HOLISTIC APPROACH TO ELECTRO-HYDRAULIC DRIVE SOLUTIONS FOR HYDROGEN PISTON COMPRESSORS

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ABSTRACT

Compressor drives with hydraulic drive technology are showing good market growth due to the trend towards hydrogen. The application seems very simple at first due to its repetitive cycle, but a closer look reveals some technical challenges. However, with a deep understanding of the end user's point of view as well as the physical principles, it is possible to design a hydraulic system that meets the target cost but also achieves robustness, high flow rate and good energy efficiency. However, no general recommendation can be given for a specific system structure. Though in general, it can be seen that variable displacement pumps should be the favoured solution for motion reversal in a wide target range.

Keywords: Industrial hydraulics, Hydrogen, Compressor drive, Efficiency

1. INTRODUCTION

To meet climate targets, many sectors must switch to renewable energies. The use of hydrogen as an energy carrier ranks among the key technologies for sectors where electrification has failed so far or for long-term energy storage. Due to its low volumetric energy density, hydrogen for mobile applications (cars, trucks, construction machinery, railroads, shipping, aviation) must be compressed so that storage units can be dimensioned as small as possible. During the refuelling process of these vehicles, a powerful compressor is required on site because the pressure in the supply line is very low or the pressure in the supply storage tank decreases with increasing extraction.

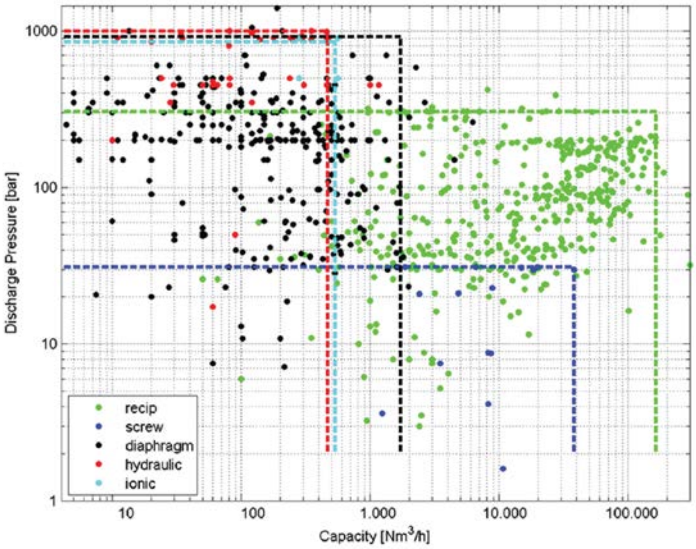

Figure 1: Compressor map with current industrial references (dots) and performance ranges (lines) for pure hydrogen. [1]

In addition to this end use, many of the pressurized components need to be validated during development and often 100% tested at the end of production, which is why industry also has a need for compressors.

Compressors for these applications can be designed using several functional principles. Due to high end pressure of up to 1000bar, large pressure differences of up to 900bar and small to medium flow rates, diaphragm and piston compressors are particularly suitable for these applications (see figure 1). Due to further requirements such as the ability to start-stop operation and low space requirements, hydraulically driven piston compressors have gained great relevance.

While hydraulic piston compressors generally exist already for a long time, the design of the hydraulic system is often simple and not contemporary, because this application didn't get a lot of attention. That's why the aim of this paper is to explain the special requirements and the basic approach for optimizing hydraulic systems for this use case.

2. SYSTEM STRUCTURE

First, the typical structure of a reciprocating compressor is explained using Figure 2. It consists of three main components: the drive, the compressor, and the system control. The task of the drive is to convert the electrical energy into mechanical energy while generating as little power loss as possible, which will be dissipated in the form of heat. In addition, it is also responsible for controlling the mass flow since the compressor usually acts only passively. In contrast to the drive unit, the heat generated at the compressor is mostly due to the physical laws of compression and not just to losses due to inefficiency. It is also often unavoidable that part of the heat output of the compressor flows back into the drive unit.

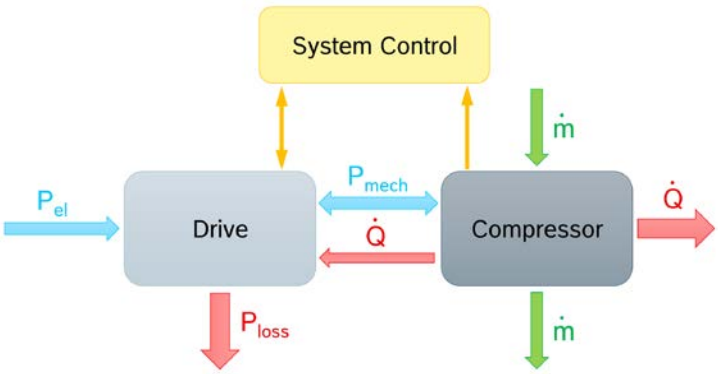

Figure 2: Basic structure of reciprocating compressors.

In addition to this very simple design, several compressor stages are usually used, which can be connected either in parallel or in series. Figure 3 and figure 4 show the typical design of a hydraulic double rod cylinder connected to single gas pistons on both sides. But the same applies to reciprocating compressors with a crank shaft.

