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Piston compressors driven by isobaric expansion heat engines

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Abstract

Compression of gases or vapors in reciprocating compressors involves multiple energy conversion steps. These steps include converting of energy of different fluids (steam, combustion products, water or wind) to shaft power, generation of electricity, feeding the electricity to an electric motor through electrical networks, transformers, frequency converters, and eventually driving the compressor kinematics. Direct compression of a gas or vapor by another fluid (referred to as vapor) is an attractive alternative to the current compression methods. Besides the elimination of the multiple energy conversion steps and associated equipment this old compression concept permits to use sustainable, abundantly available low grade heat sources instead of fossil fuels. Isobaric expansion heat engines offer an interesting opportunity for efficient vapor driven compression processes without the use of electricity. The purpose of this paper is to assess the feasibility of this method, in which compressor piston is directly actuated by a vapor of arbitrary parameters. Thermodynamic analysis performed shows that the efficiency of vapor use in the simplest compressor schemes applied in the past is inherently low. In order to eliminate this drawback different new vapor driven compressor configurations are proposed and evaluated using the same approach. These configurations include multistage vapor driven compression, reuse of the driving vapor and the use of force transmission between the compressor and driver pistons. It is shown that the proposed configurations can significantly improve the efficiency of vapor use and can be a valuable alternative to existing compression technologies.

Keywords: Compression, heat-driven compressor, isobaric expansion, renewable energy, low-grade heat.

Introduction

Energy consumption by compressors is substantial. For instance, compressed air systems alone account for 10% of industrial electricity consumption in the USA and the EU [1, 2]. An increase in electricity demand from residential and commercial air conditioning by 2050 is expected to be comparable to adding the European Union's entire electricity consumption. Air conditioning will represent 12.7% of electricity demand by the middle of the century, compared to almost 9% now [3]. Therefore, any improvements in the cost and efficiency of compression processes are of great interest.

Reciprocating, or piston compressors, are the best known and most widely used compressors of the positive displacement type [4]. Although piston compressors are simple in principle, all the used arrangements include massive and expensive kinematic components (piston, piston rod, crosshead, crankshaft, crankcase, flywheel, and gearbox).

Most piston compressors are driven by electric motors. Different heat engines (steam or gas turbines, piston engines) are also used as drivers of reciprocating compressors [4, 5, 6]. Generally, the heat engines convert the energy of different fluids (steam, combustion products, water or wind) to shaft power. This shaft power, in turn, is converted into electricity, which is fed to the electric motor through electrical networks, transformers, frequency converters, etc. The electric motor drives the compressor kinematics which performs work on the compressed fluid (gas or vapor). Ultimately, the energy of a primary fluid is transferred to the energy of the compressed fluid in such a complex multistage way.

There are also compression technologies in which one fluid is compressed directly by another one. Gas (liquid-gas) ejectors [7], wave pressure exchangers [8], free piston diesel compressors [9, 10, 11, 12] and steam compressors [6] are among them. These methods eliminate kinematic transmission and multiple energy conversion steps. However, the efficiencies of gas ejectors and wave pressure exchangers are rather low. Free-piston diesel compressors proved to possess some very attractive features. However, they did not gain widespread commercial success. No reports of serious lacks or flaws in the concept explaining this can be found [13].

A direct steam-driven air compressor is one of the earliest compressor types. It is actuated by a steam engine of the reciprocating type with a direct connection between the piston of the compressor and that of the steam engine so that both pistons move as a unit [6]. The idea of steam-driven compressors goes back to the first heat engines invented by Savery and Newcomen for water pumping more than 300 years ago [14, 15, 16]. Probably the first steam driven compressors were blast furnace bellows for the production of lead copper and iron in the mid-18th century [17].

Steam-driven compressors are similar to steam pumps although the processes are different, and in contrast to steam pumps they are almost unknown [5]. To date, no application has been found for this compression technology. The last mention of operating steam driven air compressors we found is a compressor instruction pamphlet from the Westinghouse Air Brake Company [18] that is 100 years old.

