

# DESIGN, CONTROL AND STARTUP FEATURES OF THREE PARALLEL-WORKING PROPANE COMPRESSORS EACH HAVING THREE STAGE GROUPS

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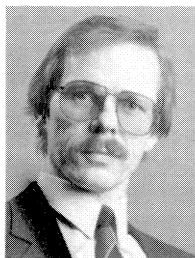
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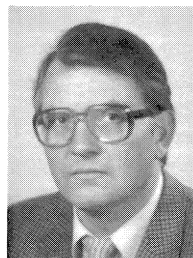
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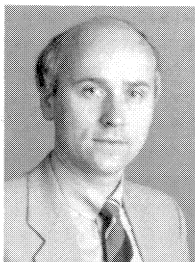
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## ABSTRACT

Two main problems must be solved in the planning and execution of compressor sets to be installed within refrigeration loops. The use of several compressors with different sidestream admissions calls for the installation of a control system for optimizing the parallel operation of the machines, even if there are differences in their performance curves, due to reactions to the pipework, minor manufacturing tolerances, or the use of varying frame sizes. To ensure uniform loading of the stage groups and the same distance from their respective surge limits in the case of constant discharge pressures, throttle valves are installed in the compressor intake pipes for actuation by the control system presented.

Protection from surging is provided by an anti-surge control system, which includes a dynamic control line to respond to sudden changes in the duty point, as well as an additional anti-surge safety line preceding the surge limit.

Unlike in other turbomachinery applications, the startup operation must be carefully studied early in the project planning stage of the refrigeration loop. In view of the great number of parameters affecting the starting procedure, these studies must resort to mathematical simulation for variation of the plant parameters, with the aim of minimizing the driving torque and, thus, the motor frame size.

One way in which these problems can be solved will be detailed by reference to a compressor set for a propane loop,

consisting of three machines each having three stage groups and two sidestream admissions.

## INTRODUCTION

Propane compressors belong to the group of refrigeration compressors which are characterized by several peculiarities in their design features, as well as in the startup and control behavior, as compared to other compressor types used in the petrochemical industry. The low temperature of the gas to be handled, which can be between  $-100^{\circ}\text{C}$  and  $-20^{\circ}\text{C}$ , requires that the materials used be tough at subzero temperatures, with an adequate notch toughness during impact testing.

Real gas behavior must be considered when rating the compressor, which is particularly evident for the various refrigerant and standard suction conditions with variables of state near the saturation line. Only in this way can the power consumption as well as the discharge pressures and temperatures be reliably predicted.

Parallel operation of compressors with several stage groups requires a fairly elaborate machine control system in order to ensure uniform load sharing among the compressors involved. Load sharing in this connection means operating the compressors at similar duty points in respect to surge line distance and efficiency. As opposed to other applications, the startup response of refrigeration compressors is a much more complex technical problem for the design engineer. Due to the closed-loop configuration, there is a great variety of factors which affect the torque required at the compressor coupling.

These plant parameters must be optimized to permit the best electric motor frame size to be selected under cost-effectiveness aspects. The following topics are detailed by reference to a specific application:

- Design features of refrigeration compressors
- Solution to the problem of controlling the parallel operation of refrigeration compressors having several stage groups
- Precalculation of the startup response of motor-driven refrigeration compressors

## DESIGN FEATURES OF PROPANE COMPRESSORS

A flow sheet of the three compressors working in parallel operation is illustrated in Figure 1, while a sectional view of a propane compressor is shown in Figure 2. Each of the three compressors has three stage groups. The first stage group consists of three impellers, the second group of two impellers, and the third group again of three impellers. Upstream admissions which are larger than the mass flow in the preceding stage group.

The first stage of each compressor is equipped with an impeller having three-dimensional twisted blades (Figure 3). The hub disc, including the blades, was milled from a blank on a numerically controlled (NC) machine, while the cover disc was welded on top of the blades. The blades extend into the intake eye a considerable distance, which reduces the inflow

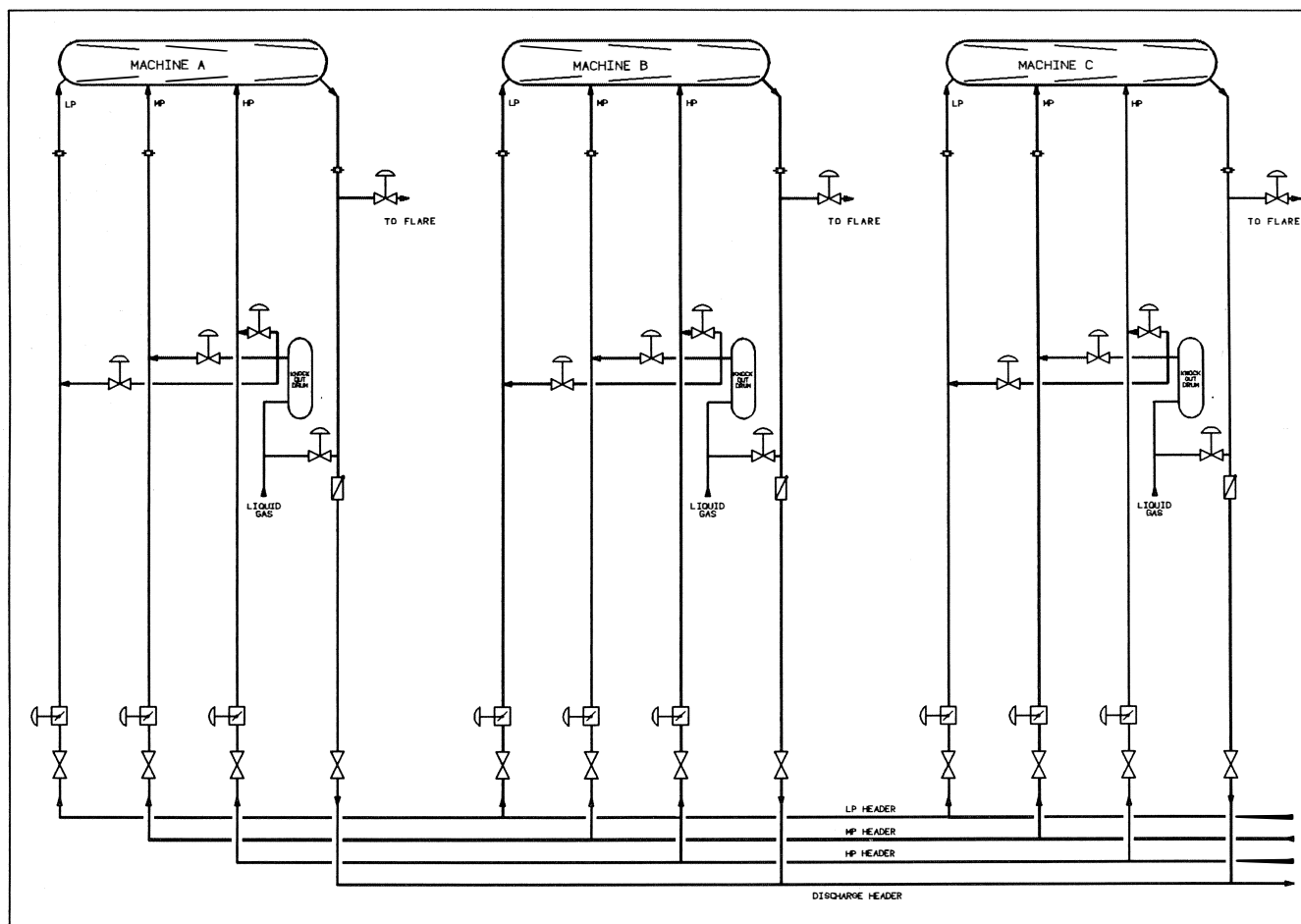


Figure 1. Block Diagram of Compressors Working in Parallel.