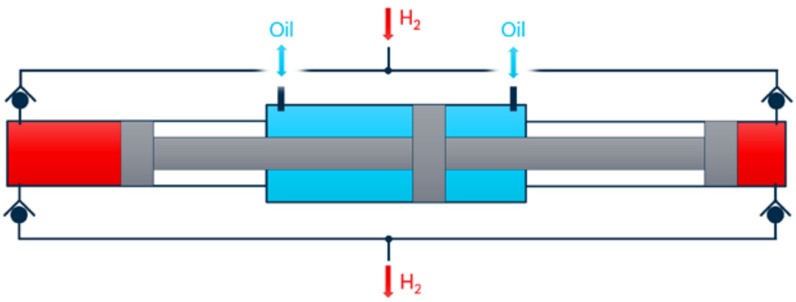

Figure 3: Typical design of a hydraulic piston compressor with parallel gas pistons.

In the case of a parallel connection, the individual pistons are moved out of phase, like in a combustion engine, so that the force of the inflowing gas can be used for compression on the other piston during the return stroke, because hydrogen already has an increased pressure in the inlet in the applications mentioned. In this way, the mass flow can be increased with reduced electrical input power but increased energy requirements. Furthermore, the flow rate is stabilized.

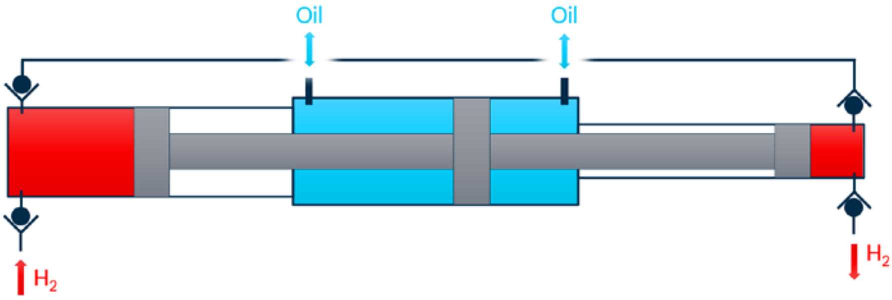

Figure 4: Typical design of a hydraulic piston compressor with serial gas pistons.

With the series connection, the sometimes very large pressure ratio is broken down into smaller steps. This primarily improves efficiency, since the dead volume is not as significant, and the compression curve approaches the isotherm through intermediate cooling. In addition, if the pressure ratio is too high, the heating of the gas can be so severe that the service life of the compressor is massively reduced.

Although multistage compressors are always required in practice for the reasons mentioned above, they are not considered below because the findings are transferable, and the complexity can be reduced.

3. CUSTOMER REQUIREMENTS

To be able to design the system in the best possible way, the customer's point of view and its requirements must first be understood. At first there are global requirements, which must always be met, such as the purity of the hydrogen or compliance with safety requirements. But in addition to them there are also various specific requirements and performance data that differ from customer to customer. The following explanation is not universal but should be a valid simplification of the complex requirements. The target is to list only requirements that can be provided with hard numbers and no soft factors.

Table 1: Specific boundary conditions of the application.

Boundary condition	Parameter	Unit
Electrical connection rating	Electric power	kW
Footprint	Area	m²
Country specific certification	-	-
Environmental condition	Temperature, dirt, corrosion protection, vibration, noise level	°C, dB, …
Max. initial cost	Price	€ / $

First, there are some constraints or filters that are imposed by the application. For each of the items listed in Table 1, there is a limit that must not be exceeded. However, it is of limited added value to the end customer if the system is better than the limit value. For example, the electrical power supply may be limited to 80kW by local restrictions. A compressor with an average power requirement of 120kW cannot be operated, but it is only of secondary importance how far one is below the limit value. The maximum initial costs in Table 1 are only intended to limit the total budget, but the economic decision should not yet be made here.

After the available systems have been pre-selected with the help of the boundary conditions, the relevant ones can be compared based on their KPIs (= Key Performance Indicator). The most important KPI is the flow rate, as it determines how quickly refueling processes can take place or how long pressure tests will last. However, the value is not static since it depends strongly on the inlet pressure and the inlet temperature of the hydrogen (see formula (5)). Therefore, standardized conditions should be used here. In the best case, different extreme points are selected that show the behavior in the relevant operating range.

Table 2: Key Performance Indicator of a compressor system.

KPI	Parameter	Unit	Quantification
Flow rate	Mass of H2 per hour	kg / h	Higher is better
Initial cost	Cost per flow rate	€ / (kg / h)	Lower is better
Energy efficiency	Energy per mass of H2	(kW × h) / kg	Lower is better
Service Life	Time interval	h	Higher is better

The initial costs are naturally very important. Since different systems rarely have the identical flow rate, the costs should therefore only be compared specifically, as acquisition costs per flow rate.

At present, energy efficiency is often not as highly weighted as the other three aspects, but it will become increasingly important in the future. Like the flow rate, the energy requirement per kilogram of compressed hydrogen can only be determined as a function of a specified operating point.

Finally, the time interval between inspections is very important, as this can seriously shift the cost structure in retrospect. Especially at refueling stations, where consumers depend on refueling their vehicles, because no other refueling station can be reached with the remaining range, a maximum availability must be realized.

