

TECHNICAL REPORT

Pump technology: Improved energy efficiency in refrigeration plants

Dr. R. Krämer



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View of a CO₂ refrigeration plant with a pump system

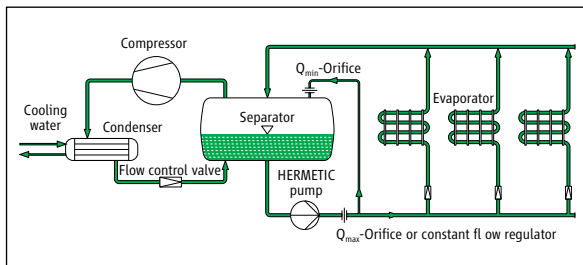


Figure 1: Schematic representation of a large refrigeration plant

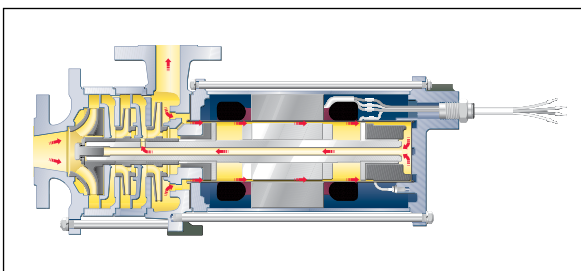


Figure 2: Diagram of a hermetic refrigerant pump, type CAM

According to the European EDL directive (final energy efficiency and energy services), 9 % energy savings over a nine-year-period – in comparison to a reference period – should be possible through targeted measures. A current topic is energy savings in industrial refrigeration plants taking natural refrigerants such as NH₃ and CO₂ as a basis. Up to 70 % of the pump energy consumption can be saved if a frequency converter with a Δp -differential pressure control unit for the pump drive is used instead of a 50 Hz mains operation.

The preferred design for large refrigeration plants today (Figure 1) is a pumping station with forced recirculation of the refrigerant on the low-pressure side. This has several advantages over other systems:

- savings of power due to lower temperature differences with direct vaporisation
- reliable distribution and control of the refrigeration capacity with many evaporators, even ones that are at a distance
- simplification of the pipework and reduction of its dimensions
- improved heat transfer to the evaporators
- concentration of the most important machine units in one room.

Refrigerant pumps must be suitable for pumping refrigerant when it is boiling, be insensitive to cavitation to a great extent, have high operational reliability, and be maintenance-free. Hermetic pumps without shaft sealing allow broad use of refrigerant pumps independent of the refrigerant for the first time.

Figure 2 shows a hermetically sealed refrigerant pump for pumping the liquid cooling medium. The thin-walled stainless steel can serves as a hermetic seal of the pump unit to the outside and protects the winding from the effects of the refrigerant. The motor is liquid cooled. A cooling stream is tapped from the side of the pump that is under pressure to the rotor space, and then fed back into an area of higher pressure after absorbing the motor's dissipated heat through the hollow shaft between the second and third impeller. This prevents vaporisation of the motor cooling stream.

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HERMETIC REFRIGERANT PUMPS IN A SUPERMARKET

For some time now, more and more NH₃/R404 refrigeration plants are being replaced by more environmentally friendly CO₂/R744 refrigeration plants. They supply the freezers in supermarkets, for example. Due to the resulting high pressure, usually only hermetically sealed refrigerant pumps (pressure stage PN40 or PN64) come into question. The following example describes the use of a refrigerant pump in a supermarket with 60 freezers and an overall length of approx. 180 m. CO₂/R744 at a temperature of -4 °C is used as a refrigerant. The pump is designed for 12 m³/h with a pump head of 26 m. The task was to control the pump in an efficient and energy-saving way with the help of a frequency converter. In the process, the pump was to adjust independently to different refrigeration consumption.

A calculation algorithm was used for the design of the pump, which permitted the conversion of the 50 Hz performance curve to smaller frequencies with the help of the similarity laws:

$Q \sim n$, $H \sim n^2$, $P_2 \sim n^3$ as well as $NPSH \sim n^{4/3}$.