Mach number at the blade inlet. Accordingly, there is a wide overload range within the performance graph, before the characteristic curve assumes a vertical course. The vertical part of the performance curve indicates blocking of the impeller at the narrowest cross-section, upon reaching of the sound velocity.

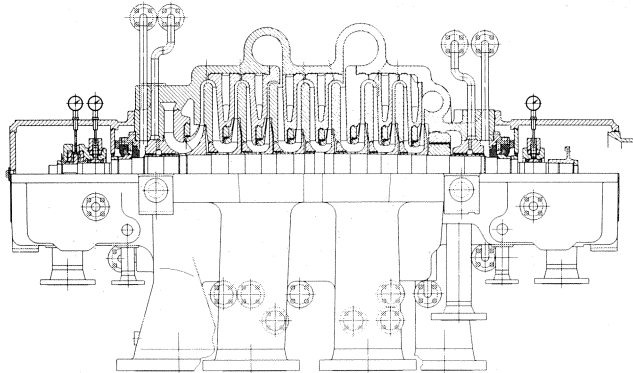


Figure 2. Sectional View of Refrigeration Compressor.

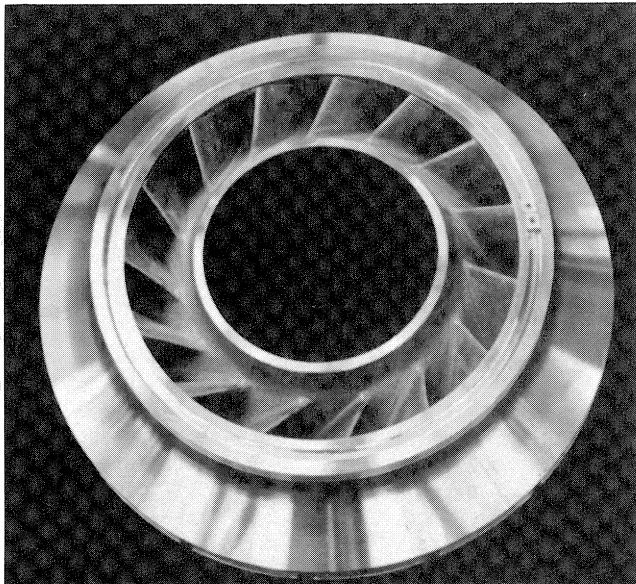


Figure 3. First-Stage Impeller with Three-Dimensional Twisted Rotor Blades.

The remaining impellers are fitted with non-twisted two-dimensional blades. Conditioned by the amount of energy admitted in the first impeller and the related temperature rise, the Mach numbers at the following impeller inlets generally decrease, so that there is no need for three-dimensional twisted blades at these locations. The blades of these impellers were also milled from blanks, with the cover disc being brazed to the blades. This approach is an obvious solution in the case of narrow impeller flow ducts or small impeller outlet widths, since there may be difficulties involved in inserting the welding electrode into the flow channels. Brazing of the impellers is affected under vacuum conditions, using a gold-nickel compound.

The radial and axial bearings supporting the rotors are tilting-pad bearings characterized by a high running stability. The axial bearing has controlled lubrication for minimizing the

bearing losses. A view of an uncovered machine with the rotor being placed in position is shown in Figure 4.

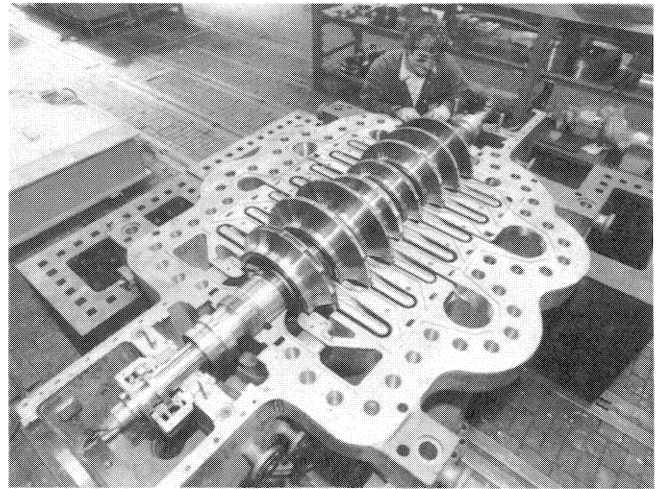


Figure 4. Compressor Being Assembled.

Sealing of the shafts toward the atmosphere is affected by a combined floating-ring/mechanical contact seal. The mechanical seal component ensures hermetical isolation of the compressor interior even under standstill conditions without requiring the feed of sealing oil. Oil discharge to the atmosphere with the machine in operation is taken care of by a floating ring. The combined lube and seal oil consoles were manufactured according to American Petroleum Institute (API) 614. In consideration of the ambient conditions at the plant site, the oil is cooled in air coolers.

The cast compressor casings are made of low-temperature-resistant steel with a high toughness at subzero temperatures. The casting pattern was a combination of several standard modules available from the process-gas compressor range (Figure 5). The pattern consists of a standard inlet and outlet casing, two sidestream admission casing parts, and the cylindrical shell sections. This modular type of construction ensures minimum pattern fabrication times, which also means shorter delivery periods.

The compressors were subjected to a mechanical running test at the supplier's shops in conformity with API 617. Performance testing was carried out under American Society of Mechanical Engineers (ASME) PTC 10 regulations in a closed cycle using a refrigerant as the flow medium. All three stage groups were tested at the same time. An electric motor with a gear train was used for driving the compressors.

Before being shipped, the gear train, plus compressor assembly, including piping and instruments, was installed on a base frame at the supplier's works. An illustration showing the compressor packages being assembled is presented in Figure 6. The electric motor was added to the base frame at the plant site.

## PECULIARITIES OF PARALLEL-WORKING COMPRESSORS

Due to the fixed speed of the electric driving motors, the compressors are controlled by throttle valves in the intake pipes of the three stage groups. The adjustment of these valves is a major problem in the control of parallel-working refrigeration compressors having several stage groups.