4. PHYSICAL PRICIPLES OF THE COMPRESSOR

For a better understanding of the optimization approaches, the basic physical process of polytropic compression and the positive and negative influencing factors will be explained first. For this purpose, the ideal process of isothermal compression without dead volume with ideal gas is explained first, and then the individual harmful influences and their effects are shown.

In practice, hydrogen behaves only like an ideal gas near atmospheric conditions (1 bar, 20°C). At higher temperatures and pressures, the repulsive forces between the molecules become increasingly significant. This means that the actual density of hydrogen is lower than calculated with the ideal gas equation. As a result, the pressure builds up earlier, similar to the difference between isothermal and isentropic compression, increasing the work required for compression and thus also increasing the heat generated. Furthermore, the mass flow rate at high inlet pressure is lower because the compressor chamber is filled with less mass. In turn, the negative effects of dead volume are slightly reduced since there is less gas mass present in the same dead volume. All these effects change the quantitative design of the compressors, but not the qualitative behaviour. Therefore, it is acceptable to use the simplified approach with ideal gas for optimization, if exact numerical values for the flow rate are not required.

The pressure build-up can be described with formula (1), where the polytropic factor n depends on the type of compression. The isothermal compression ($n = 1$) assumes constant temperature, which can only be achieved if the cooling is very intense, and the compression speed is infinitely low. The opposite would be the isentropic or adiabatic process ($n = \kappa$, for hydrogen $\kappa = 1{,}4$), where no cooling or infinitely fast compression is assumed.

$$p(V) = p_0 \times \left(\frac{V}{V_0}\right)^n \tag{1}$$

Real compressors do not follow either process, so a polytropic change of state must be used here. The pressure curve stays between the previous two mentioned but tends more towards isentropic. Since heat transfer is faster at larger temperature differences, the polytropic exponent changes continuously during the process. For simplification, an empirically determined value between 1 and κ can be chosen.

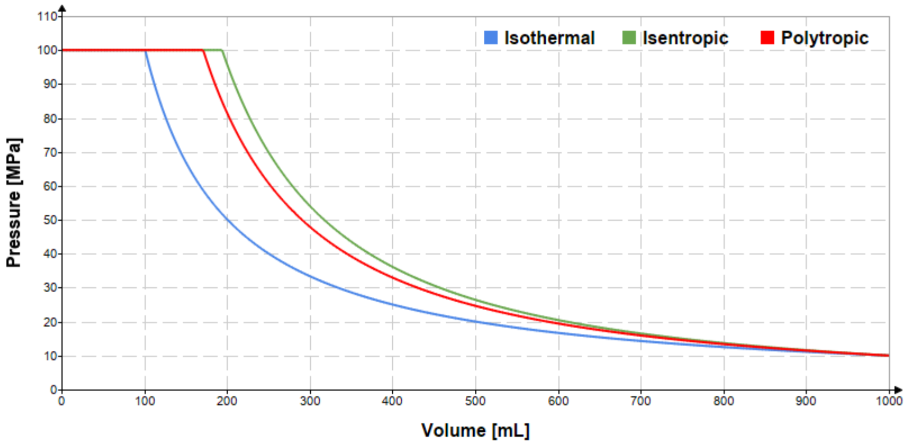

Figure 5: Pressure curve of the compression process from 10MPa to 100MPa.

In continuous operation, a cyclic process takes place, which is shown in Figure 6. The work done for compression can be calculated using formula (2) or figuratively from the area enclosed by the cycle. The waste heat, which is transferred to the cooling system, corresponds to the compression work reduced by the enthalpy change of the hydrogen (see formula (3)). The enthalpy of ideal gas only increases due to temperature, for real gas it would be temperature and pressure depended. In total the whole temperature increase must be cooled since the hydrogen can't be used while it is hot in the most applications. But for the heat dissipated at the compressor itself, it is only relevant how hot the gas is during discharge. Since the pressure increase is given, the formula thus establishes the relationship between the initial temperature of the hydrogen and the cooling capacity needed to lower the discharge temperature below the isentropic process to the polytropic process. Therefore, the discharge temperature must also be calculated with the assumed polytropic factor and is included in the formula (4).

If the maximum temperature and required cooling capacity are to be calculated for dimensioning of the system it is strongly advised not use the ideal gas formulas shown here, but rather look up the real gas values, e.g. in REFPROP. [2]

$$W_{comp} = -\oint p\, dV = \int_3^1 p_{out}(V)\, dV - \int_3^1 p_{in}(V)\, dV \tag{2}$$

$$Q_{loss} = \Delta W - \Delta H = W_{comp} - m_0 \times c_{p.H2} \times (T_{out} - T_0) \tag{3}$$

$$T_{out} = T_0 \times \left(\frac{p_{out}}{p_{in}}\right)^{\frac{n-1}{n}} \tag{4}$$

The ideal compressor without dead volume follows the dotted line in Figure 6. In real machines, however, dead volume is always present, which distorts the cycle according to the solid line. At the end of the stroke **3**, there is still volume present with the output pressure. During the return stroke, this volume must therefore first be expanded to the inlet pressure **3-4** before further gas can flow in via the inlet valve **4-1**. The greater the pressure ratio, the more noticeable the dead volume. That's why an extreme case with pressure intensification factor 10 has been chosen as an example.

