DRY SCREW COMPRESSOR PERFORMANCE AND APPLICATION RANGE

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ABSTRACT

Screw compressors have the advantage of low mechanical vibration levels similar to those found in centrifugal compressors due their rotary motion. However, screw compressors are positive displacement machines with a working principle similar to reciprocating compressors, which gives screws some advantages compared to centrifugals, such as:

- The suction volume flow and power consumption grow linearly with the compressor speed, at constant discharge pressure.
- the suction volume flow is nearly constant for variation of pressure ratio or gas molecular weight with no surging limit.
- The achievable pressure ratio per compressor stage does not depend on the gas molecular weight but is limited mostly by the allowable discharge temperature or by mechanical limits.
- By using liquid injection for cooling, pressure ratios up to 10 and in some cases higher are possible in one compressor stage.

The largest dry screw compressors built up to now have a suction volume flow of 45,300 acfm (77,000 m^3/h). Driver power may reach 9 MW. Discharge pressures up to 747 psia (51.5 bar abs) have been realized.

INTRODUCTION

The first helical rotor compressor was invented by the German Heinrich Krigar in 1878 but it did not gain industrial application at that time. In the 1930s Alfred Lysholm from the Swedish Ljungstroem Steam Turbine Company (later Swedish Rotor Maskiner, SRM) developed a dry screw compressor that was the basis for the screw compressors of today. In the 1950s several companies took licenses from SRM and started the industrial production of screw compressors. Since then the production numbers of dry screw compressors have grown due to improved rotor manufacturing methods. The maximum suction volume flow and discharge pressure have increased. Despite the successful industrial history of these machines few textbooks on screw compressors are available.

Although lots of papers have been published on the optimum design and calculation of both oil injected and dry screw compressors, many of these papers describe computerized numerical solutions to very specific questions. Unfortunately this specific approach does not help in a general understanding of screw compressor performance. The purpose of this paper is to give the compressor user an understanding of the working principle, the operating characteristics, and the application range of dry screw compressors.

BASIC DESIGN OF DRY SCREW COMPRESSORS

Screw compressors consist of two rotors that are contained in a common casing (Figure 1). Both rotors carry intermeshing helical lobes that rotate against each other with tight clearances between the rotors, and between the rotors and casing. During rotation the lobes and the casing form volumes of varying size. For dry screw compressors, synchronizing gears are used to avoid contact between the rotors. For process gas screw compressors the rotors are supported by hydrodynamic journal and thrust bearings. The rotor shafts may be sealed with a variety of shaft seal types, such as restrictive ring bushings with or without injection of seal gas or liquid, oil cooled mechanical seals, or dry gas seals.

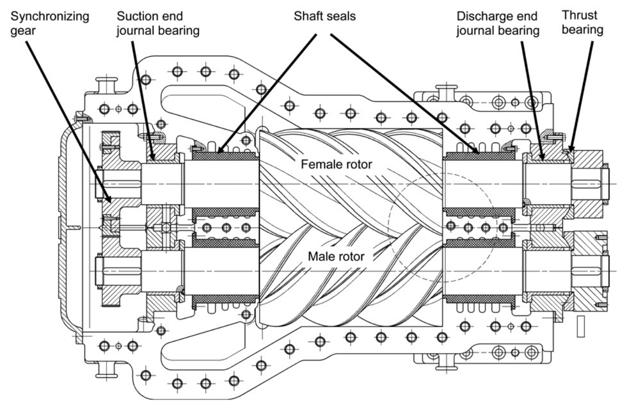

Figure 1. Horizontal Section of Dry Screw Compressor with Main Components.

Rotor Profile Considerations

The lobe profile and the number of lobes on the male and female rotor may differ for different machines. The design is influenced by many conflicting requirements such as efficiency, mechanical strength, manufacturing methods, and cost. For optimum volumetric efficiency the length of the sealing line between the rotors has to be minimized. Mechanical strength is an important issue because the rotors do not only transmit torque but are subjected to large radial and axial gas forces as well. The radial gas forces cause a deflection of the rotors and induce shaft stresses and loads on the journal bearings. The gas forces increase linearly with the pressure difference for a fixed rotor design. Therefore the pressure difference is limited by the allowable shaft stresses and the load on the bearings. The discharge pressure of a screw compressor is in most cases not limited by the design pressure of the casing but by the allowable pressure difference on the rotors. Therefore the allowable operating discharge pressure is normally less than the maximum allowable working pressure of the casing.

The swept volume per rotation should be maximized for a certain rotor diameter in order to minimize the machine size and cost. The swept volume per rotation is large for long rotors with few lobes. Unfortunately rotors with few lobes have a small root circle diameter and therefore small shaft diameters. The gas forces on a long rotor with a small shaft diameter lead to a large deflection and high shaft stress. These rotor profiles are therefore suitable for applications with low pressure difference and high volume flow. On the other hand rotors with many lobes have a large root circle that makes large shaft diameters possible and yields high mechanical strength. These rotors are well suited for high pressure applications but for a fixed outer diameter and rotor length the swept volume is less than that of a profile with fewer lobes. Cost considerations make standardization to few profile types and rotor diameters necessary.

The first rotor lobe profiles developed by SRM were symmetric consisting of four circular lobes on the male rotor and six lobes on the female rotor (4/6). In the 1970s SRM developed the asymmetric 4/6 profile. For low pressure applications a 3/5 combination is also used. Many other profiles have been suggested but for dry process screw compressors the asymmetric 4/6 profile is still the most common (Figure 2). A three-dimensional view of a typical dry screw compressor with driving gear is shown in Figure 3.