Modern production technologies and the close tolerances that can be achieved are justifying the assumption that the

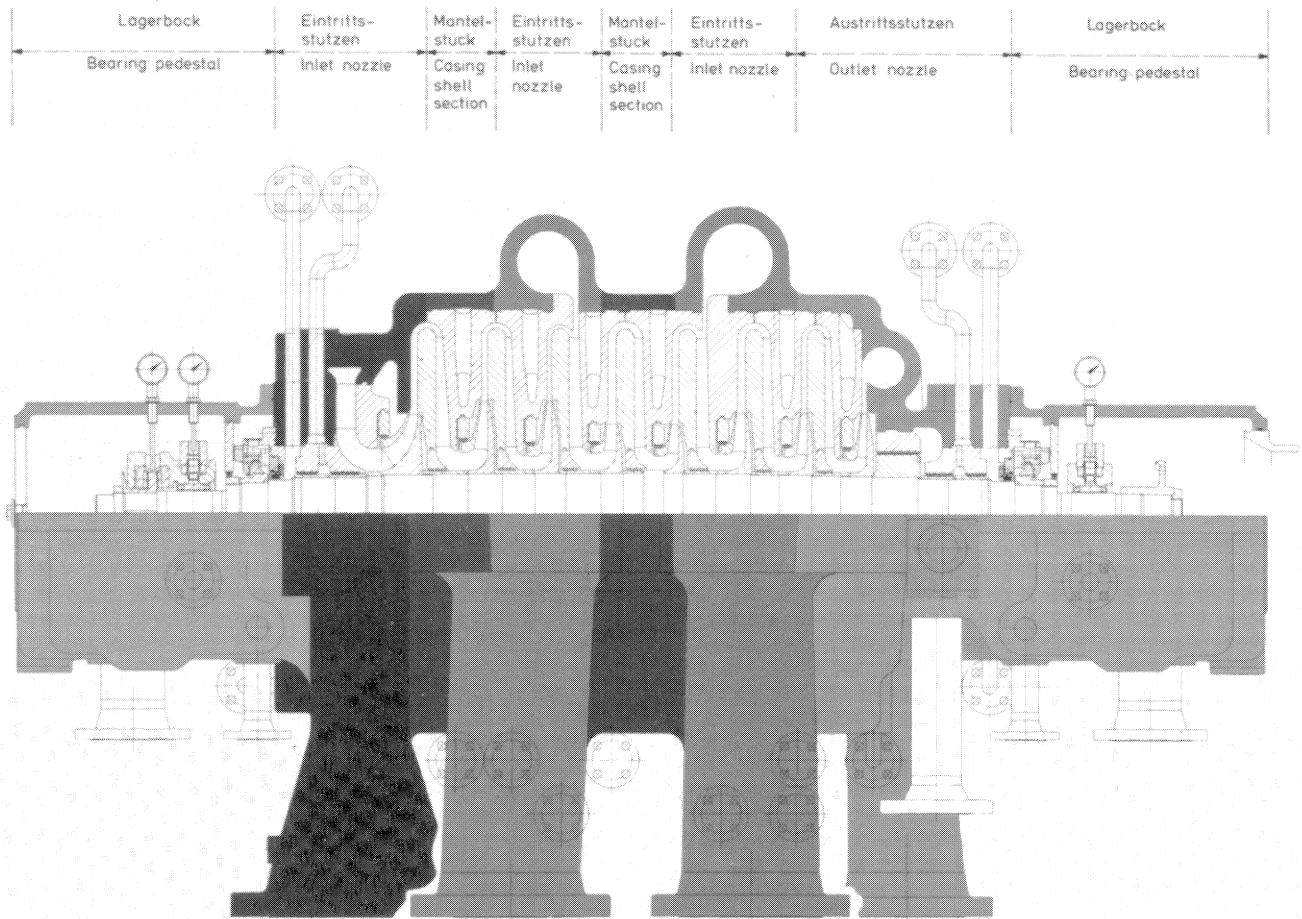


Figure 5. Compressor Casing Built of Standard Modules.



Figure 6. Refrigeration Compressors Being Assembled and Piped on Their Base Frames at the Supplier's Works.

dimensions of the flow channels in the four different rotors (i.e., three main and one spare rotor) as well as the dimensions of the casing internals are almost the same for all three machines. However, even minor deviations in pipe routing and flow channel dimensions will cause an alteration of the compressor performance curve. The most important application of this type of control is the parallel operation of compressors with different frame sizes or even of different design. The effects that can result from this for the parallel operation of two compressors, A and B, are illustrated in Figure 7 which shows the pressure ratio  $P_1/P_2$  as a function of the suction flowrate,  $\dot{V}$ .

Small variations in the curve configuration already lead to major differences in the flowrate. As the performance curve becomes flatter, the flowrate is affected more by the curve variations.

Variations in the performance curves often mean different surge limits. Therefore, each individual compressor must have its own anti-surge control and, thus, also the associated recycle piping. In compressors with sidestream admissions between the stage groups, it is even necessary to provide a separate anti-surge control for each individual group.

In commonly used load sharing controls [1], the flow is measured at the inlet to each compressor, and on this basis, the position of a throttle valve or the machine speed is modified in such a way that all compressors are operated at the same flowrate. The characteristic curves of Figure 7 clearly show that the duty point of machine A is already near the surge limit,

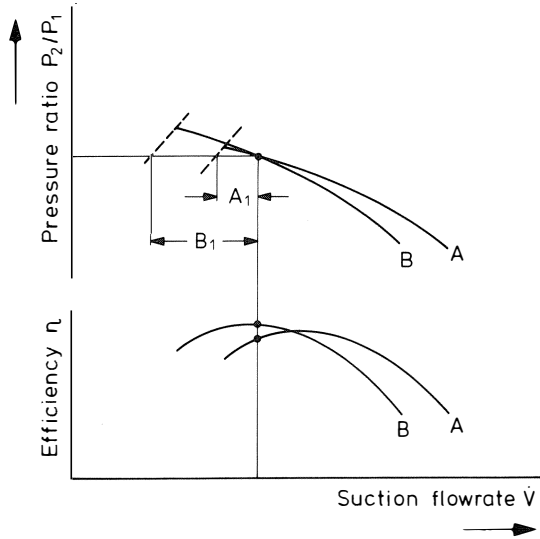


Figure 7. Schematic Presentation of Effect of Different Performance Curves of Compressors Working in Parallel on the Delivery Volume at a Constant Pressure Ratio.

whereas machine B is still far away from this line. This means that machine A is already recycling the flow medium even though it would still be possible to share the full flowrate between the machines without such energy loss, presupposing adequate load sharing control.

The requirement to be satisfied by an improved load sharing control assembly is that all machines must be operated at duty points which have the same distance from the respective surge limit. This ensures optimum utilization of all operating ranges, while also permitting all of the compressors to be operated at their optimum efficiency at the same time. A typical example of this is illustrated in Figure 8.

The load sharing control system (LOSHCO) developed for this purpose will now be described in greater detail. Patent protection has been sought for this control unit in several countries.

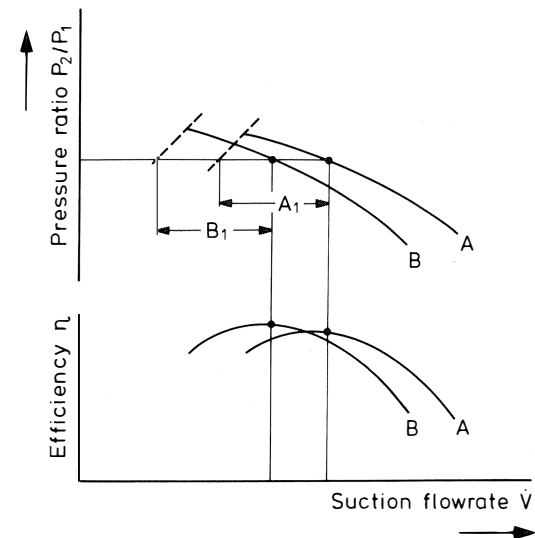


Figure 8. Standard Load Sharing with Control (LOSHCO).