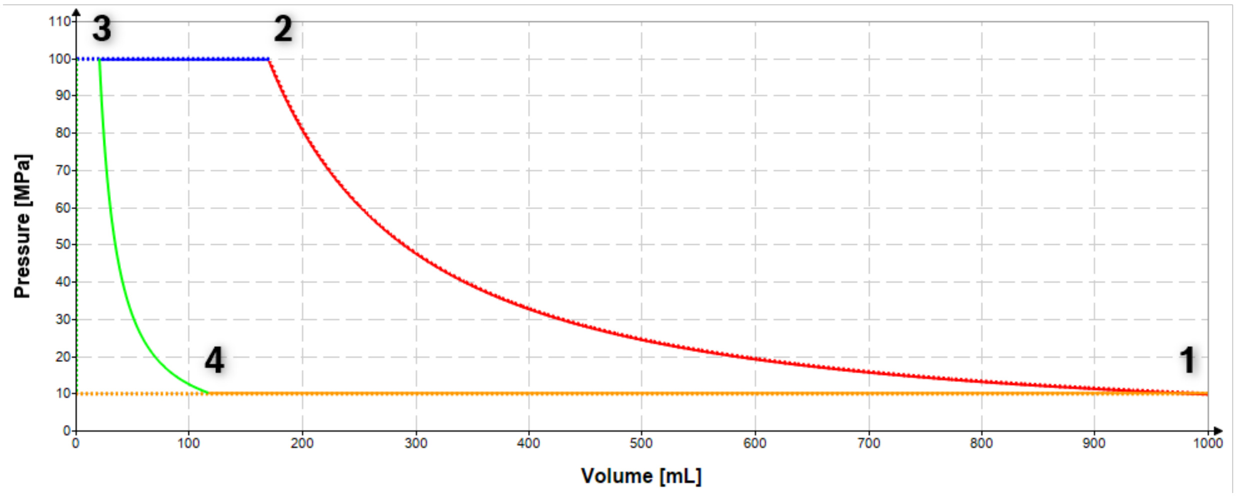

Figure 6: Cycle for continuous operation.

The flow rate per stroke is determined only by how much gas flows into the compressor during the return movement after the expansion of the dead volume **4-1** (formula (5)). For maximum efficiency, it is therefore also important to fill the compressor chamber with gas as cool as possible since the higher density leads to more mass flow for the same stroke volume.

$$m_0 = \rho_{H2} \times V_0 = \frac{p_0}{R_{H2} \times T_0} \times V_0 \tag{5}$$

Furthermore, it is also evident that the dead volume reduces the flow rate, since the volume available for the inflowing gas no longer corresponds to the displacement volume (ΔV between **3-1**), but only a reduced effective volume (ΔV between **4-1**) remains, which can be calculated with formula (6). It is important to understand that the polytropic coefficient in this equation refers to decompression and will therefore have a different value than during compression. For simplicity, identical values are used in the diagrams.

$$V_0 = V_{stroke} - V_{dead} \times \left[\left(\frac{p_{out}}{p_{in}}\right)^{\frac{1}{n}} - 1\right] \tag{6}$$

The energy efficiency is not affected by this, if the work of the expanding gas can be used, since the area of the cycle is reduced in proportion to the flow rate.

The continuous mass flow will then be determined as a function of the stroke frequency f or the mean piston speed $\bar{v}$ according to formula (7).

$$\dot{m} = m_0 \times f = m_0 \times \frac{\bar{v}}{2 \times l_{stroke}} \tag{7}$$

The consideration of further losses such as the pressure losses at the inlet and outlet valves as well as leakage via the piston seal plays no role for the optimization of the drive and are therefore not taken into account.

As a last step, the power curve of the process should be calculated to design the required drive unit. For this purpose, the pressure curve (1) must be multiplied by the area of the gas piston and the velocity profile of the drive unit (8). Since the pressure curve is dependent on the volume, but the formula for power should only be dependent on time, (9) is used to establish the time dependence of the volume.

$$P_{comp}(t) = p(V(t)) \times A_{piston} \times \dot{x}_{drive}(t) \quad (8)$$

$$V(t) = V_0 - A_{piston} \times x_{drive}(t) \quad (9)$$

Figure 7 shows two different motion profiles as examples.