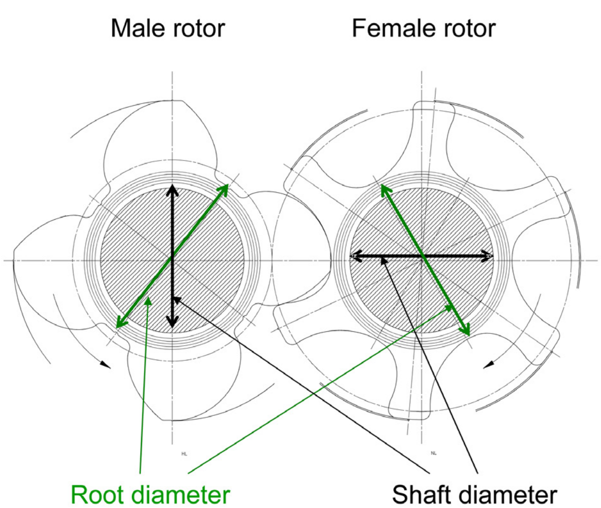

Figure 2. Rotor Cross Section for an Asymmetric 4/6 Profile.

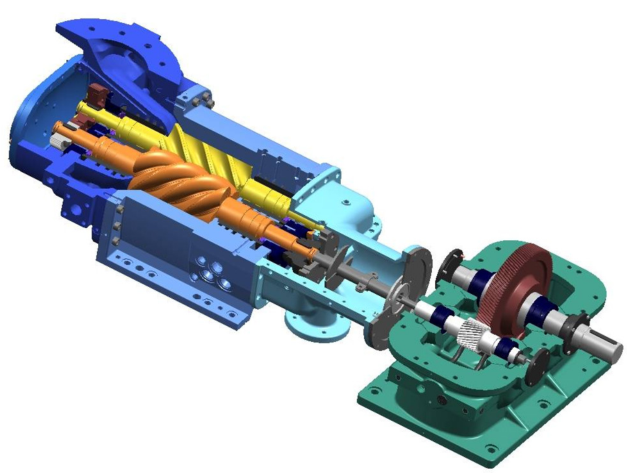

Figure 3. Three-dimensional View of Dry Screw Compressor with Driving Gearbox

WORKING PRINCIPLE OF DRY SCREW COMPRESSORS

Similar to the reciprocating compressor, the screw compressor is a positive displacement machine. The compression is repeated for each lobe of the rotors. Therefore a 4/6 profile with four lobes on the male rotor has four compression cycles during each rotation of the male rotor. Figure 4 shows the two rotors and the compression chamber from the discharge side. Successive compression phases are shown by the blue shaded area of the trapped gas. By turning of the rotors the compression chamber volume increases from zero to maximum and gas enters from the suction line. By further rotation the rotor lobes pass the inlet port boundaries and the compression chamber is closed. Now the compression phase begins. The gas is compressed by reducing the volume between the rotor lobes and the casing and simultaneously is moved axially to the discharge end. Finally the rotor lobes pass the boundaries of the outlet port. Here the compression chamber is open to the discharge line and the gas is discharged.

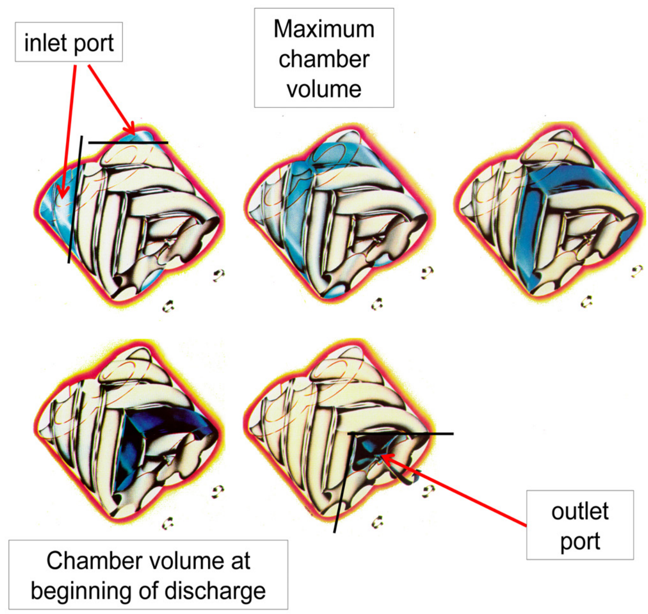

Figure 4. Screw Compressor Rotors in Successive Working Phases.

Each compression process consists of three phases: suction, compression, and discharge, which are repeated cyclically. Figure 5 shows the idealized process in a pressure-volume diagram (p-V-diagram). Unlike reciprocating compressors screw compressors do not have a dead volume that re-expands during the suction phase. During suction and discharge phases the gas volume is connected to the suction and discharge lines respectively via fixed ports in the compressor casing. During the compression phase, unlike centrifugal compressors, the gas is trapped in a closed working chamber formed by the rotors and the casing. The compression work absorbed by the gas is represented by the area between the lines of suction pressure, compression, and discharge. The compression work is given by:

$$w = \int V dp \tag{1}$$

The compression power P therefore is:

$$P = \frac{dw}{dt} = \int \dot{V} dp \tag{2}$$

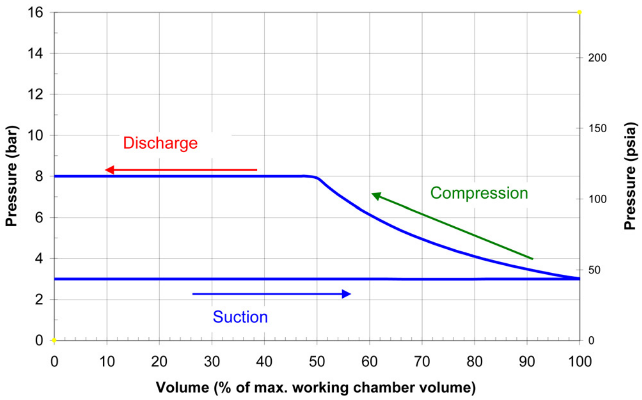

Figure 5. Idealized Pressure-Volume Diagram for a Screw Compressor with Well Suited Built-In Volume Ratio.

