

# Parallel operation of NH<sub>3</sub> screw compressors - the optimum way

**B Pijnenburg, J Ritmann**

BITZER Kühlmaschinenbau GmbH, Jens Juuls vej 16, 8260 Viby J. Aarhus, DK

E-mail: bas.pijnenburg@bitzer.de, john.ritmann@bitzer.de

**Abstract.** The use of more smaller industrial NH<sub>3</sub> screw compressors operating in parallel seems to offer the optimum way when it comes to fulfilling maximum part load efficiency, increased redundancy and other highly requested features in the industrial refrigeration industry today. Parallel operation in an optimum way can be selected to secure continuous operation and can in most applications be configured to ensure lower overall operating economy. New compressors are developed to meet requirements for flexibility in operation and are controlled in an intelligent way. The intelligent control system keeps focus on all external demands, but yet striving to offer always the lowest possible absorbed power, including in future scenarios with connection to smart grid.

## 1. Introduction

Compressor units for industrial ammonia refrigeration are typically designed for maximum operating conditions (highest temperature, load, ambient conditions, etc.). The typical system has excess of capacity in most operating conditions and in many cases it must adapt itself to continuously varying capacity demand. Different methods exist to vary the capacity of a compressor, whether it is a single compressor, several units in a rack, or several parallel compressor packages.

The choice of the right method is not always easy since each method can be better or worse with respect to the requirements to the capacity control system. The efficiency of the capacity control system is not only depending on the compressor construction. The driveline, especially in variable speed drive (VSD) applications plays an important role in the overall efficiency of the compressor unit.

The most common different capacity regulation methods are described first. A method to compare the total efficiency in the part load range of different units with either one or more compressors is developed. The method is then used to analyze the total efficiency for different number of compressors in a system and for two ways of capacity control.

## 2. Common methods and requirements for capacity control systems

Different methods exist to control the capacity of a compressor system. The quality of the capacity control systems can be measured on different parameters:

- The ability to accurately adapt the capacity to the cooling demand, i.e. how well a certain suction pressure set point can be followed.
- The influence on the efficiency of the compressor and the drive line, i.e. how much the COP of the system is affected.



- The requirements to and load on the power supply.
- The possibility to cover a large capacity range between minimum and maximum load.
- Cost of the capacity control system.
- Operational reliability.
- Noise and vibration levels.

A thorough description of different methods can be found in the publication “Competence in capacity control” [1]. In this article only a short description is given of the most common methods for capacity control with open type ammonia screw compressors:

- On/off cycling
- Speed variation (variable speed drive)
- Slide valve regulation
- Control pistons

The first two methods are not incorporated in the compressor itself, but are about how to operate it. The last two methods require a specific construction inside the compressor. The first and last method offer step wise control, whereas the other two methods enable continuous variation of capacity. Each method is described briefly here, but a comparative study on speed and capacity slide control for screw compressors is described by Blumhardt (2006) [2].

### *2.1. On / off cycling*

On/off switching is the most simple way of varying capacity. In simple systems it can lead to large variation in operating condition and high cycling rates. It can make sense on systems with small load variation or large system buffer capacity. The ability to regulate capacity precisely with this method improves with the amount of parallel mounted compressors.

### *2.2. Speed regulation*

The flow through the compressor is varied with the rotational speed of the rotors. The speed regulated screw compressor requires a specific drive line with a VSD. In most cases the speed regulation does not require changes to the compressor itself, but it affects the way the compressor operates.

### *2.3. Slide valve regulation*

A slide valve parallel to the rotor shaft can be moved to create an internal bypass. Just before the gas is compressed in the rotor cavities, it can bypass internally to the suction side. The slide valve system makes a fairly simple and robust system for both stepwise and step less variation of capacity. The slide valve regulation will also affect the volume ratio (VI) of the compressor. The influence on VI depends on the design of the compressor and the operating conditions. Some compressor designs are capable of independent regulation of capacity and VI.

### *2.4. Control pistons*

One or more radially mounted pistons create an internal bypass for the gas. The control of the bypass is discrete, often with only one or very few steps. Operation with control pistons influences the volume ratio, similarly to slide valve regulation.

The described methods are used both individually but also in combination. It is also possible to regulate capacity on the plant side through bypass systems or suction throttling systems. These systems are not common in ammonia refrigeration, due to their very poor energetic efficiency and therefore not considered here.