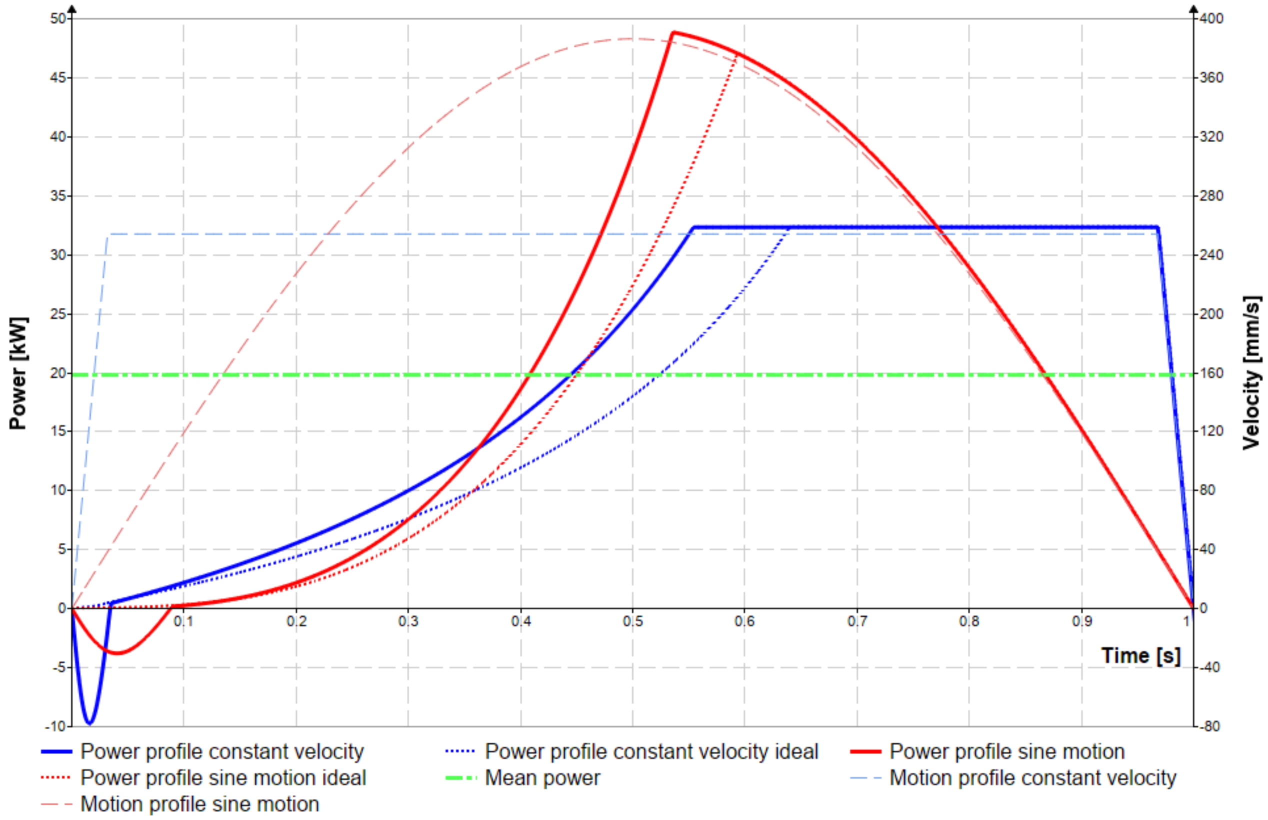

Figure 7: Power curves of different motion profiles.

In red a sine motion is visualized whereas in blue a typical motion of a hydraulic cylinder with short acceleration phases and constant velocity over a major part of the stroke is shown. To illustrate the losses in the compressor, the dotted line shows the ideal compressor with isothermal change of state and without dead volume. Two significant deviations can be seen. On the one side there is an area with negative power at the beginning of the stroke during the decompression of the dead volume, on the other side the maximum power is reached earlier, because the target pressure is reached faster due to the heating of the gas. Likewise, peak power of both motion profiles is very different, with both being significantly larger than the average power. This is due to the fact that the counterforce only builds up in the course of the stroke, and the speed is controlled independently of this.

For hydraulically driven systems, the trapezoidal profile is most important, as it requires a lower peak power and thus a smaller power unit. Furthermore, the maximum speed of the hydraulic cylinder is lower for the same mass flow, which has an advantageous effect on flow losses and seal life. Section 5 therefore assumes a trapezoidal profile only. For the acceleration and deceleration high values can be achieved, so the smoothening effect on the profile is quite small. Since the moving mass is low, no damage by pressure spikes should occur.

5. OPTIMIZATION APPROACHES

In this section, the possibilities of drive technology to optimize the customer's KPIs will be shown. It is assumed that the basic design has already been carried out according to the end application and that the boundary conditions from Table 1 are therefore already fulfilled.

5.1. Flow Rate

To maximize the flow rate, the influencing factors of formulas (5) to (7) should be analysed. The drive unit has the possibility to achieve an increase of the mass flow rate via a high average piston speed as well as a precise switching near the end position, in order to minimize the dead volume.

The high average piston speed can be achieved with a short stroke and high switching frequency as well as with a long stroke and low frequency. However, since the negative influence of the dead volume is proportional to the stroke volume (see formula (6)), a larger stroke volume, with identical amount of dead volume, can reduce the negative effects, which is why the longest possible stroke is preferable from this point of view.

To achieve precise end position switching, a highly dynamic and high-resolution measuring system is required in addition to a suitable control concept. This is particularly challenging for the design of the cylinder since double rod cylinders are state of the art and the common displacement transducers cannot be applied there. This is further complicated by the explosion protection requirements, which lead to severe restrictions in the design of the sensor electronics.

The most common solution is therefore the classic inductive limit switch, which should ideally have an adjustment option so that the manufacturing tolerances can be compensated. Encoders on the outside of the cylinder tube are more difficult to implement and are therefore not widespread, but they can be implemented in principle. However, the weaknesses of the displacement transducer can be compensated by sophisticated software since the work cycle is highly repetitive and predictable in this application.