Unlike reciprocating compressors dry screw compressors do not have valves that open and close automatically for intake or discharge of the gas. The exchange of gas is determined by fixed openings in the casing that are related to certain angular positions of the rotors and thereby to certain volumes of the working chamber. The ratio of maximum chamber volume V_1 at the inlet to the chamber volume at beginning of opening to discharge V_2 is called the *built-in volume ratio* v_i:

$$v_i = \frac{V_1}{V_2} \tag{3}$$

The built-in volume ratio is a fixed value for a certain screw compressor because it is determined by the casing geometry. During reduction of the chamber volume from the maximum value V_1 to the volume at the beginning of opening to discharge V_2 the trapped gas is compressed. The built-in volume ratio and the adiabatic exponent k give the so-called "built-in pressure ratio" Π_i:

$$\Pi_i = v_i^k \tag{4}$$

For ideal gases $k = c_p/c_v$. Please note that Π_i is not a fixed value for a certain compressor because it depends on the k-value of the actual gas compressed. Π_i makes sense only in conjunction with the respective k-value. Thus the internal pressure at the time the working chamber opens to the discharge line p_{2i} is related to the suction pressure p_1 by:

$$p_{2i} = p_1 \bullet \Pi_i = p_1 \bullet v_i^k \tag{5}$$

For some oil injected screw compressors this value may be varied by using a slide valve but for dry screws the use of a slide valve has not gained common use. Hence for a given gas with a certain value of k dry screw compressors have a fixed "built-in pressure ratio" that may differ from the external pressure ratio $\Pi = p_2/p_1$ given by the process conditions. It is the job of the designer based on the job specification to determine an optimum "built-in pressure ratio" for the specific application. A "built-in pressure ratio" less than the external pressure ratio is called *undercompression* and a "built-in pressure ratio" larger than the external pressure ratio is called *overcompression*.

A certain mismatch of "built-in pressure ratio" and external pressure ratio is often unavoidable and does not cause a real disadvantage. Large deviations between both values, however, may have severe disadvantages such as a drop of efficiency, gas pulsations, and internal overheating for the case of overcompression.

Figure 6 shows the effect of undercompression. The suction pressure and discharge pressure is the same as in Figure 5. The working chamber is opened to the discharge while the pressure inside is still less than the discharge pressure. This causes a rapid backflow of gas from the discharge line into the working chamber until the pressures have equalized. During further rotation of the rotors, the working chamber reduces its volume and the gas is expelled into the discharge line. Undercompression causes a loss of efficiency and can cause severe gas pulsations in the discharge line.

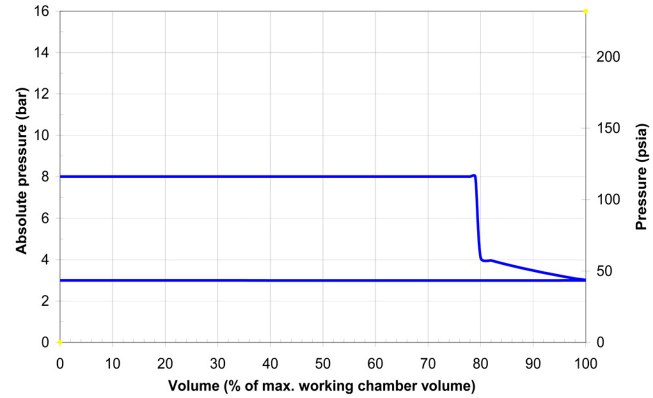

Figure 6. Idealized Pressure-Volume Diagram for a Screw Compressor with Undercompression.

The effect of overcompression is shown in Figure 7. Here also the same suction and discharge pressures as in Figures 5 and 6 are used. As can be seen the pressure inside the working chamber overshoots the discharge pressure. When the working chamber is connected to the discharge line the internal pressure drops rapidly to the discharge pressure as the gas flows into the discharge line. Overcompression leads to a loss of efficiency, because the gas is compressed unnecessarily high and expands again. Further, the rapid equalization of pressures can cause gas pulsations in the discharge line. A possible danger is overheating of the gas inside the compressor that may exceed acceptable temperatures and lead to damage even if the external pressure ratio is small. A high-temperature run-in on air or nitrogen during shop testing may not be possible for compressors with a built-in volume ratio of 2.5 or higher even at reduced pressure ratio due to high internal pressures and temperatures. If the built-in volume ratio cannot be reduced the only solution for air or nitrogen operation is liquid injection.

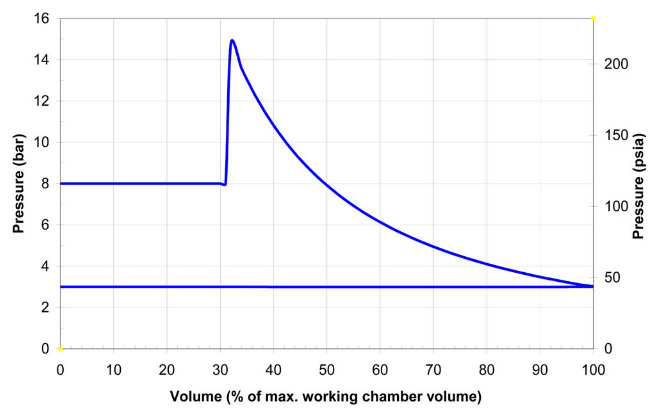

Figure 7. Idealized Pressure-Volume Diagram for a Screw Compressor with Overcompression.

Please bear in mind that Figures 5, 6, and 7 show idealized processes for educational purposes. Figure 8 shows a more realistic graph of the compression process in comparison to the idealized one.

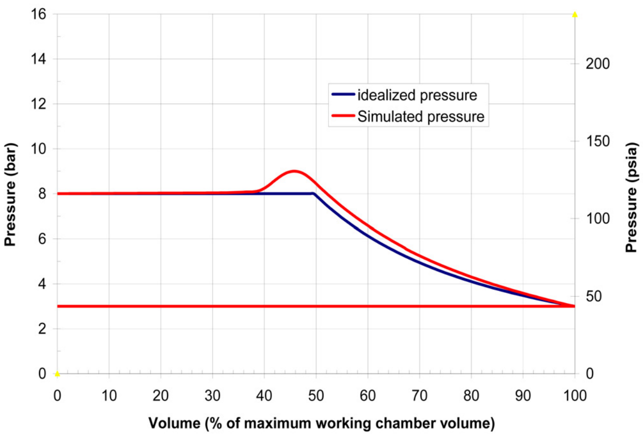

Figure 8. Realistic Pressure-Volume Diagram for a Screw Compressor.

