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Introduction

In this article we are going to examine the right choice of a two-stage compressor for a specific cooling or freezing application.
In the previous article (RCC 3) we showed why and when we use two-stage piston compressors, which is determined mainly by two criteria:

- Exceeding the "one-stage" deployment area of the compressor in combination with the chosen refrigerant
- The bad swept volume and isentropic efficiency at high pressure differences and "one-stage compression", resulting in a greater swept volume and higher energy consumption.

If a two-stage compressor is chosen on the basis of these criteria, its use will mainly be determined by the maximum interstage pressure and discharge temperature:

- **The ratio of the low pressure volume and the high pressure volume, hereinafter called φ , determines the final interstage pressure p_m.**
- The pressure ratio of the interstage pressure/suction pressure (p_m/p_o) and condenser pressure/interstage pressure (p_c/ p_m) determines the discharge temperature of the low and high pressures respectively. In practice, the limitation is mainly with the refrigerant ammonia due to the high pressure discharge temperature.

A Grasso compressor is built in modules; that means that each cylinder of a particular series is identical. If, for instance, you use the housing of a 9 cylinder model, two 2-stage compressors can be built from it:

- one with 7 low pressure cylinders and 2 high pressure cylinders, a φ of 3, 5, or
- one with 6 low pressure cylinders and 3 high pressure cylinders, a φ of 2.

With Grasso two-stage compressors, depending on series and model, the default ratios of φ go from 2 to 5, making an optimum selection possible in most cases.

Optimum energy φ

The pressure ratio that is the best for energy, giving the highest efficiency, is achieved if the pressure ratio over the low pressure stroke is the same as the pressure ratio over the high pressure stroke. For the optimum interstage pressure you then get $p_m=\sqrt{(p_o \times p_c)}$. You do always need to check whether the requirements of the area of application are met at this interstage pressure. In figure 1, for a condensation "pressure" of about 35 °C, at different suction "pressures" (evaporation temperature), the φ has been plotted out, realizing the optimum pressure ratio.

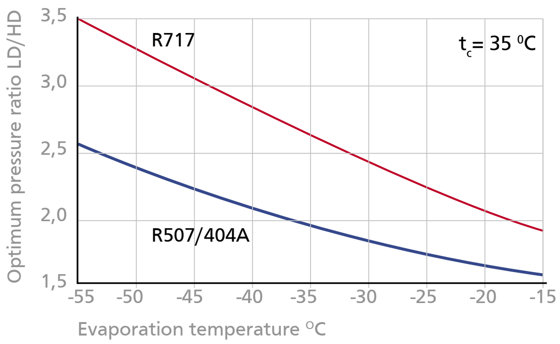

Figure 1

How many cylinders is it useful to use as two-stagers based on modular design? HP and LP cylinders therefore have the same stroke and bore.

This results in the following:

- for the heavy refrigerants with small kappa, 2-stage operation below -30 °C is commercially interesting, so we need a φ of 2, maximum 2.5. A φ of 2 offers the most alternatives:
- for ammonia in the area between -15 and -25 °C a φ of

* Total number of cylinders	3	6	9	12
Cylinders in the low pressure stage	2	4	6	8
Cylinders in the high pressure stage	1	2	3	4

2 is also required and results in the same machines as for the heavy refrigerants. In the typical freezing area, a φ of 3 is required, and for extremely low temperatures a φ of 3.5. The only solution for the latter is a 9 cylinder with 7 low pressure cylinders and 2 high pressure cylinders. For a φ of 3 only three machines can be considered in theory:

** Total number of cylinders	4	8	12
Cylinders in the low pressure stage	3	6	9
Cylinders in the high pressure stage	1	2	3

The switching behaviour of these machines must be monitored closely. See also Fast pull down.

Conclusion: for freezing with “freon” a φ of 2 is the best solution, and for ammonia a φ of 3 is the best.