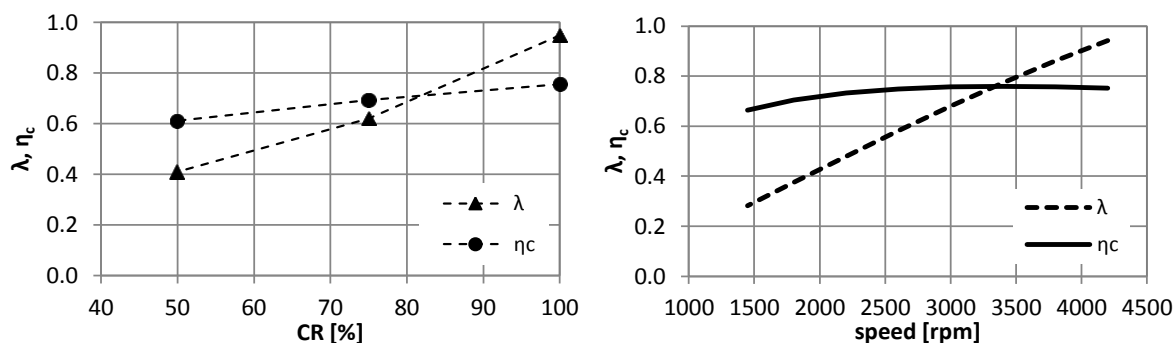
In the further perspective of this article, focus will be on regulation with speed, slide valve and on/off cycling of compressors. All three methods are common engineering practice in ammonia refrigeration today.

The variation of capacity can influence both the volumetric and isentropic efficiency of the compressor. The amount in which efficiencies are affected differ for each method and also depend on the operating conditions and the construction of the compressor. Volumetric efficiency relates to actual swept volume. In this article the term part load index ( $\lambda$ ) is used to indicate the ratio between actual volume flow (at a certain load) and the nominal swept volume at maximum capacity (full load). Variation of capacity will also influence the efficiency of the drive line (motor and or VSD).

### 3. Description of efficiency of capacity control methods

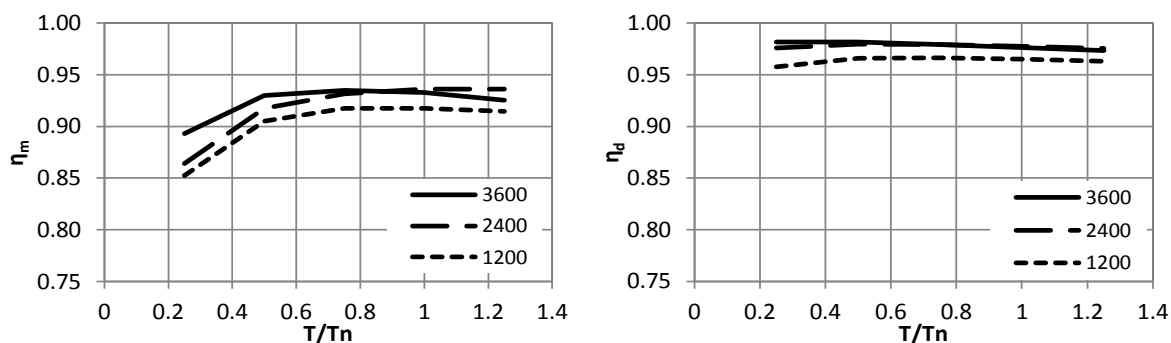
The efficiency of the compressor, the motor and the VSD can each be expressed as function of some form of capacity (not necessarily the system cooling capacity). This efficiency data, also called performance data, is typically expressed in the form of graphs, tables or polynomial functions. The total efficiency of the system can be obtained by coupling the performance data of different components in series and/or in parallel. The relation between cooling capacity and system efficiency cannot be obtained directly from the supplied performance data, but requires some calculation. The relation will also depend on the operation condition of the system (evaporating and condensing temperatures).

Typical dependencies of the compressor part load index ( $\lambda$ ) and isentropic efficiency ( $\eta_c$ ) for a typical chiller condition ( $T_0/T_c = +5/+35$ ) are shown in figure 1. The software to calculate compressor performance [3] contains data based on extensive measurements with different compressor models over the whole range of operating conditions, shaft speeds and capacity slide positions.



**Figure 1.** Example of compressor performance data curves.

A typical dependency of motor and drive efficiency, depending on shaft torque and shaft speed are shown in figure 2. Motor and drive data are calculated with software from a major global supplier of motors and variable speed drives.