SIMILARITY CONSIDERATIONS

Screw compressors are built in different sizes but with the same basic design. Most manufacturers build one or more series of geometrically similar screw compressors differing in rotor diameter and center distance between the two rotors. Mechanical and thermodynamic similarity considerations show that geometrically similar compressors of different sizes and rotational speeds can be compared using the rotor tip speed in ft/sec (m/s). Compressors of different sizes operate within the same rotor tip speed range and thus can be compared easily. As compressor sizes may vary between rotor diameters of 4 inches (102 mm) and approximately 32 inches (816 mm), the rotational speed in rpm differs widely while the rotor tip speed is in the same range. Suction volume flow and power consumption of geometrical similar compressors running at the same rotor tip speed change proportionally to the square of the rotor diameter.

CHANGE OF OPERATING CONDITIONS

Speed Variation

Figure 9 shows the suction volume flow versus rotor tip speed for two compressors of different sizes and constant suction and discharge pressure. The suction volume flow changes linearly with speed. In Figure 10 the power consumption versus rotor tip speed is shown. The power consumption increases nearly linearly with speed, which means that the torque is nearly constant for constant pressures. *Very simply, one can state: 50 percent speed gives 50 percent suction flow and saves 50 percent power. Therefore speed variation is a very good method of process control.*

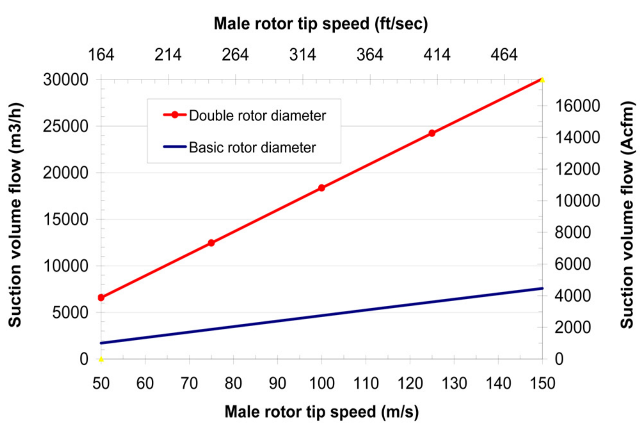

Figure 9. Suction Volume Flow Versus Rotor Tip Speed for Two Compressors Compressing Natural Gas.

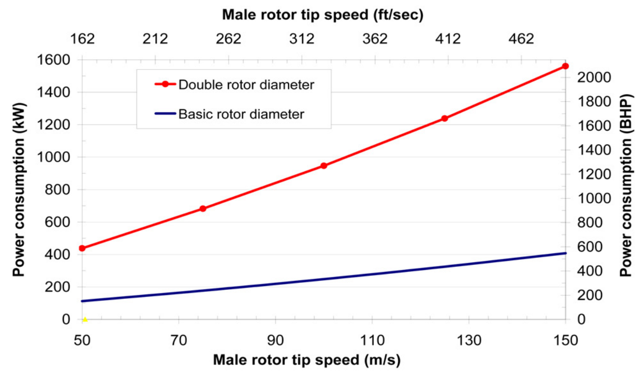

Figure 10. Power Consumption Versus Rotor Tip Speed for Two Compressors Compressing Natural Gas.

Typically the male rotor tip speed of a dry screw compressor is in the range of 164 to 492 ft/sec (50 to 150 m/s). Due to their stiff shaft design screw compressors always run at speeds less than first lateral critical speed. Passing the critical speed would excite vibrations that would cause rotor to rotor contact or rotor to casing contact due to the small clearances. Therefore the upper limit for speed variation is given by the first lateral critical speed of the rotors including a safety margin. The lower limit cannot be given as a fixed value but is normally given by a drop of efficiency and an increase in discharge temperature.

Variation of Pressure Ratio

The following variations of suction pressure or discharge pressure are all considered for a machine of fixed size running at constant speed. Therefore a change in power consumption means a proportional change in torque.

Variation of Discharge Pressure

Figure 11 shows the suction volume flow versus discharge pressure. Suction pressure, and other properties are kept constant. As can be seen the suction volume flow drops slightly with increasing discharge pressure, i.e., increasing pressure ratio. Because screw compressors do not have a surge line, the limitation in pressure ratio is given by discharge temperature, mechanical limits (bearing load and shaft stress), over- or undercompression, and efficiency drop.

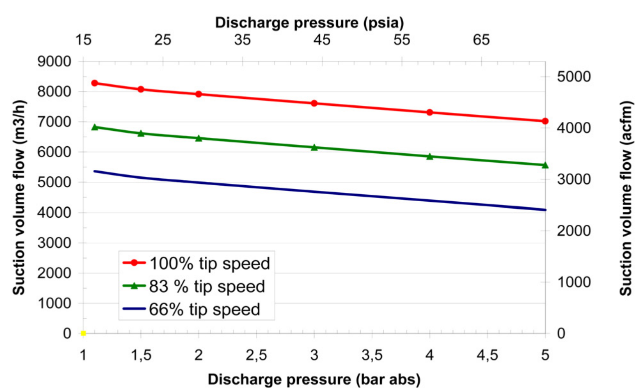

Figure 11. Suction Volume Flow Versus Discharge Pressure for Constant Suction Pressure.

Figure 12 shows the power consumption versus discharge pressure for different built-in volume ratios v_i. As the suction pressure is kept constant the same characteristic is valid if one uses the pressure ratio instead of the discharge pressure on the right-hand axis. For constant suction pressure the power consumption increases linearly with the discharge pressure. For a small v_i the slope of the power curve is steep resulting in a rapid growth of power consumption with increasing pressure ratio. For a large v_i the slope is much less than for a small v_i, but on the other hand the no-load power consumption at pressure ratio 1 (discharge pressure = suction pressure) is high. Therefore compressors with large built-in volume ratio are well suited for large pressure ratios but have a high torque even during unloaded operation.