5.2. Cost

As already shown in section 3, the costs should be considered as a specific parameter in relation to the flow rate. Therefore, the primary goal should be to maximize the flow rate with as few additional costs as possible. Taking the hydraulic cylinder as an example, one should therefore try to run it as fast as possible with a high working pressure, since this results in a cylinder that is as small as possible and therefore as inexpensive as possible for the same delivery rate.

Saving costs by choosing components with low quality or a system structure with poor energy efficiency is not recommended, as these result in worse KPIs for energy efficiency and low service life, which are also important parameters of the overall product.

5.3. Energy efficiency

In addition to the powerful cooler in the compressor, which is not influenced by the drive solution, it is primarily the main control element of the movement inversion and the power control that are decisive for the energy efficiency of the drive. Of course, the pressure losses in the piping should be kept as low as possible. The friction of the cylinder seals can usually be neglected.

Due to the increased power density of compressor drives compared to average hydraulic solutions and long continuous operating periods, there is a high cooling demand, especially at low energy efficiency. This is further complicated by the fact that, in the case of strong heat input due to power dissipation, it is difficult to transport the heat evenly to the cooling unit, which can result in highly unevenly distributed temperatures in the system. As a result, large additional costs for cooling are added and even short disturbance can damage the system due to overheating. In addition to the pure energy costs, which in some cases are still of little importance, the other advantages of efficient hydraulics should therefore be considered.

As mentioned in previous sections, compressors are mostly used in multi-stage systems. When selecting the main control element, therefore, consideration should be given to how well the flow rate and power can be controlled, so that multi-stage operation, possibly even a modular system, is possible.

Movement inversion by valve

Figure 8 shows a simplified schematic with a valve-controlled compressor drive. The motor and pump operate at constant speed and thus constant volume flow since the cylinder moves at constant speed except for the change of direction. When the end position is reached, the valve is switched over to reverse the movement. Due to the constant rotation speed, the pressure of the pump depends on the load on the cylinder. If the flow rate of hydrogen is to be reduced, the volume flow and thus the cylinder speed must be reduced, for example via the motor speed.

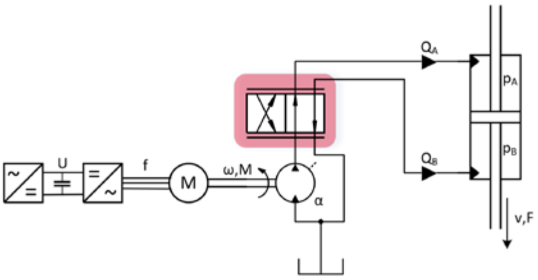

Figure 8: Simplified schematic of a valve-controlled system.

The advantages of this system structure include low complexity and low cost. Furthermore, the switching of the end position is very precise and suitable for high frequencies, which means that the flow rate of the compressor can be maximized. To evaluate the efficiency of this drive concept, the power curves of the electric motor and the hydraulic cylinder are simulated with SimulationX in Figure 9. In doing so, many assumptions must be made for the system sizing, which are not listed in detail, as they only affect the scale of the result but not the behaviour itself. An idealized behaviour of a compressor, as explained in chapter 4, was assumed as load.

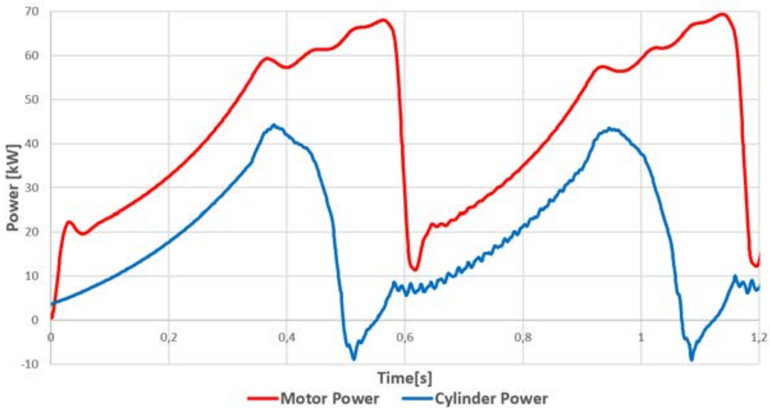

Figure 9: Motor and cylinder power in the valve-controlled system.

It can be seen that the power curves are different from each other. First, an offset during the pressure build-up can be recognized. This is primarily caused by the pressure loss at the valve and general piping losses as well as the efficiency of the motor and pump. Since the cylinder runs at a constant speed and the pressure losses depend on the flow rate, the offset is constant and is therefore more significant at low working pressures. For optimization, a valve with a low pressure loss can be selected, but a non-negligible loss always remains, otherwise the control quality is affected.

The second major deviation is that the motor power drops with a delay, meaning that the motor runs at full power for longer. This is due to the fact that the valve is briefly closed during the changeover, but the pump continues to deliver. To prevent the pressure from rising uncontrollably at this point, a pressure relief valve or hydraulic accumulator must be used. For the simulation, an accumulator was assumed which is filled while the valve is closed and keeps the pressure at an elevated level for some time after the reversal of direction.

