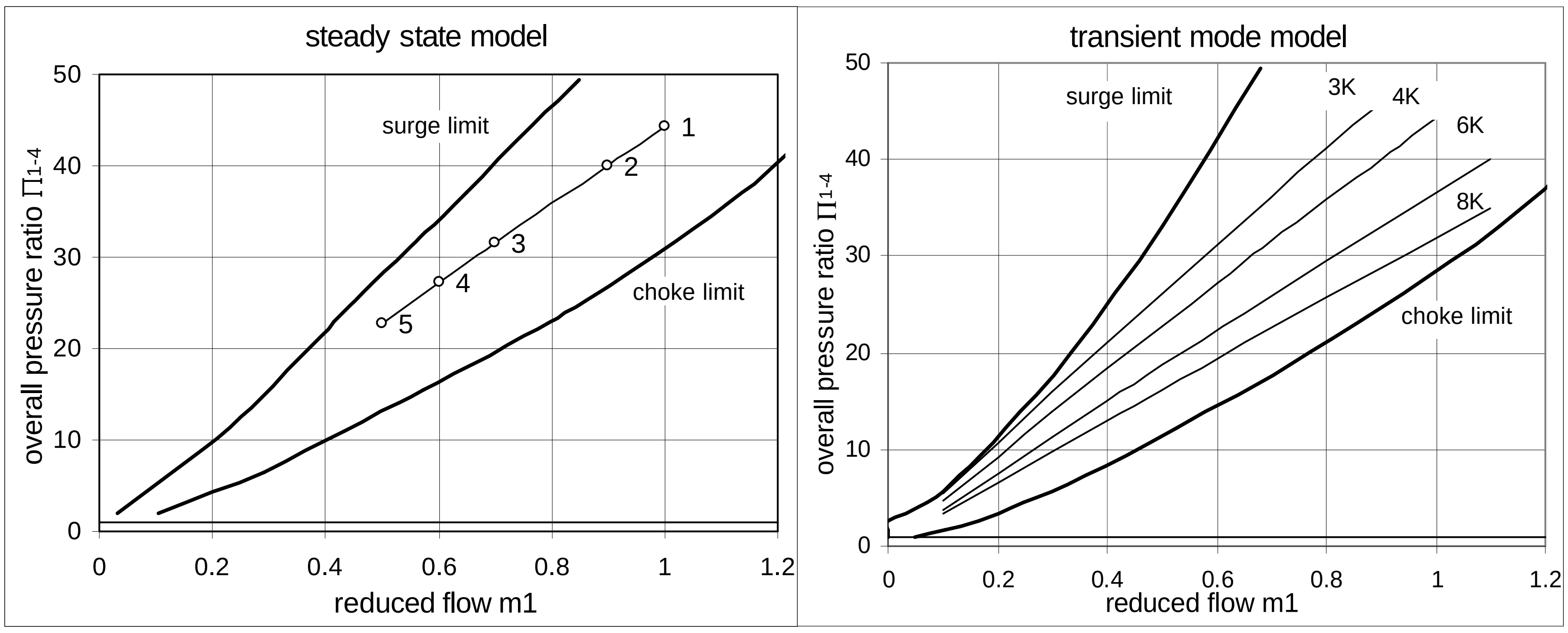

Fig 4: The overall working fields for both of the speed control models. The transient mode model performs a higher range between surge and choke. The five discrete steady state operating points from figure 2 are indicated in the steady state diagram. Lines of constant intake temperature are indicated in the transient mode diagram.

pressure ratio over the four stages (Π_{1-4}) and the normalized flow x, which is described later in this section. While the steady state model with only one parameter performs optimal results only along a given line, the transient model enables optimization for any points in the overall working field. By the same reason, as figure 4 shows, the permitted overall working field in the transient mode model is considerably larger than in the steady state model.

The five discrete steady state points theoretically use the same set of reduced speed values in both systems. The advantage of running these modes in the steady model is the fact that the normal control deviations in the one-parameter system are smaller and consequently the longtime adiabatic efficiency is slightly higher.