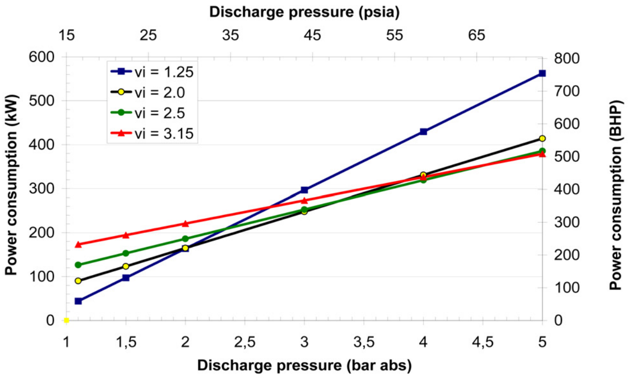

Figure 12. Power Consumption Versus Discharge Pressure for Constant Suction Pressure.

Variation of Suction Pressure

For constant speed and discharge pressure a reduction in suction pressure leads to an increase in pressure ratio. The suction volume does not change much as a result. The general rule is: *A change of pressure ratio (caused by either suction pressure or discharge pressure) does not have a major effect on the suction volume flow.*

Figure 13 shows the power consumption versus suction pressure for four different built-in volume ratios v_i. For a built-in volume ratio of approximately 2.2 to 2.4 the power consumption is constant even for varying suction pressure. For $v_i < 2.2$ the power consumption increases with decreasing suction pressure (i.e., with increasing pressure ratio and increasing pressure difference).

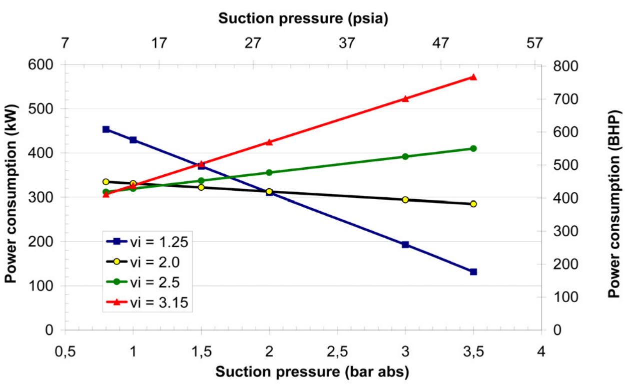

Figure 13. Power Consumption Versus Suction Pressure for Constant Discharge Pressure.

For $v_i > 2.4$ the power consumption increases with increasing suction pressure (i.e., with decreasing pressure ratio and decreasing pressure difference). This puzzling effect is due to the internal overcompression that may occur at large v_i and small pressure ratio. The growth in power consumption also means an increase in torque. This effect is important for operation of compressors with large v_i that may have an increased settle-out pressure. A high settle-out pressure leads to high torque during restart. An excessively high settle-out pressure may even overtorque parts of the machine train.

Variation of Gas Properties

The main thermodynamic property for the compressed gases is the molecular weight. Gases compressed in dry screw compressors range from hydrogen, natural gas, air, hydrocarbons, CO_2, butadiene up to heavy gases with a molecular weight of 100 kg/kmol.

The molecular weight and pressure ratio determine the optimum tip speed. For heavy gases or low pressure ratio the optimum tip speed is low, for light gases or high pressure ratio the optimum tip speed is high. Compressors operating with different gases are compared using the circumferential Mach number at the inlet conditions of the gas. Typical circumferential Mach numbers range between 0.2 and 0.4 although higher or lower values may be possible for some applications.

In most cases the efficiency curve versus tip speed is rather flat. If the discharge temperature can be limited by liquid injection, then capacity control via speed variation with small efficiency losses is possible even for very different gases. Figure 14 shows the adiabatic efficiency versus the rotor tip speed for coke oven gas with molecular weight 11 kg/kmol and for lime kiln gas with molecular weight 44 kg/kmol.

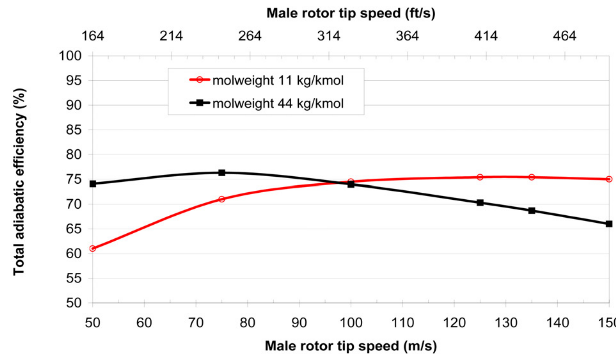

Figure 14. Adiabatic Efficiency Versus Rotor Tip Speed for Gases with Different Mole Weight.

Figures 15 and 16 show the effect of different molecular weights on the performance of two dry screw compressors. The data have been calculated for a tip speed of 393.7 ft/sec (120 m/s) with a pressure ratio of 3 and a built-in volume ratio of 2.2. The gases in this example range from a mixture of 50 percent hydrogen and 50 percent methane with a molecular weight of 9 kg/kmol up to CO_2 with a molecular weight of 44 kg/kmol. Figure 15 demonstrates that the suction volume flow does not change very much within a large range of molecular weights. Only for very low molecular weights the decrease of volumetric efficiency and suction volume flow is more pronounced. Figure 16 shows that the power consumption also does not change very much for a change in molecular weight. Because Figure 16 has been calculated for constant speed the torque is proportional to the power consumption and therefore does not change much with changing molecular weight. For very low molecular weights the drop in volume flow is more pronounced than the drop in power consumption thus leading to an increase in discharge temperature. If the discharge temperature can be limited by liquid injection even larger variations of molecular weight are possible. The final limitations must always be determined on a case by case basis.