The practical implementation of the concept of steam or (more generally heat driven) reciprocating compressor depends on the availability of an energy efficient heat engine. So-called isobaric expansion (IE) engines, which directly convert heat to mechanical energy in the form of a high pressure fluid (liquid or vapor) have the potential to revive the steam-driven compression concept. The isobaric expansion technology was introduced in [19], although it was previously studied under different names and for various applications [20, 21, 22, 23, 24, 25]. With their hydraulic or pneumatic power output, the IE engines are ideal for pumping and compression applications. The impact of this method can be significant because, in addition to eliminating multiple energy conversions and associated equipment, they allow replacing primary fossil energy sources with abundantly available low-grade heat, even at temperatures below 100 ^{o}C.

The energy efficiency of heat driven IE engine-pumps has been studied both theoretically and experimentally [19, 26, 27]. However, according to our knowledge, studies of compression processes in which a compressible fluid acts as an actuating agent are absent in the literature; the interesting system proposed in [28] has not been evaluated properly as the used assumptions violate both momentum and energy conservation. Recent publications on this topic, [29, 30], do not differentiate between compression and pumping, which can lead to gross errors.

The purpose of this paper is to assess the feasibility of a direct transfer of energy from one compressible fluid to another one as it occurs in the steam-driven reciprocating piston compressor. In the processes under consideration, the compressor piston is driven directly by the piston of a heat engine (driver) using a working fluid of arbitrary parameters. Any fluid (gas, liquid, supercritical fluid or fluid undergoing phase transition) can be considered as the actuating or driving agent. In the following part of the present work, we will use the terms vapor or driving fluid to refer to all possible fluids.

The question of how the driving vapor is generated (i.e. heat engine process) is not considered. Therefore, this analysis is equivalent to the analysis of the energy efficiency of a steam-driven compressor operating on an open cycle. In the case of a closed cycle, the overall process efficiency is also determined by the engine process which is beyond the scope of the present work.

A thermodynamic analysis of the efficiency of vapor use (defined as work performed by the driver per unit mass of vapor) is performed. At first, the simplest compressor schemes are considered. In these schemes the compressor piston is rigidly connected to another piston accommodated in a cylinder, called in this paper driver. It is found that in such schemes the direct transfer of energy from one fluid to another is inherently inefficient since the energy of the driving vapor is largely spent on the compression of the driving vapor itself.

In order to eliminate the drawback of simple vapor driven compressors different new vapor driven compressor configurations are proposed and evaluated using the same approach. These configurations include multistage vapor driven compression, reuse of the driving vapor and the use of force transmission between the compressor and driver pistons.

Basic vapor driven compressors

Two basic schemes of single- and double-acting direct vapor driven compressors are shown in Figure 1. They consist of a compressor cylinder and a driver cylinder provided with pistons that are coupled by a connecting rod. The driver cylinder is combined with a vapor generation circuit (shown in the single acting scheme) consisting of a heater **H**, recuperator **R**, cooler **C** and feed pump **P**. The driver together with the vapor circuit forms an IE heat engine. Its operation is described in [19]. The heat engine replaces the electric motor or internal combustion engine and the crank mechanism of the conventional piston reciprocating compressors.

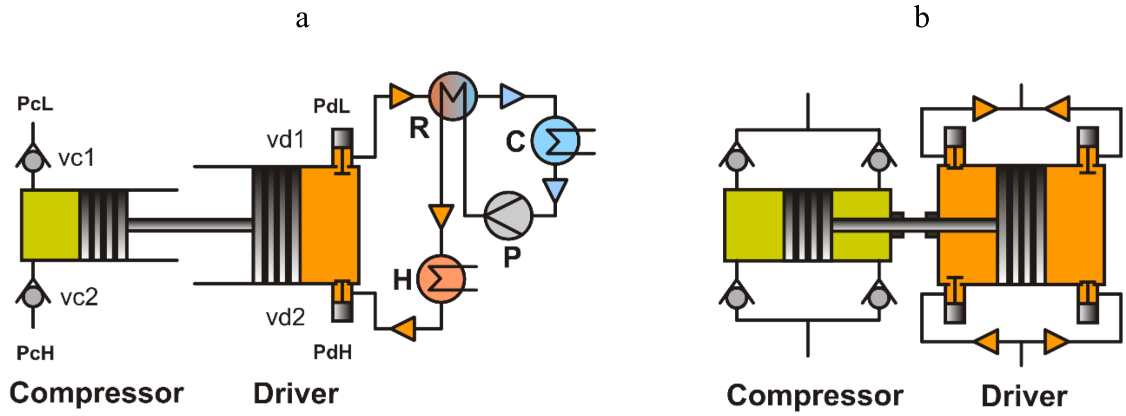

Figure 1. Basic schemes of vapor-driven compressors: a – single acting; b – double acting.