**Figure 2.** Example of motor and drive performance data curves.

The graphs show that cooling capacity is not mentioned directly. It must be derived from the performance data and the actual operating conditions of the system.

The capacity regulation systems can have other positive or negative effects on system performance, other than on energetic efficiency. Some of these effects that can be mentioned are:

- Potential for increased capacity with operation at speeds above synchronous with VSD.
- Built-in soft-start function in the VSD results in low motor and power supply load at start-up.
- Positive displacement compressors require practically a constant torque over the complete speed range. Therefore voltage vs. frequency ratio of the VSD must be constant. A VSD can normally not supply voltages above the supply voltage and this means that the motor will be supplied with “under-voltage” during operation above synchronous speed. This means it cannot supply the full torque, which can limit the possible operating conditions of the compressor in this speed range.
- Operation with economizer can be utilized for a large capacity range with speed regulation for most compressor designs. The economizer port is normally closed very early in the unloading process on compressors with slide valve regulation in combination with a fixed economizer port.
- Parallel compounding will require smaller compressors to deliver the same maximum capacity.
- Compressor, motor and drive efficiency increase to some degree with size.
- Parallel compounding automatically gives redundancy.
- Complexity of systems increases with the amount of compressors and combination of different regulation systems.

#### 4. Method of part load efficiency evaluation

The part load efficiency of a compressor system can be evaluated based on the performance data of the components and the equations described in this section. The cooling capacity that a screw compressor can deliver is defined by equation (1).

$$\dot{Q}_0 = \dot{m} \cdot q_0 = \lambda \cdot V_s \cdot \rho \cdot q_0 \quad (1)$$

The shaft power of a screw compressor is defined by equation (2).

$$P_c = \frac{P_{is}}{\eta_c} = \frac{\dot{m} \cdot p_{is}}{\eta_c} = \frac{\lambda \cdot V_s \cdot \rho \cdot p_{is}}{\eta_c} \quad (2)$$

The electrical power consumption of the screw compressor package is defined by equation (3).

$$P_{el} = \frac{P_c}{\eta_d \cdot \eta_m} = \frac{\lambda \cdot V_s \cdot \rho \cdot p_{is}}{\eta_d \cdot \eta_m \cdot \eta_c} \quad (3)$$

The coefficient of performance of the screw compressor package is defined by equation (4).

$$\text{COP} = \frac{\dot{Q}_0}{P_{el}} = (\lambda \cdot V_s \cdot \rho \cdot q_0) / \left( \frac{\lambda \cdot V_s \cdot \rho \cdot p_{is}}{\eta_d \cdot \eta_m \cdot \eta_c} \right) = \frac{q_0}{p_{is}} \cdot \eta_d \cdot \eta_m \cdot \eta_c = \frac{q_0}{p_{is}} \cdot \eta \quad (4)$$

Plotting the COP as function of the cooling capacity ( $\dot{Q}_0$ ) is the traditional way of evaluating the part load performance. One can see from the above equations, that for a certain operating condition and for a certain size of compressor system, there is a linear relation between the cooling capacity and the part load index ( $\lambda$ ) and between the COP and the energetic efficiency ( $\eta$ ). By showing the energetic efficiency as function of the part load index we have a way to show the part load performance on a dimensionless scale from 0 to 1, which makes it easy to compare for different operating conditions and different sizes of systems.

The total part load index and energetic efficiency for a compressor package with multiple compressors can be found from the weighted sum of the individual efficiencies according to equation (5) and equation (6).

$$\lambda = \frac{\sum_{i=1,n} \lambda_i \cdot V_{s,i}}{\sum_{i=1,n} V_{s,i}} \quad (5)$$

$$\eta = \frac{\sum_{i=1,n} \lambda_i \cdot V_{s,i}}{\sum_{i=1,n} \eta_{d,i} \cdot \eta_{m,i} \cdot \eta_{c,i}} \quad (6)$$

The performance data curves (figure 1) show that part load index ( $\lambda$ ) and energetic efficiency ( $\eta_c$ ) for a compressor can be controlled by two parameters: speed and capacity slide position. Operating conditions are assumed constant in this work, but they can also depend on capacity, i.e. evaporating temperature will rise as evaporator load decreases.