Capacity control

The volume ratio changes with capacity switching in some machines. That also changes the interstage pressure, so the machine must be checked to see whether it can still operate within the area of application under those conditions. A useful aid for determining this are the graphs below (see figures 2 and 3). The R404A+R507 graph only gives the lines for maximum φ (φ_{max}). The φ used for this condition must remain below this so as not to exceed the maximum interstage pressure. For ammonia, the minimum φ (φ_{min}) lines have also been added. The φ used for this condition must remain above this to prevent the maximum discharge temperature of the high pressure from being exceeded.

φ calculation:

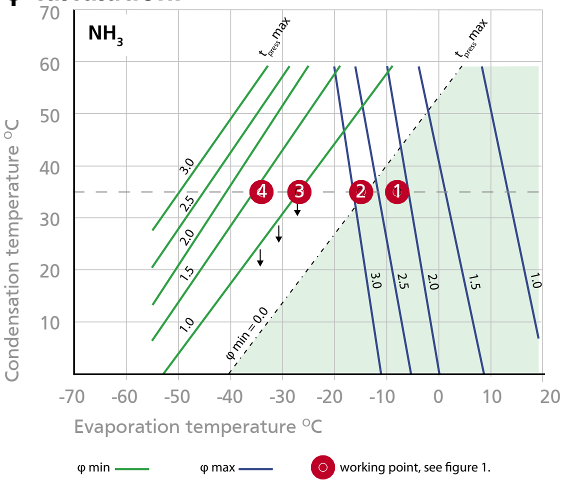

Figure 2

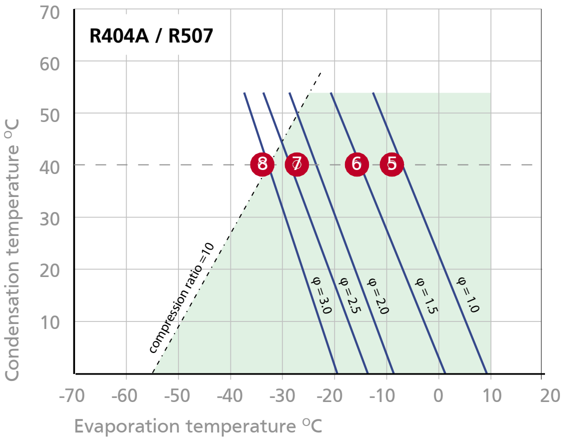

Figure 3

Of course, in the prevailing work situation the allowable φ's can be calculated by the control.

Figure 4. From the enthalpy and mass balance over the interstage pressure, the φ formula can be derived:

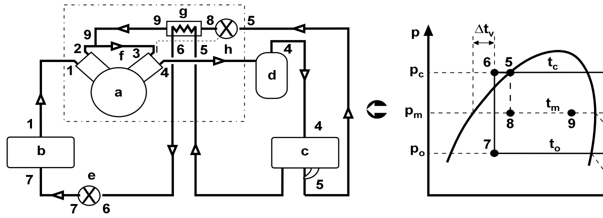

Figure 4

$$\varphi = \frac{\upsilon_1'' \cdot \left(A \cdot \dfrac{p_c}{p_m} + B \right)}{\upsilon_3'' \cdot \left(A \cdot \dfrac{p_m}{p_o} + B \right)} \cdot \frac{h_3 - h_5}{h_2 - h_7}$$

h	enthalpy of the condition in question (J/kgK)
υ″	specific volume of the gas for the condition in question (m³/kg).
p	pressure of the condition in question (bar)
A and B	constants depending on compressor and refrigerant, used to calculate the swept volume:

$$\lambda_{vol} = A \cdot \frac{p_{press}}{p_{suction}} + B$$

For a Grasso 10 these currently look like this:

The endeavour to achieve increasingly better swept volumes requires continuous development. That means that the constants A and B are continually changing.