In the case of the single-acting unit (Figure 1a) both the compression and actuation occur on one side of the compressor and driver pistons. In the double-acting type unit (Figure 1b) compression and actuation take place on both sides of the pistons. In the case of the single-acting unit, the pressures on the outer surfaces of the pistons are usually constant ambient pressure, whereas for the double-acting system they are intake and discharge pressures of the compressor and driver, respectively.

The pistons are free in the sense that their movement is controlled only by the fluid forces acting upon them. The driver piston is actuated by vapor generated in the vapor circuit. The reciprocating motion of the driver piston in each scheme is transformed into the reciprocating motion of the compressor piston.

The processes in the compressor and driver can be explained for the single acting scheme, Figure 1a, as follows. Let us assume that initially the compressor cylinder is filled with the compressed fluid at its intake or low pressure P_{cL} and the volume of the cylinder is maximal V_{c0}. During the compression step the discharge valve of the driver vd2 is closed; the driving fluid is supplied to the driver through the intake valve vd1. The pressure of the driving fluid is transmitted through the driver piston and connecting rod to the compressor piston. As a result pressure in the compressor increases from the initial/intake or low pressure P_{cL} to the final/discharge or high pressure P_{cH} and then the compressed fluid under influence of the driving fluid at its maximum pressure P_{dH} is discharged from the compressor through the outlet check valve vc2 at constant high pressure P_{cH}. After that the inlet valve of the driver vd1 closes, the outlet valve vd2 opens and pressure of the driving fluid in the driver drops from P_{dH} to the low driver pressure P_{dL}. Due to equalization of the forces acting on the pistons, they move a little to the right so that a small amount of the compressed fluid remaining in the compressor (mainly in the intake and discharge valves) expands and its pressure decreases to the intake pressure P_{cL}; simultaneously part of the driving fluid is pushed out of the driver through the discharge valve vd2. After that the compressor intake stroke begins during which pressure in the compressor remains constant. Simultaneously the rest of the driving fluid discharges from the cylinder through the valve vd2 at constant pressure P_{dL}. At the end of the compressor intake and the driver discharge stroke the system returns to the initial state and the system is ready for the cycle to be repeated.

As in the case of steam piston engines the crucial problem is to find a method to supply the driving vapor to the power cylinder (driver), providing the highest work. If the pistons of the compressor and driver are coupled by a mechanism such as a crank gear with a massive flywheel acting as energy storage, the solution is well known: a certain amount of steam is to be injected at high pressure during an initial stage of piston movement, after which the intake valve closes and the supplied steam expands further adiabatically. However, this method will not be satisfactory in the case of low-inertia, directly coupled pistons (low mass or low reciprocation frequency), as shown in Figure 1. A large difference in the forces acting on the compressor and driver pistons, especially at the beginning of the compression stroke, results in an uncontrollable acceleration of the pair of pistons, fluid to be compressed and excessive consumption of the driving vapor. Therefore, toto avoid detrimental piston acceleration, the pressure change in the driver should correspond to the variable pressure in the compressor.

Ideally, the pistons move without acceleration (except for the top and bottom dead points). Technically, such an operation can be achieved if the characteristic time of the pressure rise in the driver is lower than the characteristic time of the force equalization in the compressor and driver. In addition, appropriate sizes of the compressor and driver pistons and the high and low pressures of the driving fluid should be selected for the compression process.