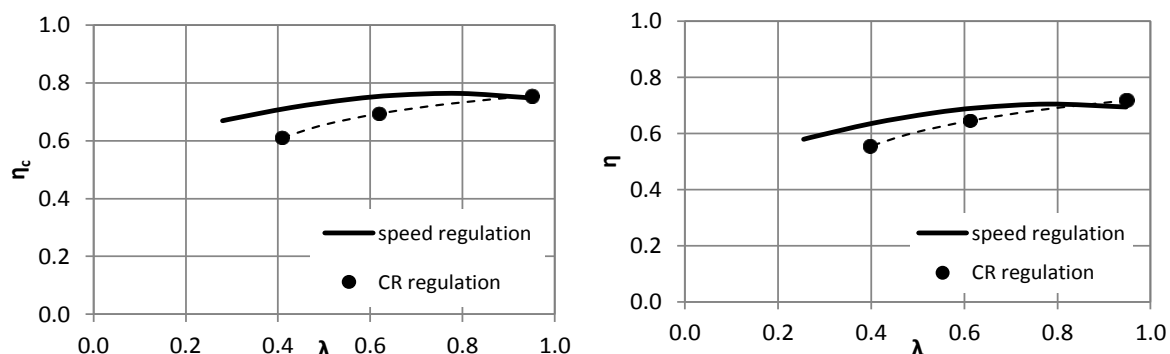
The energetic efficiency of the motor ( $\eta_m$ ) and the VSD ( $\eta_d$ ) depend on speed and torque (as seen in figure 2). The torque and in case of VSD also the speed will vary with capacity. The compressor shaft power relates to the torque and speed according to equation (7).

$$P_c = \frac{\lambda \cdot V_s \cdot \rho \cdot p_{is}}{\eta_c} = T \cdot \omega \quad (7)$$

The shaft speed is either constant or controlled actively, thus the shaft torque depends only on compressor part load index, energetic efficiency and speed. A motor is typically selected with 10% torque reserve at maximum (or nominal) load giving the torque at maximum system load to be 90% of the nominal motor torque ( $T/T_n = 0.9$ ). The factor 0.9 is not constant and will differ based on the available motor sizes and application type. The part load torque of the motor relative to its nominal torque is defined by equation (8).

$$\frac{T}{T_n} = 0.9 \cdot \frac{\lambda}{\lambda_n} \cdot \frac{\eta_{c,n}}{\eta_c} \cdot \frac{\omega_n}{\omega} \quad (8)$$

The part load performance, described by the dependency between part load index and energetic efficiency can be derived quite easily from the compressor performance data. By combining equation (8) with interpolation of motor and drive performance data, it is also possible to derive the part load efficiency of the motor and drive as function of the part load index. An example of the relation between the part load index ( $\lambda$ ) and the energetic efficiency ( $\eta$ ) is shown in the figure 3.



**Figure 3.** Energetic efficiency as function of compressor part load index for compressor only (left) and complete system (right).