## CONTROL SYSTEM

### Tasks

The control system of this plant, which is to be considered a combination of 9 compressors, must fulfill the following tasks:

1. Each compressor stage must be protected from surging.
2. The pressures at the intakes are to be controlled to a preset constant level.
3. Compressor stages operating in parallel must be controlled to handle the same load.
4. The machine must be capable of being operated in the following way:
  - One machine only;
  - Two optional machines in parallel;
  - All three machines in parallel.
5. Upon acceleration of the machine set, the refrigeration loop is to be started such that
  - all stages operate at a duty point which corresponds to their design pressure ratio, independent of the respective suction pressure;
  - the pressure level at the third discharge is increased to the pressure in the process system and the machine coupled to the process;
  - finally, the sidestream pressures are adapted to the pressure in the headers and the isolating valves are opened as required.
6. Since there is no cooling water available for cooling of the recycle gas flow, the pressure in the recycle loop must be reduced to a level permitting liquid refrigerant to be injected for cooling purposes.

A control circuit diagram of this plant is shown in Figure 9.

From the intake pipes, the gas first flows to the throttle valves of the respective stage groups. These valves are adjusted by a command from the pressure controller (PC) and the load sharing control (LC). The input signal from LC (i.e., the distance between the duty point and the control line) is formed in a logic circuit from the input variables of the anti-surge control unit (AC).

These variables comprise the measured suction flowrate, as well as the suction and discharge pressures of the respective stage. The flowrate is determined via orifice measurement in the compressor inlet piping, by ascertaining the pressure difference across the orifice, as well as the pressure and temperature at the compressor inlet. The flowrate of the second stage group is ascertained by adding the suction flowrate to the first sidestream admission flowrate. Finally, the flowrate of the third stage group is determined not in the suction piping, but rather in the discharge piping. In the event that the duty point is located outside the stable working range of the stage group concerned, a gas partial flow from the discharge piping is fed through a pressure-reducing valve to a drum, where it is mixed with evaporating propane used for cooling purposes. The reducing valve receives its signal from a summing unit which, in turn, is fed with input commands from all three anti-surge controls. Various recycle valves are arranged downstream of the drum for each stage group. The valve position is governed by the anti-surge control of the stage group concerned.

The instrument panels with all necessary control facilities and the other machine protection equipment is featured in Figure 10, while a view inside of an opened panel is depicted in Figure 11.

The following section will describe the way in which the aforementioned tasks were solved.



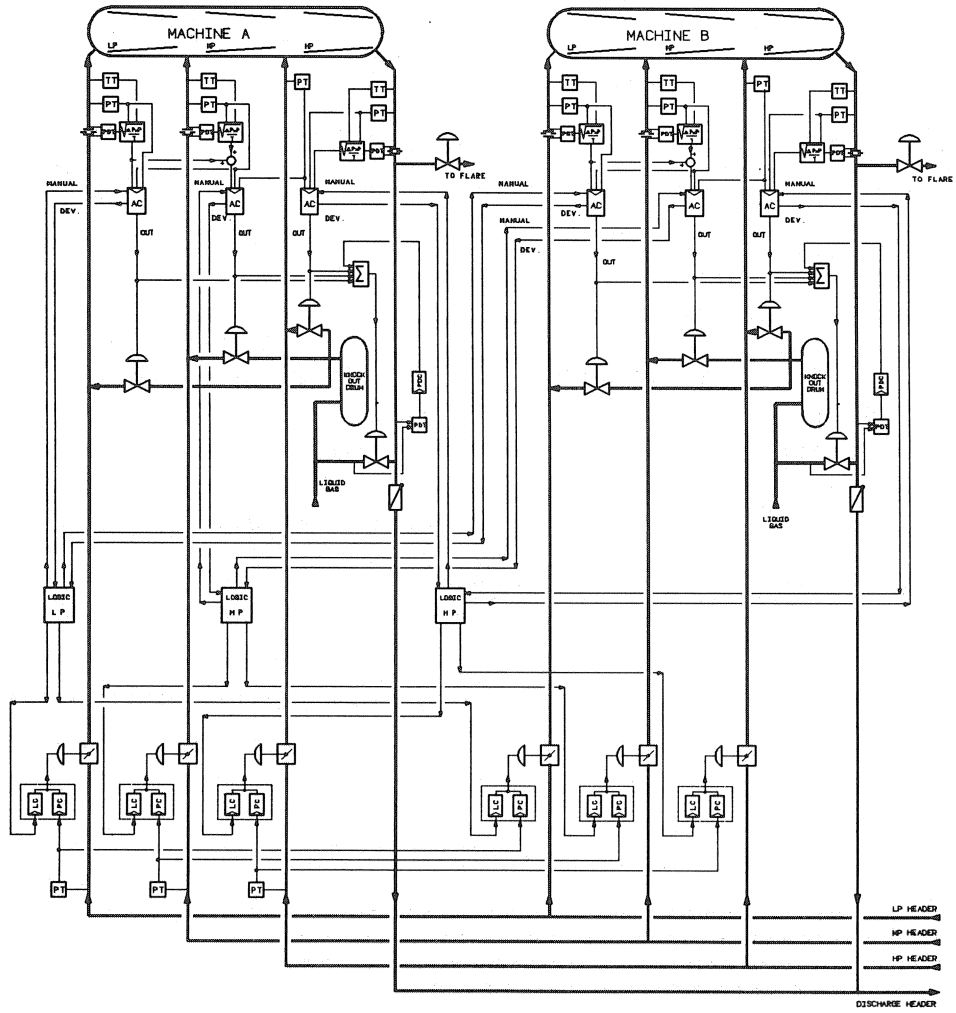


Figure 9. Process and Control Flow Sheet of Three Compressors Working in Parallel.