There were hardly any problems with the conversion. Clarifying the following questions was more difficult:

- What restrictions are there as a consequence of the changed heat balance between the pump and motor?
The vapour pressure of CO₂ at -4 °C is still 30 bar.
- What losses does the motor have at low speeds?
- What is the situation with the load rating of the slide bearings?
- Is the motor capacity sufficient even for the maximum refrigeration capacity of the plant?
- How does changing the speed affect Q_{\min} und Q_{\max} and / or the NPSH (Net Positive Suction Head)?



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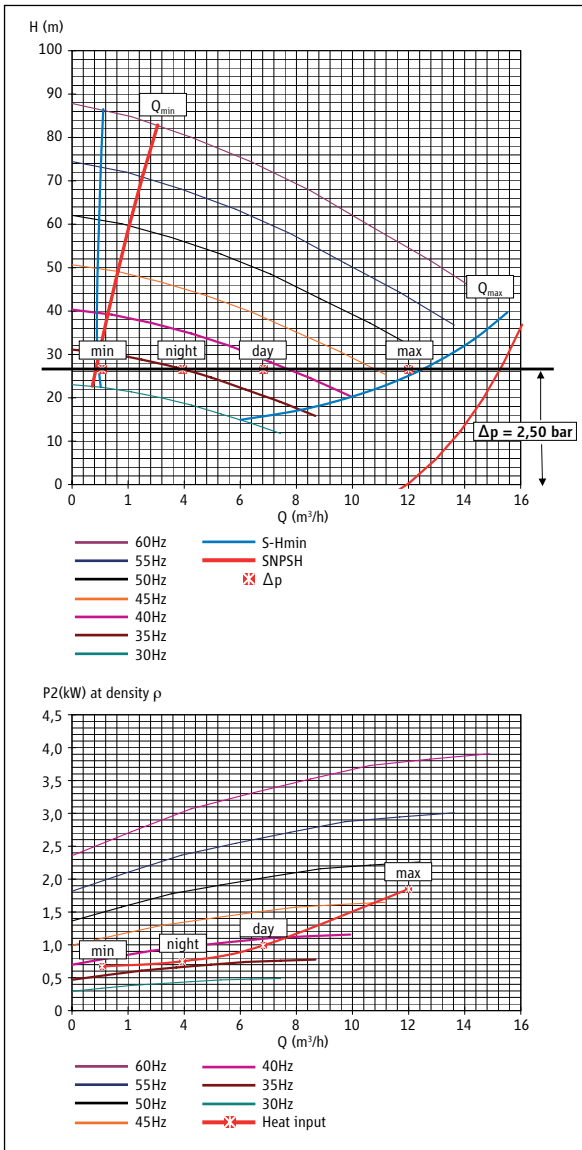


Figure 3: Characteristic curves, 30-60 Hz

Top: H-Q characteristic curves with limiting curves to prevent vaporisation of the motor cooling stream (blue) and to prevent cavitation (red) respectively.

Bottom: P-Q characteristic curves. Equally calculated and measured operating points for Q_{min} and Q_{max}, as well as operating points for day and night operation.

By including the speed as a new degree of freedom, the calculation is mathematically more difficult as the problem becomes multidimensional. The pump head H is a 2-dimensional function of Q and f. Likewise, the shaft power P₂ is a 2-dimensional function of Q and f. With the help of the calculation algorithm, it is possible to show the pump head H and the shaft power P₂ as 2-dimensional cubic spline functions of the variables Q and f. With these spline functions, it was possible to interpolate in any way in the characteristic curve family. Hence, f could be calculated for a stipulated Q and H, or alternatively Q could be calculated from f and H.

A minimum speed of approx. 1800 rpm corresponding to a minimum frequency of 30 Hz resulted from the bearings load rating calculation taking the CO₂ viscosity into consideration. The 50 Hz characteristic curve of the pump, the equivalent circuit of the motor, as well as the physical properties of CO₂ depending on the temperature were used as input values for the calculation program. In addition, there were specifications for the geometry, the outlet of the motor cooling stream, minimum security S-min against vaporisation of the side stream and a minimum security SS-min = NPSHa – NPSHp (plant pump) against cavitation.