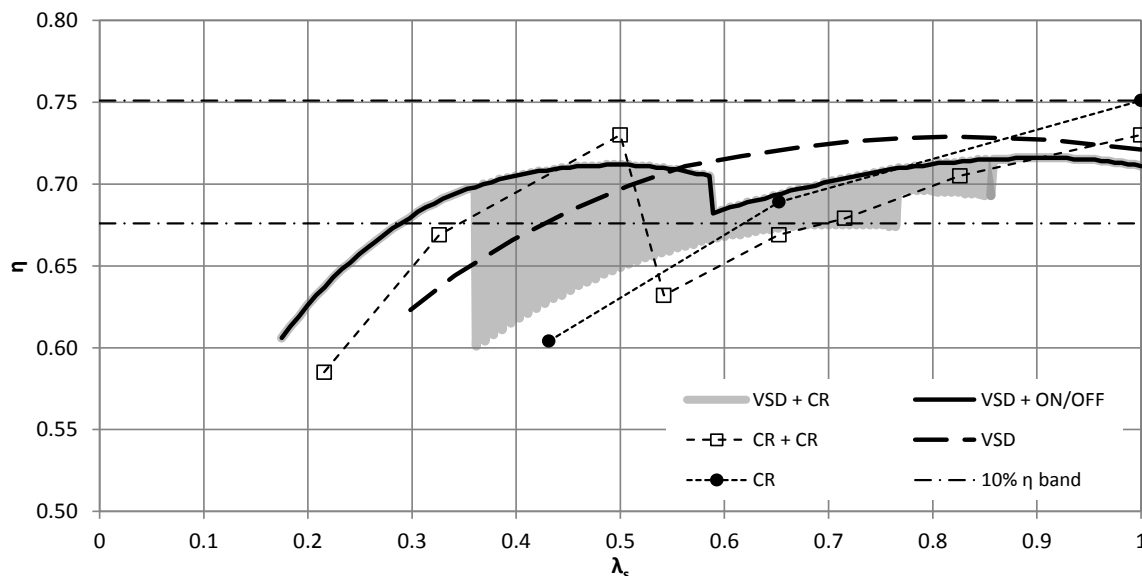
## 5. Comparison of different solutions

The part load performance of different compressor package solutions is compared. The most basic compressor only regulates with a slide valve. Next step is to add a VSD and finally to use multiple compressors. Simulations of many different solutions were done to map the different combinations.

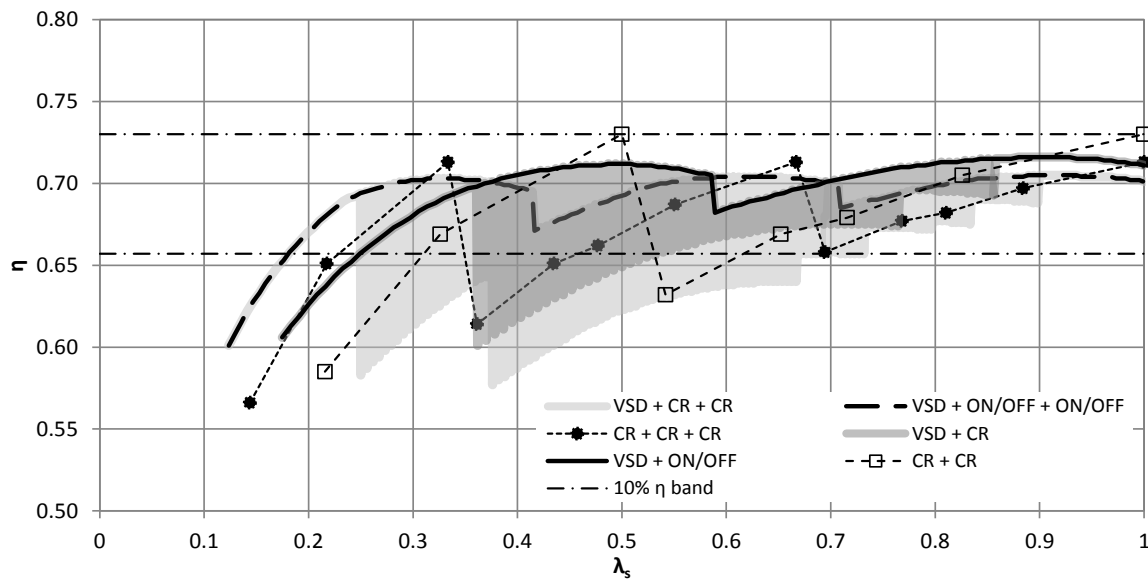
All comparisons are done for the same total maximum capacity. For easy comparison, the values for part load index were scaled (indicated by  $\lambda_s$ ) to be exactly one at maximum capacity. The effect of size, where larger machines tend to have better efficiency is taken into account by using actual performance data from appropriate sizes of compressors, motors and drives. The effect of size and the effect of using a VSD becomes evident when looking at the nominal maximum load efficiency. Multiple smaller compressors will have a slightly lower maximum load efficiency when compared to fewer larger compressors. Solutions with VSD will have a slightly lower maximum load efficiency than comparable size compressors without. The losses in a VSD on multiple compressor systems naturally only affect the one compressor with VSD, which has a positive effect on the total losses.

Figure 4 shows a comparison of the total efficiency over the capacity range for a solution with one or two compressors. It shows that solutions without VSD only have an efficiency close to the maximum value in a limited range. The solutions with VSD show a more constantly high efficiency level. The solution with two compressors can keep the high efficiency down to approximately 28% capacity, whereas a single VSD compressor shows considerable drop in efficiency below 43% capacity. All solutions have high efficiency in the upper range above approximately 65% capacity.

The horizontal dash-dotted lines show a band width of 10% efficiency, in order to show the magnitude of the differences between the compared solutions. It can be noticed that compressor performance data according to EN12900 allow a tolerance of up to 10% on published compressor efficiency data (COP). Since 2014 this tolerance is applicable for the entire part load range.