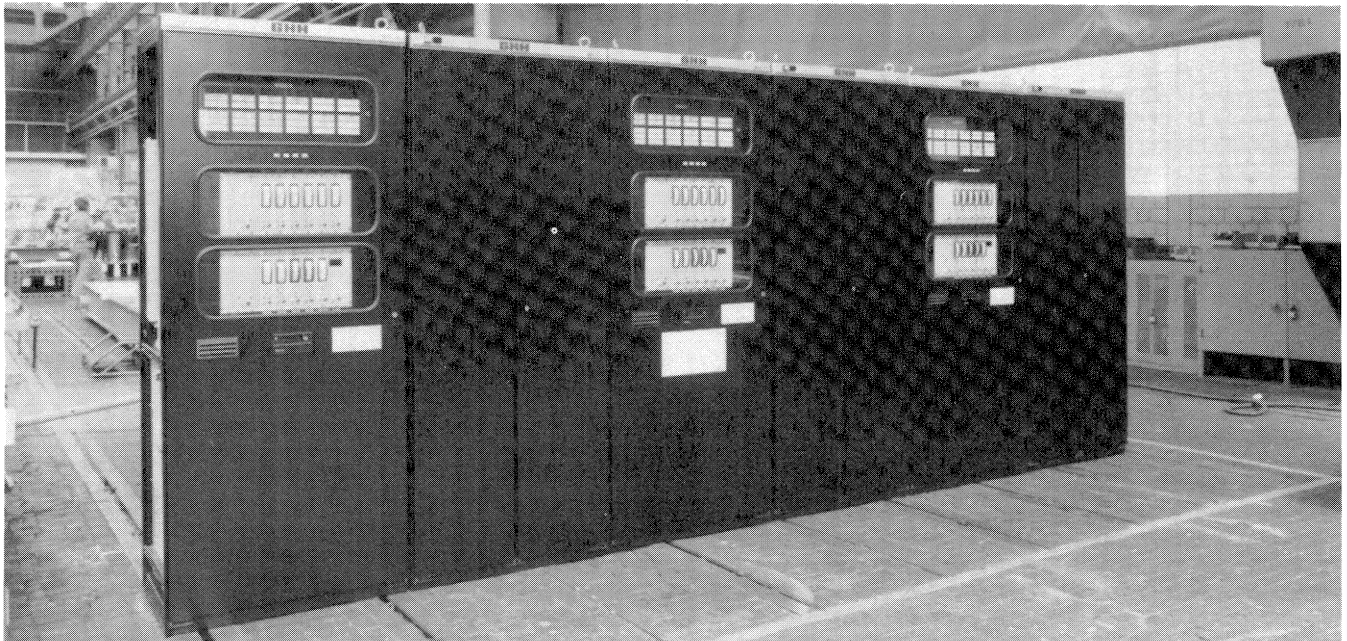


Figure 10. Control Panels for Monitoring and Controlling Functions (Temperature, Axial Shaft Position, Vibration Amplitudes), Including Related Alarms and Trips.

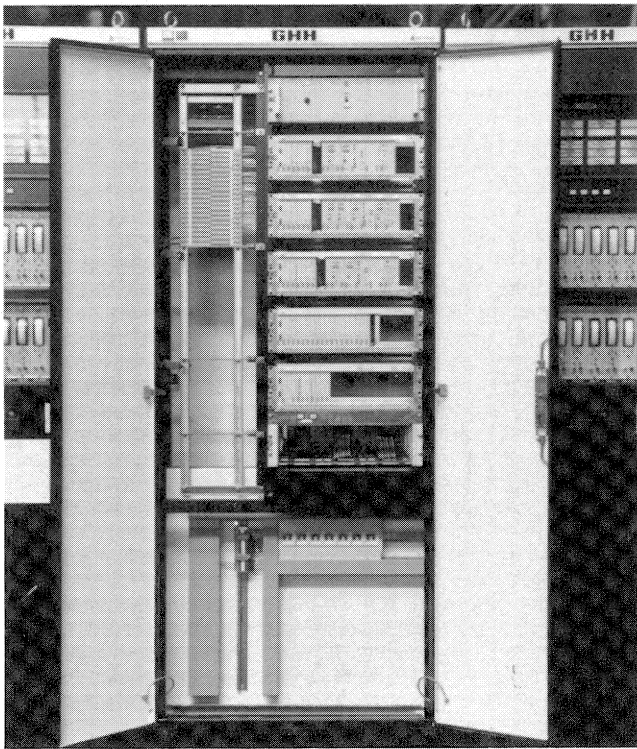


Figure 11. View into Control Panel.

#### Anti-surge control

Each compressor stage group has its own anti-surge control system (AC) of the Turbogol type [2], with the following special features:

*Dynamic control line (patented).* With a duty point far inside the performance graph, if a fast-acting valve closes on the compressor discharge side, any standard control system would not be activated until the control line is exceeded. This means a loss of valuable time before the machine protective system becomes effective. Often the control system cannot prevent the surging of the compressor within the short time remaining, until the surge line is reached. It is therefore desirable to increase the safety distance from the surge line when there is a quick change in duty points. The control line is made "dynamic" through a specific functional element. The safety distance from the surge line is varied as a function of the rate of change of the distance between the duty point and the control line. This provides much better protection for the machine in critical situations, with a minimum safety distance under steady-state conditions.

*Non-linear amplifier (patented).* When the control line is exceeded, a non-linear amplifier is made operative to increase the control deviation, simulating considerably wider duty point overshoot. This causes the controller to intervene much more vigorously than it would without this additional amplifier. The measures described have improved the dynamic response of the blow-off control system to such an extent that the control line normally is exceeded by no more than five percent of the current throughput, no matter which type of disturbance is concerned.

*Anti-surge safety line (patented).* If the control line still is exceeded by more than five percent despite the above precautionary measures (caused, for instance, through an incorrectly set control response time), there is yet another safety facility to be made operative: the control deviation is

monitored by a threshold stage. When this module responds, the control or recycle valve is opened instantaneously through direct effect on the control-power loop (e.g., via a solenoid valve), bypassing the controller.

*Vertical and horizontal control line (for pressure limitation) (patented).* This enables the anti-surge control to function at the same time as a safeguard against rotating stall and excessive pressure to protect the downstream pipework.

*Computer calculated pressure ratio.*

*Computer calculated flowrate with pressure and temperature compensation.*

*Computer converted mass flowrate to volumetric flowrate.*

*Temperature compensation of the control line.* This is especially important in an atmosphere with a wide range of temperature variations.

*Fully automatic startup device.*

*Soft changeover from manual to automatic operation.*

Precise operation of the anti-surge control system positively ensures that only so much gas is bypassed as is absolutely required to protect the compressor. The actual value for anti-surge control, i.e., the suction flowrate, can be directly measured only for the first stage, while the flowrate for the second stage must be determined by adding the first stage mass flowrate and the first sidestream admission flowrate. The Turbogol system calculates the suction flowrate as an actual value through conversion to inlet conditions.

Compression adds heat to the gas. This heat must be withdrawn from the system when in recycle operation. Since there is no cooling water available, heat withdrawal from the flow medium is affected by the injection of liquid flow media. The required pressure drop between the liquid-gas tank and the machine is generated by a reducing valve connected in series with the recycle valves. Therefore, there is no need for installing an additional pump. Being actuated in parallel to all three recycle valves, the reducing valve generates an almost constant pressure drop at all duty points. Precise adjustment is affected by means of a superimposed pressure differential controller (PDC).

#### Pressure control and load sharing control

The pressure in the suction piping is maintained by pressure controllers (PC), whose outputs actuate throttle valves, with one pressure controller being associated with each one of the nine valves. A logic circuit is provided to ensure that only one pressure controller per header is set to the automatic working mode, thus avoiding malfunction. In addition, the throttle valves are operated by the load sharing controller (LC) in such a way that the valves of two stages operated in parallel are adjusted in opposite directions, in order to maintain the same distance between the surge limit and the duty point in both stages.

- Balancing the adjustment of the compressor stages to the same distance from the control line means an optimal utilization of the machine performance, even if there is extensive scattering of the characteristic curves. In the case of load reduction, all compressor stages operated in parallel are bound to reach the control line at the same time.