OBSERVATION OF THE MOTOR FREQUENCY

The result is the characteristic curve family shown in Figure 3, which also takes the slippage of the motor into consideration. First of all, the heat balance calculation results in a blue limiting curve. This reflects the operating points at which the minimum security S is still 3 m. S is greater than 3 m for all points inside the blue curve. Physically, this means that vaporisation of the motor side stream is not to be expected for all points inside the blue limiting curve. It can be seen that the motor frequency cannot be lowered arbitrarily. The frequency range is limited to approx. 30 Hz by the blue curve. 30 Hz is the lower limit for the bearings load rating of the slide bearings with CO₂.



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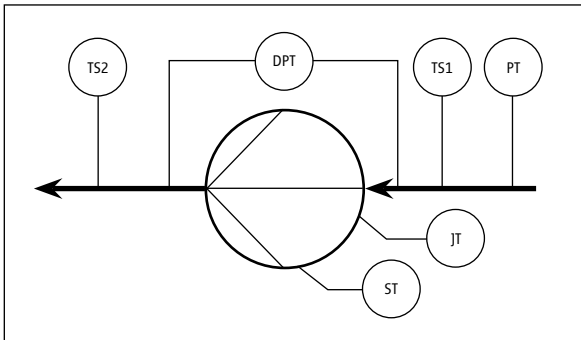


Figure 4: Differential pressure diagram: DPT differential pressure, TS1 suction-side temperature, TS2 pressure-side temperature, PT pressure on the suction side, ST frequency (at the converter), JT power consumption (at the converter)

A second criterion concerns NPSHa and NPSHr. For small rates of delivery, NPSHr increases again as a consequence of the heat generated in the first impeller. If a minimum clearance $SS_{\min} = 0.3 \text{ m}$ is required between NPSHa and NPSHr, then this results in the red limiting curve in the chart. The pump is cavitation-free inside the curve. Cavitation is to be expected outside the curve. One can see that the red curve determines the minimum rate of delivery in this case. In contrast, the maximum rate of delivery in the characteristic curve family is defined by the blue curve.

ADAPTATION OF THE PUMP TO THE PLANT

Practically, the differential pressure between the suction and pressure flange is used (see differential pressure diagram Figure 4) as a controlled variable for the frequency converter. The converter regulates the frequency in accordance with the required rate of delivery or alternatively the refrigeration capacity so that the differential pressure remains constant across the entire control range. Hence, Q becomes a clear function of the frequency. To save energy, the differential pressure must be set to a setpoint in the lower range of the characteristic curve family (Figure 3). There, the pump energy consumption P_2 are proportionally $\sim n^3$ smaller. However, in order to allow the largest possible control range, the frequency should not be too low, or alternatively too close to the minimum of the blue curve. In our example, the setpoint was defined as 26.6 m, corresponding to 2.5 bar pressure differential. The minimum rate of delivery is then the result of the intersection of the $H = 26.6 \text{ m}$ curve and the red curve for $Q_{\min} = 1.07 \text{ m}^3/\text{h}$ at 32.7 Hz. The maximum rate of delivery is the intersection of the $H = 26.6 \text{ m}$ curve and the right blue curve for $12.0 \text{ m}^3/\text{h}$ at 47.3 Hz.

65 % ENERGY SAVINGS

The operating points for day and night operation that were actually measured in the plant were on the $\Delta p = 2.5 \text{ bar}$ lines and are also shown in Figure 3. The operating point during the day was 38.7 Hz / $6.8 \text{ m}^3/\text{h}$ and 35.0 Hz / $3.9 \text{ m}^3/\text{h}$ at night. Thereby, Q nearly changes by factor 2. Both operating points are sufficiently away from Q_{\min} and Q_{\max} . The energy efficiency is obvious. The shaft power at night was only 0.66 kW compared to 1.91 kW at 47.2 Hz.