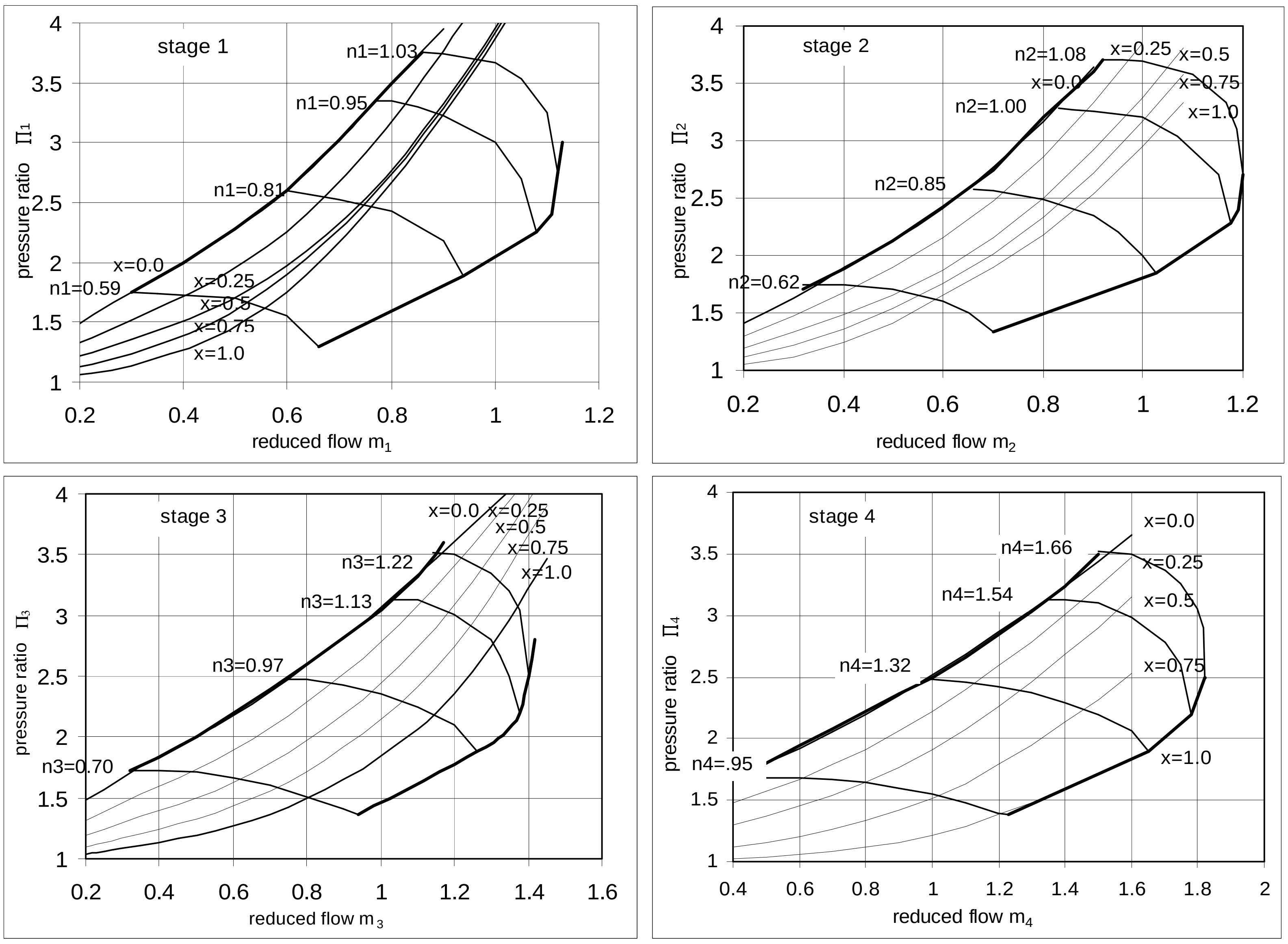

Fig 5: Lines of a constant normalized flow value x in the four stages.

By definition, the set of reduced speed functions n_{Ti} ($\Pi_{1-4,}x=0$) performs a given overall pressure ratio Π_{1-4} with the lowest possible reduced flow m_1. The set of reduced speed functions n_{Ti} ($\Pi_{1-4,}x=1$) performs the same pressure ratio Π_{1-4} with highest possible reduced flow m_1. In between, for x values between zero and one ($0 < x < 1$), the reduced flow m_1 is linearly increasing with x. By varying the normalized flow x, the pressure ratio on the single stages may change, but the reduced speed values correlate in a way, which keeps their product, the overall pressure ratio, constant.

Using this definition the reduced speed functions n_{Ti} ($\Pi_{1-4,}x$) are not yet uniquely defined. As a first estimate, the lines of a constant normalized flow x in the single stage working field were supposed to split the range of reduced mass flow between the limits of surge and choke in proportions of x to 1-x. Later, by computing a huge number of operating points, it was recognized that this estimate is almost correct for the last stage but not for the preceding stages. The result of this optimization is indicated in figure 5. Applying the lines in figure 5, the reduced speed functions are exactly defined. They are indicated in figure 6.

The control program simply interpolates between the discrete operating points of these diagrams using Π_{1-4} the currently measured overall pressure ratio and adjusting x by a PI-controller performing the required intake pressure.