- The load sharing controller is combined with the pressure controller. A special type of circuitry for both controlled variables has permitted the pressure and load sharing controllers to be assembled in one module. Thus, there is only one package control unit to process both variables. This eliminates the need for any tracking circuitry. Furthermore, any instabilities resulting from the interaction between the pressure and the load sharing controllers are excluded. When compared

to conventional load sharing controls, which formerly consisted of a cascade-type circuit for the pressure controller in conjunction with several flow controllers, this approach has the further merit that one of the controllers can be eliminated. In addition to cost savings, there is no need to consider the time response of the flow control loop in selecting the control parameters, which means that the pressure controller can be adjusted much more quickly.

The control functions are depicted in Figure 8, where the distance between the duty point and the surge limit has been illustrated by A1 and B1. If A1 exceeds B1, the load sharing controllers open the throttle valve of machine B, and close the valve of machine A by a corresponding amount. Since the throttle valves are operated in the linear range of their characteristic curves, this means the overall flowrate remains essentially constant. The pressure controller does not intervene during this load sharing action.

If the process requires a different operating pressure, only the pressure controller set in the automatic mode becomes operative (e.g., machine A) to adjust the related throttle valve. This necessarily causes distance A1 to become greater. The load sharing controllers readily take notice of this "asymmetry." Analogously to the above-described procedures, the controllers close throttle valve A slightly, while opening valve B. Because the pressure and load sharing controls are operated at the same time, the load sharing controller merely restricts the opening rate of valve A. Any countereffect between the controllers is prevented by a corresponding throttle valve design.

The pressure and load sharing controllers of the various stage groups operate independently of one another. It is possible for both the pressure and load sharing controllers to be independently switched to automatic or manual control, with the load sharing controllers always being actuated in pairs. Inadmissible combinations are prevented by a logic circuit, thus avoiding any operational problems for the overall process.

#### Startup Control

Before being started, the gas in the compressors is expanded to the ambient pressure with closed isolating valves toward the process. The recycle valves are fully opened in order to reduce the starting torque of the machine set. After the compressor train has been accelerated, process liquid is injected into the circuit, which causes the system pressure and the power consumption of the compressors to increase. To protect the driving motor from being overloaded, the recycle valves must be readjusted, after motor acceleration, in a way that each stage is operated near the design pressure ratio. This requirement is satisfied by applying recycle-valve closing pulses to a control input of each anti-surge controller, until the required pressure ratio has been achieved. The anti-surge controller is designed so that the control line is not exceeded at any given time.

### CALCULATING THE STARTUP PROCEDURE OF REFRIGERATION COMPRESSORS

One of the major variables in the rating of three-phase motors to be used as compressor drivers is the acceleration time, which supplies useful information about the thermal loading of the stator and rotor windings during the starting process. With complex start sequences, the heat stored in the motor during various starting attempts may reduce the time available for further starts, which, with larger units, means that plant operations could be impeded by unproportionally long cooling phases. It must be duly considered, in view of explosion hazard regulations, that the gas ignition temperature must

not be reached at any time or any motor location. The decision whether or not a specific acceleration time and the inherent motor heating should be accepted depends on several conditions which, in turn, result from the technical features of the motor design [3,4].

Precalculation of the startup procedure of a motor-gear train-compressor package is a relatively simple affair if there are constant compressor suction conditions, and if the resistance characteristic of the consumer follows a parabolic curve, such as in the case of air supply systems. The curves for the motor torque,  $M_M$ , and compressor torque,  $M_K$ , as a function of the speed,  $n$ , for a typical application are illustrated in Figure 12. Using the curve patterns of the motor and compressor torques, the startup time,  $t_A$ , for a shaft system with a mass moment of inertia of  $\theta$  to speed  $n_N$  can be determined as follows:

$$t_A = \frac{\pi}{30} \theta \int_0^{n_N} \frac{dn}{M_M - M_K}$$

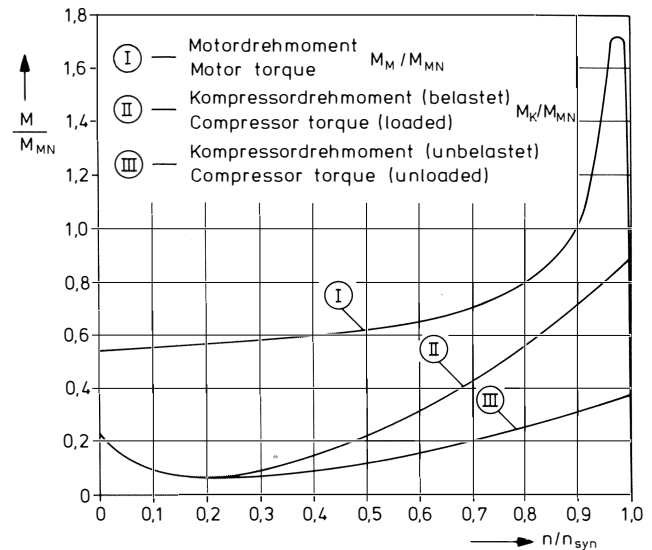


Figure 12. Motor and Compressor Torque Curves Relative to Speed ( $M_M$ : Motor Torque;  $M_{MN}$ : Motor Rated Torque;  $M_K$ : Compressor Torque).

The greater the difference between the motor torque and the compressor torque, the higher the acceleration of the shaft system and the shorter the acceleration time. However, a high motor torque requires a big motor, which means a high capital outlay and large starting electrical currents.

Matters are much more complicated for compressors in refrigeration plants than with compressors operating with a constant suction pressure. The number of factors affecting the compressor torque curve is considerably greater in such a case. In a complex system of this kind, it is no longer possible by conventional means to define the compressor torque as a function of speed. To obtain specific data for rating the driver despite these difficulties, development work was started several years ago on a computer program [5], which has meanwhile been steadily expanded and repeatedly confirmed by comparison with measured results from operating machines sets.

The computer program permits the simulated operating response of a complete plant. This offers the great advantage of the possibility of determining torque curves and acceleration times for the compressor systems. The operating response of



the plant can also be assessed by reference to transient variable-of-state curves. The starting procedure of the complete plant can be adjusted in an optimum way to the operating response of the turbomachinery through the installation of corresponding circuit connections, or the variation of final control elements, and the specification of different starting data.

*Structure of the simulation program*

The adaptability of the simulation program to the loop to be examined was of the utmost importance for solving the tasks involved. Special programming and processing of the loop permitted optional circuits to be structured from individual components, such as compressors with different drivers, piping, vessels, coolers, nonreturn and other valves, and studied for determining the interaction under the various marginal conditions in question. The horizontal program structure (Figure 13) does not include any type of loop that would be permanently programmed. A description of the loop concerned is predetermined by a special indexing method. An illustration of the indexing mode chosen for the plant under consideration, based on the flow sheet shown in Figure 2 and the described startup control, is presented in Figure 14.