These two basic issues lower the efficiency of the hydraulic system, in the case of the performed simulation to a value of 40%. Depending on the degree of optimization of the system and the load, this efficiency underlies a tolerance and is not constant, but without a fundamental change in the schematic, a significantly better value is not achievable.

Movement inversion by pump

In the system approach with movement reversal using the pump, the valve is not required, and the movement is carried out by the pump displacement adjustment. In this assumption, the motor rotates at a constant speed. The simplified schematic is shown in Figure 10.

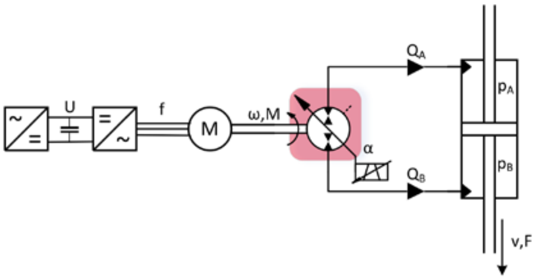

Figure 10: Simplified schematic with the pump as control component.

Compared to the valve-controlled system, the adjustment with the pump is less precise and slower. The lower precision can be compensated with suitable software, but high switching frequencies are not possible, especially with larger pumps. In order to still achieve a good flow rate, the stroke of the cylinder must be increased so that a good average piston speed is still possible.

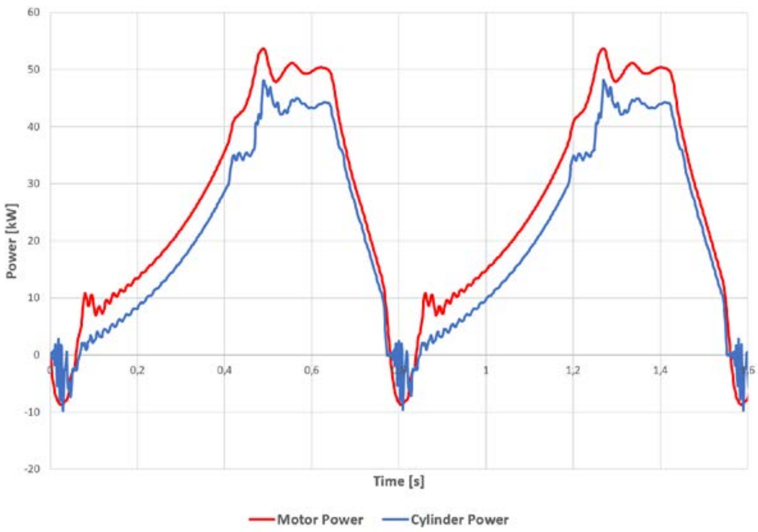

Figure 11: Motor and cylinder power in the pump-controlled system.

A system simulation was performed also for this system with the same boundary conditions as for the valve and is shown in Figure 11. It can be clearly seen that the motor power follows the cylinder power with a slight offset. The offset is due to motor and pump efficiencies and piping losses. However, since the valve from the previous section is removed, the level of offset is significantly lower. In addition, the engine power decreases equally with the cylinder power in this system structure.

Another characteristic is that the negative power reaches the pump and thus also the electric motor through the decompression of the dead volume. This can be used regeneratively with various concepts, e.g. charging a capacitor in the DC link or in the form of mass inertia of the motor shaft. However, it should be weighed up whether the additional expense is worth the energy saved. In any case, this circumstance should be taken into account and the energy should be used or dissipated in a targeted manner so that no uncontrolled over- or underpressures occur.

In the simulated example, with the same boundary conditions as in the previous section, an average efficiency of about 80% could be achieved. This increase of 40% compared to the valve-controlled system makes it clear that in many applications it can be worthwhile to invest the extra effort and cost in a variable displacement pump.

Movement inversion by motor

For the completeness, in addition to the two previous options, the adjustment with the motor should also be shown. Figure 12 shows the simplified schematic, which does not differ much from the pump-controlled system.

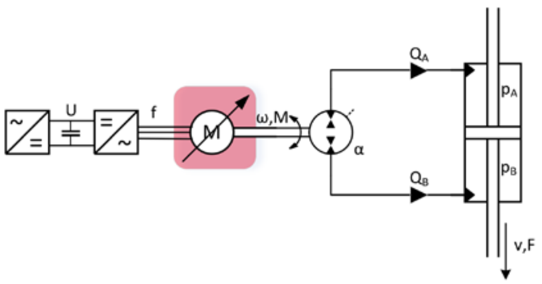

Figure 12: Simplified schematic with the pump as adjustment component.

There is a very strong similarity to the pump-controlled system in terms of advantages, disadvantages, and performance curve. The decisive difference, however, is that in the case of motion reversal, the mass inertia of the motor and pump shafts acts against the changeover. The extent to which this effect is significant depends on the dimensioning of the drive unit and can therefore be a decision criteria for or against this approach. Small drives are more suitable for changing the direction of rotation in a highly dynamic manner. It should therefore be verified for the case at hand whether this concept can be implemented. Of course, the energy stored in the rotation can also be used for the process, but this is only possible if the speed is continuously reduced towards the end of the movement. This fact therefore shows restrictions with regard to frequency maximization with this system structure, which is why it is probably only suitable in exceptional cases, like particularly small compressors with only a few kW.