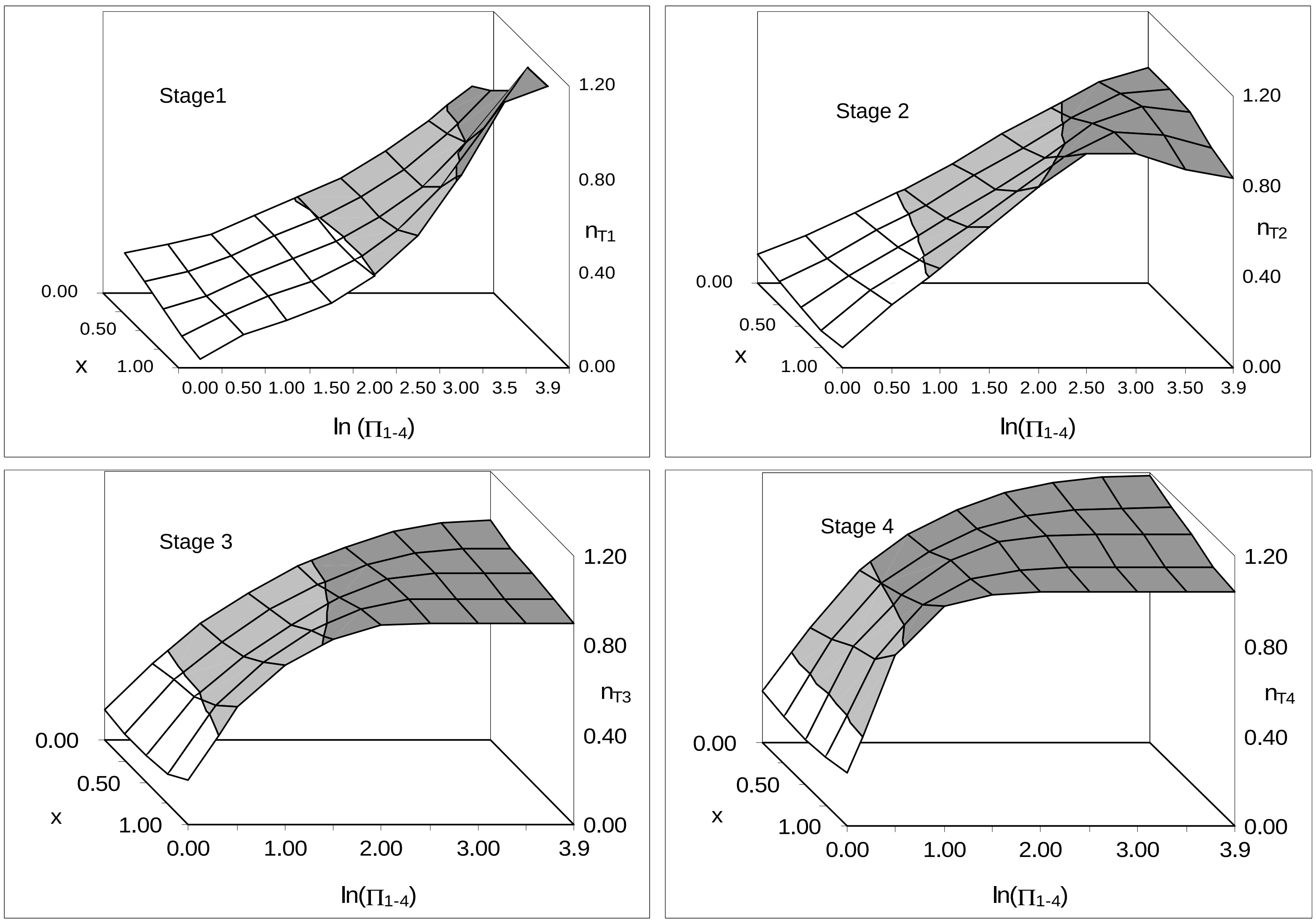

Fig 6: The reduced speed values for transient mode model. The reduced speed values are functions of the measured overall pressure ratio Π_{1-4} and the flow control value x.

EXPERIENCE

A pre-series type of the cold compression system is in operation at CERN since end of March 2002. It is connected to a test cryostat, which delivers the expected helium operating conditions from the LHC. Both described speed control models are implemented into a Siemens S7-300 PCS. The steady state model is applied when the connecting valve is opened, the mass flow is more than 45 g/s, the suction pressure is less than 20 mbar and the suction temperature is between 3.7K and 4.5 K. The transient model is used otherwise.

The predicted operating points of the single stages in the two different models have been almost exactly reproduced. After three days of parameter tuning the system was able to pump down the test cryostat automatically and to keep the intake pressure constant at 14.5 ± 0.5 mbar at various flow conditions.

The automatic control actions are efficient and considerably faster than they could be performed by a manual control. However, the enormous sensitivity of the cold compression system against small perturbations like the normal movements of control valves was not expected. These perturbations can destabilize the compressing chain and push it into surge flow. The state is manifested first by temperature fluctuations, followed then by pressure fluctuations. In this case a new start is necessary.

CONCLUSIONS

New models for speed control of multistage turbo compressor systems with independent frequency drives have been developed and were successfully tested at CERN in a four stage cold compressor application. The combination of two models of speed control, perform at highest efficiency in steady state operations and at the maximum permitted flow range during transient operation modes.

Further investigations and testing will be performed for automatic detection of beginning oscillations and in developing measures to smoothen the oscillations before the whole compression chain enters into surge flow.