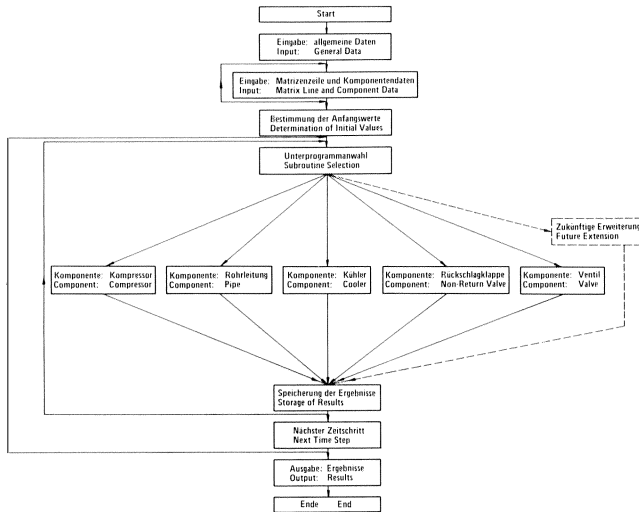


Figure 13. Matrix Structure of Calculation Program with Different Program Modules.

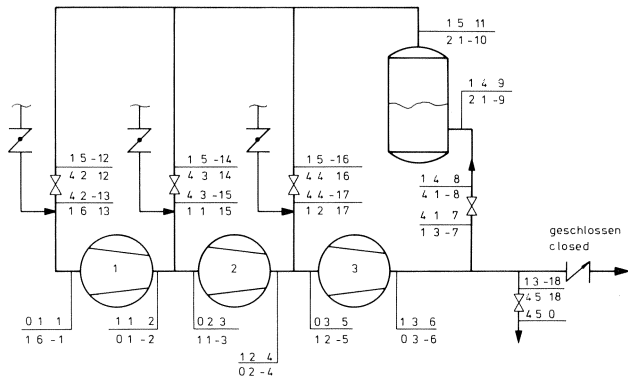


Figure 14. Indexing of Startup Schematic for Calculating the Compressor Torque.

*Mathematical Description of Loop Components*

*Compressor Operating Response*

The startup of turbomachines within a ramified system is governed by the performance graph characteristics of the machines. Conventional performance curve conversions, such as used in other simulation processes, are bound to fail if the basic requirement is that surge limit curves and efficiencies should be precisely represented for all operating ranges. The program discussed herein is designed for the processing of performance graphs, in a polynomial form, which were either measured or converted from actual data measurements [6].

For stationary operating conditions, there is an equilibrium of power between the driver and the compressor. If this equilibrium is disturbed, the resultant torque  $M_B$  causes a variation of speed.

$$M_B = \frac{\eta}{30} \theta \frac{dn}{dt}$$

The torque of acceleration,  $M_B$ , is the difference between the motor torque,  $M_M$ , and the torque from the gas forces,  $M_G$ , as well as the frictional losses in the gear train and bearings,  $M_R$ .

$$M_B = M_M - \sum_i (M_{G,i} + M_{R,i})$$

The equations cannot be solved explicitly, because of the varying dependencies of torques on speed, time and the compressor duty point, even at a great distance from the rating conditions. Calculations have to be performed in sufficiently short time steps, during which the variables of state may be assumed to be constant.

*Piping and Vessels*

The storage capacity of piping or vessels exerts a decisive effect on the configuration of the startup torque. It is at these locations where differing gas flows tend to be mixed. Any time-related displacement of the pressure curves has an effect on the duty points of the compressors and their power consumption.

The sum of incoming and outgoing mass flows is equal to the change in the stored mass,  $m$ ,

$$\frac{dm}{dt} = \sum_i \dot{m}_i$$

from which follows a pressure variation equation

$$\frac{dp}{dt} = \frac{R T_m}{V} \times \frac{dm}{dt}$$

where  $T_m$  is the energetic mixing temperature,  $R$  the gas constant, and  $V$  the storage volume.

*Coolers*

Considering the effect of heat-exchanging vessels during the starting process is a problematic affair. The cooler is mainly treated as a component with a passing throughflow, and the pressure loss, heat exchange, and corresponding temperature variations are duly calculated.

*Nonreturn Valves*

The simulated calculations permit consideration of non-driven nonreturn valves which are opened by the flow pressure,  $M_S$ , and closed by a counterweight,  $M_W$ , or by return flow effects. Any variation in the mass flow causes an adjusting torque to affect the rotating valve disc,

$$M_K = M_W - M_S$$

which is partly required for overcoming the hydraulic braking

device, with  $K$  as a damping factor, as well as for the disc rotating movement,  $\omega$ .

$$M_K = \theta_K \frac{d\omega}{dt} + K\omega$$

#### Valves

The mass flow of a globe or butterfly valve can be described by:

$$\dot{m}_V = \alpha A \psi \sqrt{\frac{2}{k-1} p_1 \rho_1}$$

where  $\alpha$  is the flow coefficient,  $A$  the cross-sectional area,  $k$  the isentropic exponent,  $p_1$  the inlet pressure, and  $\rho_1$  the density. Being a flow function,  $\psi$  depends on the type of gas and the pressure ratio, reaching a constant maximum at the critical pressure ratio:

$$\psi_{\max} = \sqrt{\left(\frac{2}{k+1}\right)^{\frac{2}{k-1}} - \left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}}$$

#### Physical Variables Affecting the Compressor Torque

As a rule, the startup procedure is calculated on the basis of standstill conditions at time  $t=0$ . The driving torque of the electric motor,  $M_M$ , is a function of the speed, as is illustrated in Figure 12, by  $M_M/M_{MN}$ , where  $M_{MN}$  is the motor rated torque. Such a curve pattern is characteristic of the selected motor and is specified by the motor supplier. The machine train is accelerated and passes through an area of bearing mixed friction in the lower speed range with increased frictional torques. The discharge and suction storage volumes (piping, heat exchangers, evaporators) are filled or emptied according to the respective compressor throughput, and different pressures are produced by the effect of pressure losses or throttle valves. Physical parameters influencing the compressor torque, and thus the required acceleration time, include the following:

- Compressor performance graphs, which are defined by the number and kind of impellers, as well as by the speed, stator-blade position, diffusers, return ducts, etc.
- Moment of inertia of the compressor, couplings, gear train and motor
- Effect on the driving torque by hydraulic couplings [7] for load-relieved acceleration
- Piping and vessel volumes upstream or downstream of the compressor
- Pressure, temperature and molecular weight of the gas in the piping system at the beginning of startup
- Type, characteristics and adjustment of flow throttling equipment (globe valves, gate valves, control valves, non-return valves) in the piping.

From the electrical point of view, the startup time is affected by:

- The motor torque curve, which is governed by the motor type and any equipment possibly installed for starting-current limitation,
- The voltage drop upon motor startup, i.e., the capacity of the electricity main supply circuits.

#### Typical Examples

The following sections will discuss in greater detail the effect of the various plant components on the compressor torque. This discussion is designed to demonstrate the way in which a solution can be found to the starting problem, without having to make any changes to the compressor or electric motor.

#### Influence Exerted by Suction and Discharge Volumes

The various flow sheets and connection diagrams shown in Figure 15 depict a simplified closed refrigeration loop. The principles therein allow the influences, during the starting phase, of the differing suction and discharge volumes to be examined. In total, four different alternatives will be handled:

**Alternative a:** There are large piping and vessel volumes upstream ( $V_1$ ) and downstream ( $V_2$ ) the compressor. Both volumes are of the same size.