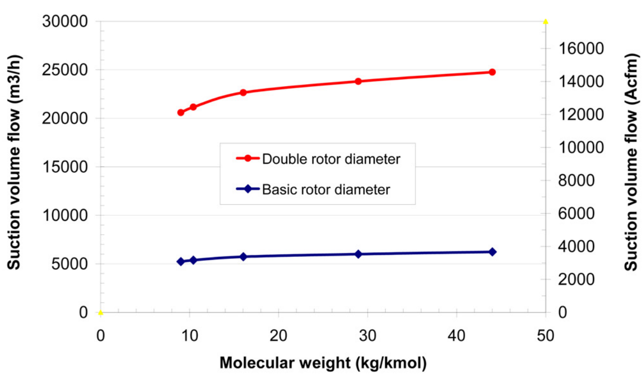

Figure 15. Suction Volume Flow Versus Mole Weight for Constant Speed and Constant Pressures.

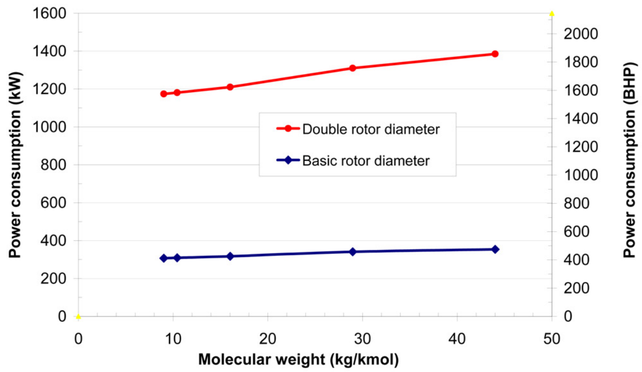

Figure 16. Power Consumption Versus Mole Weight for Constant Speed and Constant Pressures.

The second important gas property is the adiabatic exponent k. A high adiabatic exponent leads to high discharge temperatures. If the discharge temperature exceeds the design limits of the compressor, then an injection of liquid into the suction nozzle may be used to cool the gas. The most common injection liquid is demineralized water but other liquids like methanol or heavy hydrocarbons may also be used if they are compatible with the process. Part or all of the injected liquid evaporates and thereby cools the compressed gas and the compressor. The screw compressor tolerates relatively large amounts of liquid injection. Especially for dirty gases like coke oven gas or lime kiln gas large amounts of water are injected and even on the discharge side a large part of the water is still liquid. Here the injection is used not only for cooling but also for washing the compressor and the gas.

SPECIAL OPERATING CONSIDERATIONS

Dry Screw Compressor Systems

A simplified process diagram for a two-stage dry screw compressor is shown in Figure 17. A large unloading valve is used for starting and stopping the screw compressor by reducing the pressure difference and lowering torque. Even for speed variable machines a bypass control line is needed if zero flow is required. In most cases a gas cooler between the compressor stages is necessary in order to reduce the suction temperature for the second stage. An additional gas cooler downstream the second stage may be required for process reasons, but is not necessary for the screw compressor itself. If individual bypass control lines are used around both stages, a control of the interstage pressure is possible. This may be necessary for compressors with large variations in suction and/or discharge pressure in order to counter over- or undercompression effects.

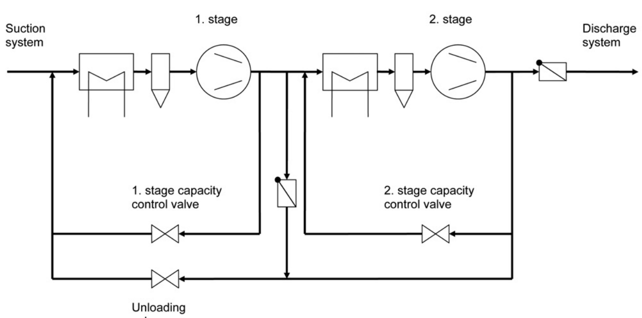

Figure 17. Simplified Two-Stage Screw Compressor Process Diagram.

Gas Pulsations

The cyclic nature of the working principle of the screw compressor causes gas pulsations that emit noise from the piping. This is especially important for dry screw compressors with high discharge pressures and/or with varying operating conditions. Therefore it is very important to apply properly designed suction and discharge silencers on dry screw compressor sets. The discharge silencer is especially important because most of the noise is generated by the gas pulsations in the discharge. An unsuitable discharge silencer design may even lead to mechanical damage of the silencer or piping internals.

The frequency of the working process is called pocket passing frequency and is calculated by male rotor frequency multiplied by the number of lobes on the male rotor. The gas pulsations occur mainly at pocket passing frequency and its multiples (typically 4, 8, and 12 times male rotor frequency for a 4/6 rotor profile). Depending on compressor size and application the male rotor speed may range from approximately 1500 rpm up to 25,000 rpm, The corresponding pocket passing frequency range is 6000 cpm (100 Hz) to 100,000 cpm (1667 Hz). Compared to reciprocating compressors these frequencies are much higher and the relevant sound wavelengths much shorter.

The gas pulsations are excited by the difference between internal compression and the pressure in the discharge line. The pulsation excitation increases with gas density and with compressor speed. The gas pulsations may cause mechanical vibrations of the compressor, silencer, and piping. The expertise of the compressor manufacturer should be sought for the silencer design. The silencer must be individually designed for each compressor. Figure 18 shows an example for a discharge silencer system consisting of three parts: a venturi nozzle, a damping plate, and a downstream silencer. The venturi nozzle is a massive casting flanged directly to the compressor discharge. The venturi nozzle reduces the gas pulsations by first accelerating and then decelerating the gas velocity. Downstream the venturi nozzle a damping plate is placed. This is a massive metal plate with a number of drilled holes through which the gas passes. The free area of the holes is determined by the gas molecular weight, gas mass flow, and the process conditions. The damping plate reduces the pulsations further and isolates the downstream silencer and piping from the gas pulsations emitted from the compressor discharge. Downstream the damping plate a silencer is placed. The silencer may be working by absorption and/or resonance. The typical pressure drop at design conditions for such a silencer system is 2 percent of the discharge pressure. In most cases it will be necessary to add a noise enclosure around the compressor and gear as well.