**Figure 4.** Comparison of part load efficiency for package with 1 or 2 compressors.



**Figure 5.** Comparison of part load efficiency for package with 2 or 3 compressors.

Figure 5 shows the comparison between a solution with either two or three compressor. Again it can be seen that solutions with VSD show a constant high efficiency. The solution with three compressors (one VSD) can even keep a very high efficiency down to approximately 17% capacity. It can also be noticed that a three compressor solution with only slide regulation (CR) has a high efficiency down to approximately 23% capacity, except from a minor load area around 37 - 43%.

The analysis of the different scenarios were done for operation at a fixed operating condition ( $T_0/T_c = +5/+35^\circ\text{C}$ ). The results will change depending on the operating conditions and this should be considered when searching for the optimum solution. The results show that the flexibility to vary capacity increases with the amount of compressors and when adding minimum one VSD. Multiple compressors, combined with variable speed, increase the capacity range significantly in which the efficiency stays very close to the full load efficiency. The control system must off course be capable of handling this in order to take full advantage of this potential.

## 6. Conclusions

The optimum way of providing the highest possible part load efficiency over a widest possible range of capacities seems to be a compressor system with multiple compressors where one is fitted with VSD. The investigated systems can regulate capacity as far down as 17% (for a 3 compressor unit), while at the same time keeping the efficiency at a very high level (within 10% of the maximum).

If only operating in the upper capacity range (above approximately 65%), then all the presented solutions are within 10% of the maximum efficiency and the use of VSD may not even be justified economically.

Over a wide range of capacities, the differences in efficiency between the solutions are smaller than the allowed EN12900 COP-tolerance for the compressor only. Therefore a more detailed analysis is often needed to find the best solution. This analysis must take into account the variation of capacity demand,  $T_0$  and  $T_c$  over time. If this analysis is not available, then the solution with multiple compressor systems and VSD always adds flexibility in regulation and redundancy to the system.

## 7. Perspective

The importance of selecting the correct size of compressor, motor and drive suitable for the nominal load shall be mentioned. Very good part load efficiency should not be jeopardized by selecting excessively oversized equipment.

Control systems on the compressor package and other parts of the refrigeration system play a very important role to be able to take full advantage of the potential that multiple compressors and VSD offer. The maximum potential can only be exploited if the control system is able to control all parameters in an optimum way. Future control systems may also need to communicate with smart grid systems and production planning systems to be able to control capacity with some information of the expected future load. High flexibility and high efficiency in a large range of capacities is highly requested for these demands.

### Nomenclature

COP	Coefficient Of Performance	[-]
CR	Capacity slide Regulation position	[%]
$\eta$	Total energetic efficiency of the compressor package	[-]
$\eta_c$	Isentropic efficiency of the compressor	[-]
$\eta_d$	Energetic efficiency of the variable speed drive	[-]
$\eta_m$	Energetic efficiency of the electrical motor	[-]
$\lambda$	Part load index, related to compressor nominal swept volume	[-]
$\lambda_s$	Scaled part load index, related to compressor nominal capacity	[-]
$\dot{m}$	Mass flow of refrigerant through compressor	[kg/s]
$P_{el}$	Electrical power consumption of the compressor package	[kW]
$P_{is}$	Isentropic power consumption of the compressor	[kW]
$p_{is}$	Specific isentropic compression work	[kJ/kg]
$\dot{Q}_0$	Cooling capacity of the compressor	[kW]
$q_0$	Specific cooling capacity	[kJ/kg]
$\rho$	Density of refrigerant at compressor inlet	[kg/m <sup>3</sup> ]
$T$	Shaft torque	[Nm]
T0	Evaporating temperature	[°C]
Tc	Condensing temperature	[°C]
$V_s$	Nominal swept volume of the compressor at maximum load	[m <sup>3</sup> /s]
VSD	Variable Speed Drive	[-]
$\omega$	Shaft rotational speed	[rad/s]
$n$	Subscript to indicate nominal values at maximum load	[-]

### References

- [1] BITZER publication A-600-5, 2014, Competence in capacity control
- [2] Blumhardt R, 2006, Capacity control of screw compressors: speed or slide control – a comparative study, BITZER publication SV-0402-GB
- [3] Performance data for screw compressors: BITZER Software v. 6.4.2