**Alternative b:** The compressor receives the fluid from a small volume  $V_1$ , while delivering into a large discharge volume  $V_2$ . The  $V_2/V_1$  ratio is 10.

**Alternative c:** There is a large volume  $V_1$  on the suction side, and a small discharge volume  $V_2$ , with the volume ratio  $V_2/V_1=0.1$ .

**Alternative d:** As opposed to alternative "a", the large vessel volumes have been bypassed, thus reducing the suction and discharge volumes to ten percent of the value applicable in alternative "a". The ratio of  $V_2/V_1$  is unity.

The standstill pressure of the plant was assumed to be four bar in all four alternatives. The position of the throttle valve was not varied in the individual alternatives.

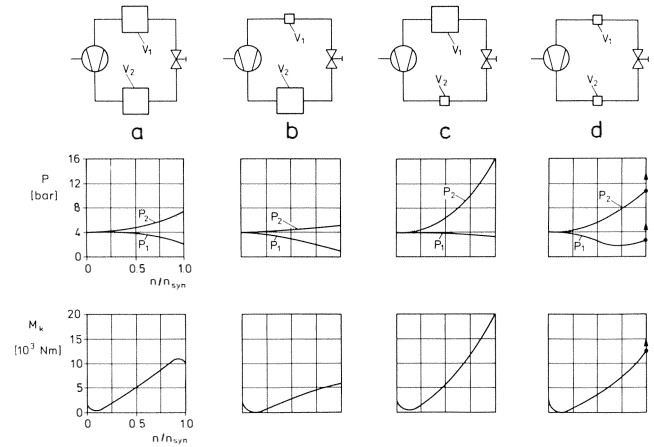


Figure 15. Effect of Vessel Volumes ( $V_1$ ,  $V_2$ ) on the Compressor Suction ( $P_1$ ) and Discharge Pressures ( $P_2$ ) as well as the Compressor Torque ( $M_K$ ).

A diagram of the pressure curve upstream ( $P_1$ ) and downstream of the compressor ( $P_2$ ) is also shown in Figure 15, along with the compressor torque requirement curve,  $M_K$ . In comparing the diagrams for alternatives "a" and "b," it is clearly evident that a smaller suction volume means reduced compressor torque. A negative mass flow balance for the smaller accumulator  $V_1$ , where more mass is withdrawn than freshly added, causes the pressure to drop very rapidly, which means a corresponding decrease in the density of the medium to be handled. The compressor torque is directly proportional to the density.

The opposite case, where a large volume is maintained on the suction side while decreasing the discharge volume (alternative "c"), causes the compressor torque to be increased relative to all of the alternatives. The suction pressure (suction density) experiences only a small change during the startup procedure according to alternative "c," which therefore means a higher torque. To satisfy the requirement of a larger power at the coupling, alternative "c" therefore needs a bigger electric motor. Furthermore, the discharge volume is filled rapidly, which means a higher pressure  $P_2$  downstream from the com-

pressor and upstream of the throttle valve, when compared to alternatives "a" and "b."

In alternative "d", all large vessel volumes are isolated from the compressor. The volume remaining in the suction piping is emptied within a short time, and the discharge volume is also filled quickly. As in alternative "c," the pressure downstream from the compressor rises at a high rate, although the compressor torque, due to the reduced suction pressure and thus suction density, does not reach the same values as in alternative "c."

The arrows entered in the diagrams for alternative "d" indicate that the startup procedure has not yet come to an end when the final speed is reached. There is an infeed of energy by the compressor into the compressed gas which cannot be dissipated, since there is no heat exchanger. The gas temperatures and pressures in the loop will rise at a distinctly higher rate than in the other alternatives, once synchronous speed has been reached, since the operating conditions are not yet stationary.

When considering alternatives "a" and "d," it is obvious that the isolation of large vessel volumes does not automatically mean a reduced compressor torque.

#### *Influence Exerted by the Suction Pressure and Circuit Gas*

Two of the most important variables for the process are the pressure and temperature of the refrigerant at the beginning of the startup procedure. If the compressor is not isolated from the evaporator of the refrigeration loop, the same pressure will exist in the compressor and in the evaporator. During extended downtime, the temperature in the evaporator will become ambient. If a sufficient amount of liquid is available, saturation pressure conditions will arise, depending upon the refrigerant vapor pressure curve. The vapor pressure curves for various refrigerants are presented in Figure 16. For pro-

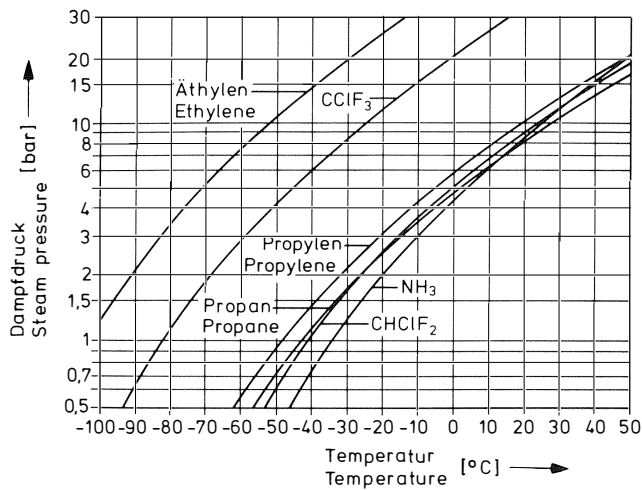


Figure 16. Vapor Pressure Curves for Different Refrigerants.

pane, the saturation pressure at a temperature of about 28°C is approximately ten bar. Such a high pressure also means a high suction density. In the event a very large volume is contained by the vessels and piping upstream of the compressor, the suction conditions will change very little during startup, which at a suction pressure of ten bar as compared to one bar would require the compressor to absorb ten times as much driving power. A common approach in overcoming this problem is to release a sufficient amount of refrigerant from the loop until there is a loop pressure of about one bar. Additionally, the

compressor is isolated from the process by closing the throttle valves in the process piping, with startup being performed in a closed-loop operation by bypassing the flow through the anti-surge control valves, such as previously described for a typical application.

#### CONCLUSION

Possible differences in the performance curves must be duly considered in the parallel operation of refrigeration compressors. This is achieved by installing a load sharing controller in addition to the pressure and anti-surge controls. Load sharing control is to ensure that there is the same distance from the surge limit to the duty points of all the compressors, and that all machines can be simultaneously operated at an optimum efficiency.

The selected anti-surge control ensures, through the use of a dynamic control line, a non-linear amplifier and an additional anti-surge safety line, that even in extreme operating conditions (e.g., during startup and rundown), the compressors are operated within the stable range of the performance graph. The pressure and load sharing controllers have been combined in such a way that there are no instabilities or interactions between them.

When rating a refrigeration compressor driven by an electric motor for closed-loop operation, careful attention must be paid to the torque required at the compressor coupling, which depends on a great variety of physical factors. Dynamic simulation of the startup procedure is a basic prerequisite to ensure that the selected electric motor is able to accelerate the machine train to its rated speed within the permissible starting time.

Some of the most important plant parameters affecting the acceleration time of the compressors are the suction and discharge volumes upstream and downstream of the compressor. As a general statement, it can be said that small suction-side and large discharge-side volumes have the effect of reducing the startup torque.

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