$T_{discharge}$ max=155°C	Grasso 10
P_{mmax} R507	8.5
A R507	-0.055
B R507	0.97
p_{mmax} R717	8.5
A R717	-0.0473
B R717	0.9659

Calculating the maximum φ (φ_{max}), to keep the interstage pressure below the maximum:

To calculate the maximum φ (φ_{max}) that can be used for running, the maximum values must be used for the values for the interstage pressure:

- υ''_3 and h_3 at the maximum interstage pressure and 10K (system B) superheating

- p_m is the maximum interstage pressure
- h_2 at the maximum interstage pressure and maximum discharge temperature
- h_7 at the saturation temperature of the maximum interstage pressure plus possible superheating
- the other values appropriate to the work condition.

Calculating the minimum φ($φ_{min}$), to keep the discharge temperature of the high pressure stroke below the maximum with ammonia:

The discharge temperature that arises after compression depends on too many factors to be able to calculate the minimum φ($φ_{min}$) the same way as above. However, by restricting the application to the refrigerant ammonia and an interstage "pressure" area of -25 to +15 °C, we are able to make a reliable calculation. The minimum interstage "pressure" can be read from a table after calculation of t_{mmin}:

- t_{mmin} t_c -M+NxdT$_{om}$
- dT_{om} the anticipated superheating at the interstage pressure
- Full load M=50 and N=0.45
- Part load M=40 and N=0.35
- the other values appropriate to the work condition

For each refrigerant and application tables will have to be drawn up in the control, that can be referred to in order to continually calculate the allowable φ's. If you are interested, you can contact Grasso Products for more detailed information.

In practice there are 2 types of two-stage installations:

Installation with 1 temperature level:

For these systems the evaporation temperature, so suction pressure, is relevant and is matched to the process. The interstage pressure in that case is determined by the swept volume ratio between the low and high pressure parts of the two-stage compressor(s). This is typically an application for a compound compressor.

Installation with several (2) temperature levels:

This type of installation is generally built as a booster system with separate HP and LP compressors. That means both temperature levels can be kept at the desired values, regardless of the load on the temperature levels. Compound compressors can only be used successfully here if the load is constant.

As the aim of this article is to provide some insight into the right choice of compound compressor, we will restrict ourselves to installations with 1 temperature level.

To provide insight into the above, we will follow a freezing process step by step:

- 12 ton watery (80%) product per day
- Pre-cooling time about 2 hours (6 °C to freezing point -1 °C)
- Freezing time about 12 hours
- Continued freezing time about 2.5 hours (from freezing point to -20 °C in the core)
- Air circulation such that the temperature difference between product and the air is 7K
- Temperature difference over cooler 7K (average air temperature/evaporation temperature)
- If we explain the air coolers according to the completion of the process determining freezing, the evaporation temperature t° is -1-7-7=-15 °C.
- Dehumidification during pre-cooling means the heat transfer of the cooler will be higher; that means the required temperature difference will be smaller. Average t_o= -8°C.
- In continued freezing, the product will dispose of its warmth less well. This, together with the increasing layer of frost, increases the required temperature difference. On average, you run at -26 °C, and finally at -34 °C.

The required capacities now become:

- Pre-cool : 12,000 x 3.52 x 7 /(2x3600) = 41 kW
- Freeze : 12,000 x 268 /(12x3600) = 74 kW
- Continued freeze : 12,000 x 1.77 x 19 /(2.5x3600) = 45 kW

A 2-stage Grasso 10 compressor can supply this. If we examine the 2 different refrigerants, ammonia and R507, then:

- for the ammonia refrigerant an evaporative condenser will be used (t_c=35 °C). A Grasso 3110 at 1470 rpm can therefore handle the job. See table 1, refrigerant R717
- for the refrigerant R507 an air-cooled condenser will be used (tc=40 °C). The prescribed suction gas superheating is >15K. The less capable heat transfer of R507 puts its evaporation temperature 1 to 2K lower than for ammonia. The Grasso 3110 is not suitable for this application. A Grasso 4210 at 980 rpm can handle the job. See table 1, refrigerant R507.