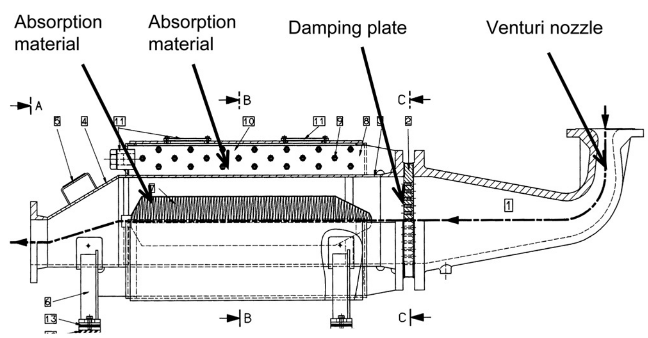

Figure 18. Example of Discharge Silencer Design.

Starting of Dry Screw Compressors

Screw compressors always start unloaded, i.e., with discharge pressure = suction pressure. The discharge system is separated from the compressor by a check valve. The gas is recycled around the compressor through a large unloading valve with a pressure drop of approximately 7 psi (0.5 bar). After the compressor has reached its minimum speed the unloading valve is closed and the pressure in the discharge line up to the check valve rises. The run-up time with open unloading valve should not exceed 30 seconds. When the pressure in the compressor discharge line exceeds the pressure in the downstream system the check valve opens and the compressor begins to deliver into the downstream system. Prolonged operation at discharge pressure very close to suction pressure must be avoided due to the gas pulsations in the discharge line. Caution: During closing of the unloading valve the compressor rapidly sucks gas out of the suction line. If the volume in the suction line is too small, or if insufficient gas can be supplied by the upstream system, the suction pressure drops rapidly and the compressor will be tripped. Therefore a large volume of the suction line is helpful to ensure smooth starting of the compressor.

Stopping of Dry Screw Compressors

Screw compressors must be unloaded during shutdown. Upon stop signal of the driver, the unloading valve immediately opens (opening time should be less than 2 seconds) and the gas is recycled around the compressor through the unloading valve. The pressure in the discharge line between compressor and check valve drops rapidly and the check valve closes thus separating the compressor from the downstream system. As the gas is recycled around the compressor the suction pressure may rise. During rundown of the compressor to standstill, the suction and discharge pressures equalize at the settle-out pressure. The run down time depends on machine size and operating conditions. Typical rundown times may range from 5 seconds for small compressors in high pressure applications to 1 minute for large compressors in low pressure applications.

Caution: If the settle-out pressure is much higher than the normal suction pressure this may lead to high torques for machines with large built-in volume ratio. Also the correct operation of the seals may be endangered because the seal supply pressure may be insufficient or the control system may be to slow for the quick rise of suction pressure. Therefore a high settle-out pressure must be avoided by using the following rules:

- Volumes at high pressure (interstage volume and volume between the discharge nozzle and check valve) should be minimized.
- The volume of the suction line should be as large as possible thus giving the recycled gas space to escape.
- Check valves in the suction line close to the compressor must be avoided. If a block valve is placed in the suction line it should be closed only after the compressor has come to standstill.

If all efforts to increase the suction volume have been exhausted the settle-out pressure can be reduced by blowing off gas from the discharge line to an independent low pressure system (e.g., a flare) during stopping. After the required maximum settle-out pressure is reached the blow-off valve may be closed again.

APPLICATION RANGE OF DRY SCREW COMPRESSORS

Figure 19 gives a very rough indication of the flow and pressure range of dry screw compressors. The volume flow of dry screw compressors ranges between that of reciprocating compressors and of centrifugals.

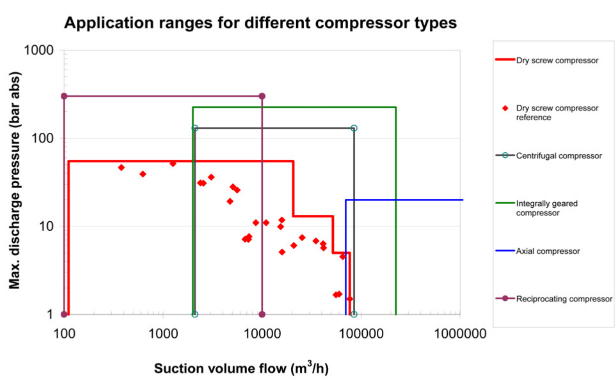

Figure 19. Application Ranges for Different Compressor Types.

Dry screw compressors are built with rotor diameters ranging from 4 inches (102 mm) to approximately 32 inches (816 mm). Typical rotor tip speeds for dry screw compressors range from 164 ft/sec (50 m/sec) up to and in some cases exceeding 492 ft/sec (150 m/s). Oil injected screw compressors run at much lower tip speeds (typically one-third of these values) due to the viscous losses of the oil. Therefore a dry screw compressor normally has a much larger capacity than an oil-injected screw compressor of the same size. Dry screw compressors have been built with gas flows from 177 acfm (300 m^3/h) up to 45,300 acfm (77,000 m^3/h).

Table 1 shows the design data for a large screw compressor with a capacity of 45,300 acfm (77,000 m^3/h). Figure 20 shows the assembly of this compressor stage with a rotor diameter of 32 inches (816 mm). The compressor package of this machine is shown in Figure 21.