Combination of movement inversion concepts

As is so often the case in technology, none of the extremes explained are the single best solution, but rather a suitable mixture of system structures that balance each other's disadvantages. Since these special solutions can create an USP, their implementation can’t be shown in detail.

As a first example, consider a valve-controlled system with a variable displacement pump. The offset due to pressure losses mentioned cannot be improved in this case, but the flow rate of the pump can be reduced during switching, when the valve is closed. With this measure, the efficiency can be significantly improved, but the advantage in cost and simplicity of the system are reduced.

Alternatively, it is possible to provide a system with variable displacement pump and variable speed motor. This makes it possible to select the most efficient operating point in terms of speed and displacement angle to maximize the efficiency of the motor-pump unit. Furthermore, the system allows the decompression energy to be stored in the form of shaft inertia without having to dissipate it in a motion reversal.

These two examples show that there is a large solution space of suitable system structures, but the characteristic advantages and disadvantages of the adjustment component for the motion reversal remain and can only be improved within certain limits.

5.4. Service Life

As a rough estimate 10,000 hours of operation between service intervals should be the target. This period of time is normally well within the capabilities of the pump, valves and motor, as long as the system is well implemented. This implies that there should be no unacceptable pressure peaks or cavitation. Likewise, the temperature and oil quality should remain within the specified limits of the individual components.

Although cylinders are known to be very robust components, a rough estimate of mileage shows why they can often be the critical component in this application. The average piston velocity $\bar{v}$ should be designed between 0.5 m/s and 1 m/s. In this range, the highest possible flow rate is achieved, but the sealing system should not yet be overloaded. With this specification and the time interval of 10,000 hours, it is estimated that 18000km to 36000km will be achieved. For industrial cylinders, 3000km is usually a rough guideline for when the sealing system might be worn out. One of the main challenges is therefore to design a sealing system that is optimized for the typical operating cycle and can thus seal off above-average mileage.

In addition to the mileage, the load cycles may also be estimated. These range from 10 million (0.3Hz) to 100 million (3Hz) between service intervals. Of course, only the seals and, if necessary, the running surfaces should be renewed during servicing and there should be no mechanical damage. This results in the requirement that all force-carrying components on the cylinder must be absolutely fatigue-proof.

6. CONCLUSION

Hydraulics are a suitable drive solution for small to medium compressors. However, to fully utilize its strengths, the physical principles of a gas compressor must be understood in addition to a deep understanding of the end application. Already when determining the system structure, there are several possibilities, which are more or less suitable depending on the target product. In the following dimensioning of the components, it is then essential to know how their properties contribute to the KPIs of the overall system. Even if this application seems quite simple at first due to the repetitive cycle, it is possible to create a great value added through specially developed components, system architectures and software, which contributes to the success of the OEM.

From the point of view of Bosch Rexroth AG, purely valve-controlled system structures are only a suitable solution in rare cases due to their low efficiency. Motor-controlled drives may be suitable for very small compressors in test technology, but a variable displacement pump will certainly be the best choice for the broad range of applications with higher flow rates. Furthermore, it is advisable to design the system together with the OEM in order to point out the specific characteristics and to benefit from the advantages of hydraulics. Particularly in the case of cylinders, the use of standard products should be avoided, as these are not suitable for the loads involved and may result in premature fails.

NOMENCLATURE

A_{piston}	Piston surface area	m^2
$c_{p.H2}$	Constant pressure heat capacity of hydrogen	$J \times kg^{-1} \times K^{-1}$
f	Movement frequency	s^{-1}
ΔH	Enthalpy difference	J
l_{stroke}	Stroke length	m
m_0	Mass intake of stroke	kg
$\dot{m}$	Mass flow	$kg \times s^{-1}$
n	Polytropic exponent	-
p	Pressure	MPa
p_{out}	Discharge pressure	MPa
p_{in}	Inlet pressure	MPa
P_{comp}	Compression power	W
P_{el}	Electrical power	W
P_{loss}	Power loss	W
P_{mech}	Mechanical power	W
$\dot{Q}$	Heat flow	W
R_{H2}	Gas constant of hydrogen	$J \times kg^{-1} \times K^{-1}$
T_0	Starting temperature of hydrogen	K
T_{out}	Discharge temperature	K
$\bar{v}$	Mean piston velocity	$m \times s^{-1}$
V	Volume	m^3
V_0	Effective stroke volume	m^3
V_{dead}	Dead volume	m^3
V_{stroke}	Total stroke volume	m^3
ΔW	Work difference	J
W_{comp}	Compression work	J
x_{drive}	Piston position	m
$\dot{x}_{drive}$	Piston velocity	$m \times s^{-1}$
ρ_{H2}	Density of hydrogen	$kg \times m^{-3}$

REFERENCES

[1] European Forum for Reciprocating Compressor (2022) Hydrogen Compression Boosting the Hydrogen Economy.

[2] NIST Applied Chemicals and Materials Devision (2018) REFPROP Database 23. United States

[3] Eifler W, Schlücker E, Spicher U, Will G (2009) Küttner Kolbenmaschinen. Wiesbaden, Germany

[4] Langenheinecke K, Kaufmann A, Langenheinecke K, Thieleke G (2020) Thermodynamik für Ingenieure. Wiesbaden, Germany