Fast pull down (FPD)

At a standard φ of 2, the freezing condition would result

Refrigerant	Process	Time in hours	Desired cooling capacity kW	Product temperature from °C	Product temperature to °C	Average/final evaporation temperature °C	Compressor operation	Capacity %	Cylinders low pressure stage	Cylinders high pressure stage	φ	Cooling capacity compressor kW	Interstage "pressure"°C	Shaft capacity kW	Point in R507 area of application Fig. 2/3	Discharge temperature high pressure stage °C
NH_3 R717	pre-cooling	2	41	6	-1	-8/-13	1-stage	-	0	1	-	42	-8	12.5	1	140
	freezing	12	74	-1	-1	-15	2-stage	67	2	1	2	76	9	24.5	2	92
	continuous freezing	2.5	45	-1	-20	-27/-34	2-stage	67	2	1	2	46	-2	21	3	122
	final condition		45	-	-20	-34	2-stage	100	3	1	3	44	-1	23.7	4	120
R507	pre-cooling	2	41	6	-1	-9/-14	1-stage	-	0	2	-	44	-9	18	5	-
	freezing	12	74	-1	-1	-16	2-stage	75	3	2	1.5	74.5	9	37.1	6	-
	continuous freezing	2.5	45	-1	-20	-28/-35	2-stage	75	3	2	1.5	50	-2	31	7	-
	final condition		45	-	-20	-35	2-stage	100	4	2	2	46.9	-1	32.8	8	-

Table 1

in an excessive interstage pressure. Because each cylinder has its own part load mechanism, it is possible to deviate from the standard capacity arrangement in certain cases. This option is called the "Fast pull down" option.
Table 2 shows the standard as well as the "fast pull down" for the compressors used above. For compressors on a batch freezer that is started several times per day at high temperatures, besides the normal part load steps, a pull down schedule will also have to organized. This includes both the 1-stage part and the 2-stage part. The switch from 1 to 2-stage occurs on the basis of the evaporation temperature (suction pressure) or by using the calculated allowable φ. Depending on the number of cylinders available for 1-stage operation, lowering the suction pressure will take a certain time. Time = money; so, if we can shorten this time, for instance by connecting extra high pressure cylinders during this "pull down" time, that provides an economic advantage. A φ =1 is even possible as a transitional solution. The possibilities for this depend of course on the construction of the compressor, especially on the total number of cylinders. Once the design condition has been achieved, the standard capacity control comes into force.

Type	Capacity	Note	Cylinders	Magnetic valve	φ
Grasso 3110	0%	Starting	2	-	0.0
	67%	-	1+2+3	1	2.0
	100%	-	1+2+3+4	1+2	3.0
Grasso 3110 + FPD	0%	Starting	2	-	0.0
	33%	Starting	1+2	FPD	1.0
	67%	-	1+2+3	1+FPD	2.0
	100%	-	1+2+3+4	1+2+FPD	3.0
Grasso 4210	0%	Starting	2	-	0.0
	50%	-	1+2+3	1	2.0
	75%	-	1+2+3+6	1+3	3.0
	100%	-	1+2+3+4	1+2+3+4	2.0
Grasso 4210 + FPD	0%	Starting	2	-	0.0
	25%	Starting	2+4	FPD	0.0
	50%	Starting	1+2+3+4	1+FPD	1.0
	50%	-	1+2+3	1	2.0
	75%	-	1+2+3+4+6	1+3+FPD	1.5
	75%	-	1+2+3+6	1+3	3.0
	100%	-	1+2+3+4+5+6	1+2+3+FPD	2.0

Table 2

Next month we will go into more detail about the latest situation in the field of interstage cooling systems. If you have any reactions, questions, and/or comments about this article or
the Grasso articles from earlier months, contact: Grasso Products b.v. Jan-Pieter Habraken Tel: 073 – 6203 845 jphabraken@grasso.nl.