Table 1. Design Data for a Dry Screw Compressor with Suction Volume Flow 45,300 ACFM (77,000 m^3/h)

	US units		**SI units**	
Molecular weight	15.6	lb/lbmol	15.6	kg/kmol
Suction pressure	3.2	psia	0.22	bar
Discharge pressure	21.8	psia	1.5	bar
Suction volume flow	45321	acfm	77000	m3/h
Rotor diameter	32.1	inch	816	mm
Power consumption	2319	BHP	1729	kW

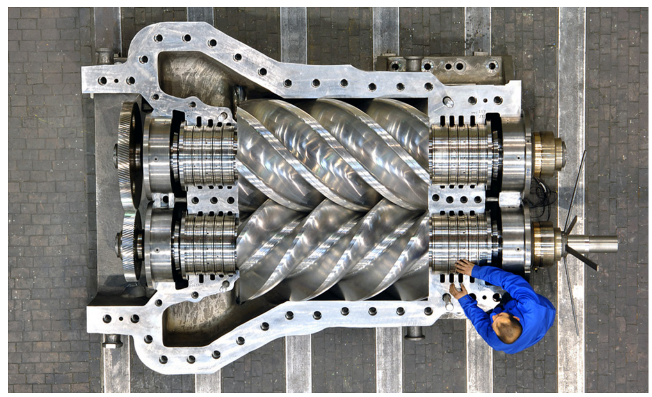

Figure 20. Assembly of a Dry Screw Compressor with Rotor Diameter 32 Inches (816 MM).

Figure 21. Single-Stage Screw Compressor for a Suction Volume Flow 45,320 ACFM (77,000 m^3/h) Driven by a Steam Turbine.

Since 2001 a dry screw compressor has been running successfully in a gasoline desulfurization unit with a discharge pressure of 747 psia (51.5 bar abs). The design data for this compressor are given in Table 2.

Suction temperatures down to −153°F (−103°C) have been realized in ethylene boil-off applications. Discharge temperatures may reach 482°F (250°C).

Table 2. Design Data for a Dry Screw Compressor with Discharge Pressure 747 PSIA (51.5 bar).

	US units		SI units	
Molecular weight	4.1	lb/lbmol	4.1	kg/kmol
Suction pressure	515	psia	35.5	bar
Discharge pressure	747	psia	51.5	bar
Suction volume flow	699	acfm	1187	m3/h
Rotor diameter	6.4	inch	163	mm
Power consumption	908	BHP	677	kW

Dry screw compressors may be built as one-stage, two-stage, or in some cases even three-stage compressor sets driven by one common driver. Drivers are normally electric motors (fixed speed or variable speed) or steam turbines. In order to design the compressor for optimum tip speed a gearbox should be used between compressor and driver. Figure 22 shows a typical two-stage design. Both stages are driven by an individual pinion thus enabling the designer to choose optimum speed for each stage.

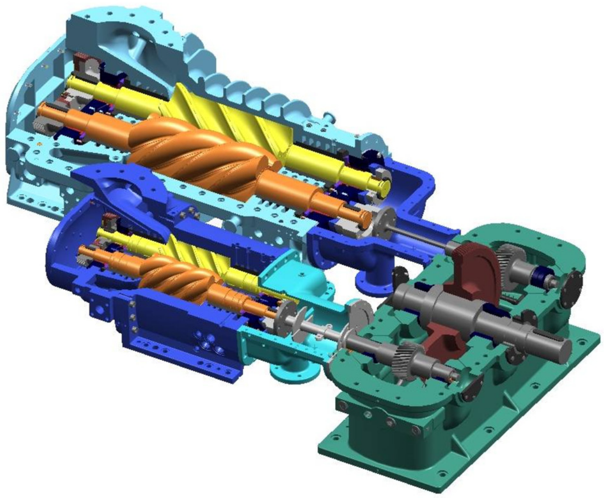

Figure 22. Two-Stage Arrangement with Individual Pinions for Optimum Speed of Both Stages.

To the best of the authors knowledge the largest screw compressor worldwide consists of two stages with 32 inches (816 mm) rotor diameter in the first stage and 16 inches (408 mm) in the second stage. This compressor is driven by a 9 MW steam turbine and compresses 38,500 acfm (65300 m^3/h) hydrogen mix gas from 15 psia (1.1 bar) to 167 psia (11.5 bar). Table 3 shows the design data for this compressor.

Table 3. Design Data for a Two-Stage Screw Compressor with 9 MW Steam Turbine Drive.

1. stage

	US units		SI units	
Molecular weight	13.5	lb/lbmol	13.5	kg/kmol
Suction pressure	16.0	psia	1.1	bar
Discharge pressure	65	psia	4.5	bar
Suction volume flow	38453	acfm	65331	m3/h
Rotor diameter	32.1	inch	816	mm
Power consumption	6216	BHP	4635	kW

2. stage

	US units		SI units	
Molecular weight	13.3	lb/lbmol	13.3	kg/kmol
Suction pressure	63.8	psia	4.4	bar
Discharge pressure	167	psia	11.5	bar
Suction volume flow	9327	acfm	15846	m3/h
Rotor diameter	16.1	inch	408	mm
Power consumption	7790	BHP	5809	kW

Dry screw compressors are often used for operating conditions with varying molecular weight or large pressure ratios (up to 10 in one stage). The insensitivity to droplets or dust in the gas and the possibility for liquid injection make dry screw compressors the preferred option for rugged operation conditions. Screw compressors are used in a variety of industries and processes, e.g., in the oil and gas industry, refineries, chemical industry, for flare gas recovery, in steel mills, soda ash plants, for compressing coke oven gas, in H_2S applications, and others. Dry screw compressors are operating successfully in offshore applications as low pressure stages for gas recompression as well.

Figure 23 shows two small two-stage screw compressor packages at different completion phases. These are machines designed for an offshore application.

Figure 23. Assembly of Two Two-Stage Screw Compressor Sets for an Offshore Application.

NOMENCLATURE

Note: all pressures must be expressed as absolute pressures

c_p	= Specific heat capacity at constant pressure
c_v	= Specific heat capacity at constant volume
k	= Adiabatic exponent (also isentropic exponent)
P	= Compression power
p	= Pressure
p_1	= Suction pressure
p_2	= Discharge pressure
p_{2i}	= Pressure inside the working chamber at the beginning of opening to discharge line
V	= Volume
$\dot{V}$	= Volume flow
V_1	= Maximum volume of working chamber
V_2	= Volume of working chamber at the beginning of opening to discharge line
v_i	= Built-in volume ratio V_1/V_2
w	= Compression work
Π	= External pressure ratio p_2/p_1 given by the suction and discharge pressure
Π_I	= "Built-in pressure ratio"

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