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Operating Point	Min	Night	Day	Max
Frequency (Hz)	32.70	35.00	38.70	47.20
Rate of delivery (m³/h)	1.07	3.93	6.81	12.00
Pump head (m)	26.60	26.60	26.60	26.60
NPSH pump (m)	0.11	0.16	0.32	0.78
Pump head reserve above vapour pressure (m)	4.28	9.89	8.97	3.87
NPSH reserve above NPSH-pump (m)	0.64	1.25	1.14	0.69
Motor cooling stream (m³/h)	0.92	0.92	0.92	0.92
Heating of motor cooling stream (°K)	1.53	0.94	1.03	1.58
Efficiency of pump (%)	17.00	41.00	47.11	43.41
Efficiency of motor (%)	58.70	64.65	68.40	71.42
Min. pump head at cooling stream outlet (m)	18.57	18.57	18.57	18.57
Increase of vapour pressure height (m)	14.29	8.68	9.59	14.70
Total heating of the delivery stream (°K)	1.00	0.31	0.23	0.25
Power dissipation (kW)	0.67	0.75	0.99	1.85
Current consumption (A)	2.53	2.87	3.36	4.63
Power consumption (kW)	0.74	1.03	1.46	2.68
Shaft power (kW)	0.44	0.66	1.00	1.91
Speed (l/min)	1913	2031	2226	2674
Torque (Nm)	2.18	3.12	4.29	6.83

Chart 1: Calculated numerical values for operating points for min, nights, days, and max

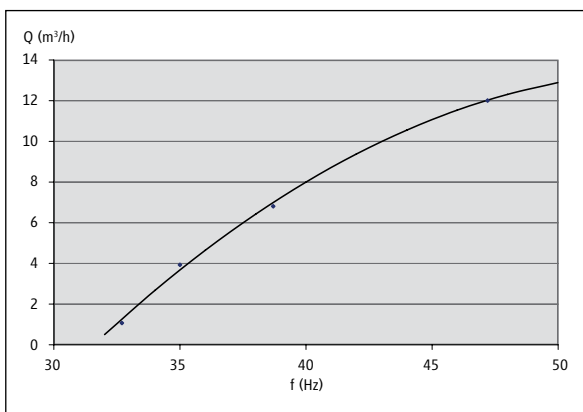


Figure 5: Dependency of the rate of delivery Q (m³/h) on the converter frequency f (Hz)

This means an energy savings of 65 %. Compared to 50 Hz mains operation, that would be even greater savings, i. e. 70 %. The total heat input into the system is reduced to 0.75 kW at 35 Hz compared to 1.85 kW at 47.2 Hz, which means savings of 59 %, or even 65 % compared to 50 Hz mains operation.

Although the rate of delivery Q cannot be measured directly, it can be simply calculated from the frequency with a constant differential pressure. To do this, form a quadratic regression Q vs. f for the above four operating points thereby obtaining the functional connection.

$$Q \text{ (m}^3\text{/h)} = -61.596 + 2.74 * f - 0.025 * f^2$$

This equation only applies for a differential pressure of 2.5 bar. It can be used to calculate the rate of delivery from the measured frequency, and can possibly be shown directly on the converter or tapped externally. Hence, trouble-free pump operation is ensured for the following parameters and within the following operating limits:

- Differential pressure: 2.5 bar
- Rate of delivery Q (m³/h): 1.1-12.0 m³/h
- Converter frequency f (Hz): 32.7-47.2 Hz
- Pump speed n (rpm): 1913-2674 rpm

Under these conditions, there is sufficient pump head reserve (S >3 m) compared to vaporisation of the motor cooling stream. Likewise, there is sufficient NPSH reserve (S >3 m) to exclude cavitation of the suction side of the pump. The pump speeds are also sufficient to ensure the bearings load rating of the slide bearings. The Max operating point corresponds to the intersection point of the blue curve with the Δp lines according to Figure 3, thereby giving the condition of maximum refrigeration capacity of the plant. Mathematically, this results in the numerical values for this operating point that are given in chart 1 under the Max operating point. This also corresponds to the design data of the pump.



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CONCLUSION

The use of a frequency converter with Δp control can save up to 70 % of the pump energy consumption compared to a 50 Hz mains operation. The heat carried into the refrigeration system is also reduced by 65 % in the process. This also means that the necessary refrigerating capacity of the compressors is reduced accordingly. The speed control of the pump therefore results in a double benefit. Both the pump as well as the compressors experience an efficiency increase and hence contribute to the energy savings of the entire refrigeration plant.