For the double-acting compressor-driver combination forward and back strokes of the pistons are identical. In this case, taking into account the forces acting on the pistons, it is possible to obtain a relation between the areas of the pistons and the pressures that allow the compression cycle to be carried out:

$$\frac{A_c}{A_d} = \frac{P_{dH} - P_{dL}}{P_{cH} - P_{cL}} \tag{1}$$

The maximum pressure of the driving fluid P_{dH} can be higher than that obtained from Eq. 1. However, this is not justified from the energy efficiency standpoint in view of unnecessary consumption of the driving fluid at the end of the compression stroke. The minimum pressure of the driving fluid P_{dL} can be lower than that obtained from Eq. 1, however, such an operation would require a controlled discharge valve of the driver to maintain the driving fluid pressure at a level that is necessary for uniform movement of the pistons.

A remarkable feature of a double-acting unit is that for any given values of low and upper pressures in the compressor P_{cL} and P_{cH}, arbitrary values of low and upper pressures in the driver P_{dL} and P_{dH} can be used.

For the single-acting compressors, shown in Figure 1a, the requirements towards the low and upper pressures in the driver are different. In this case, the relation between the pressures conforming the condition of the uniform piston movement is

$$(P_c - P_a)A_c = (P_d - P_a)A_d \tag{2}$$

where P_a is the ambient pressure.

Eq. 2 places a restriction on the driver pressure range for given compressor pressures. This can be understood considering a high-pressure process when ambient pressure can be neglected. In this case the forward and back piston strokes will be performed if $P_{dH}A_d \geq P_{cH}A_c$ and $P_{dL}A_d \leq P_{cL}A_c$ accordingly. From these requirements it follows that the driver pressure ratio should be not less than the compressor pressure ratio: $\frac{P_{dH}}{P_{dL}} \geq \frac{P_{cH}}{P_{cL}}$. However, for many practically interesting applications the pressure ratio in the compressor $r_c = \frac{P_{cH}}{P_{cL}}$ is much higher than the available pressure ratio in the driver $r_d = \frac{P_{dH}}{P_{dL}}$, especially within the scope of the current research focused on the low temperature difference applications of the IE engines.

The restriction on the pressure of the driving fluid in the single-acting scheme can be avoided using a modified, more flexible, scheme shown in Figure 2a.

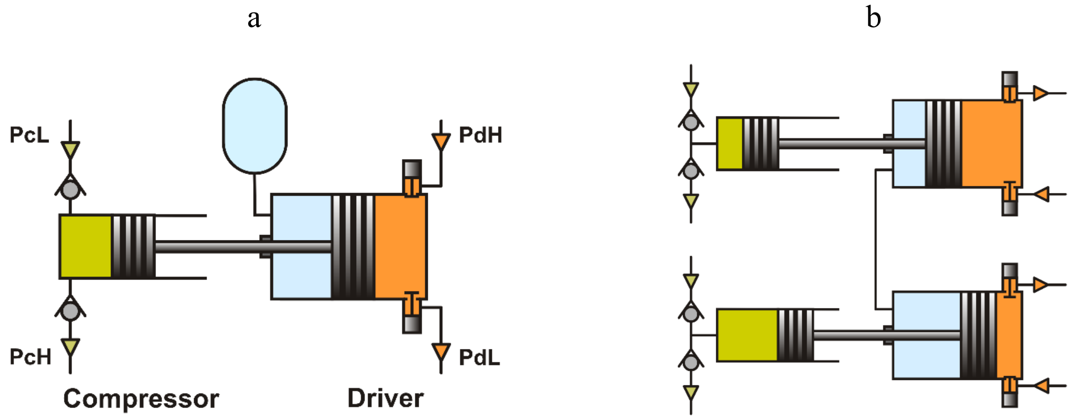

Figure 2. Modified single-acting schemes: with a receiver (a) and duplex scheme (b).

In this improved scheme there is a large volume accumulator or receiver with a specified pressure P_r which becomes an additional process parameter. Taking it equal to

$$P_r = \frac{P_{dL}(P_{cH} - P_a) - P_{dH}(P_{cL} - P_a)}{P_{cH} - P_{cL}} \tag{3}$$

allows for using any low and upper pressures in the driver, P_{dL} and P_{dH}, for given values of low and upper pressures in the compressor, P_{cL} and P_{cH}. It can be shown that in this case Eq. 1 also becomes valid.

The scheme with the receiver, Figure 2a, is a useful theoretical configuration. Better functionality can be achieved by using the duplex design, Figure 2b. Duplex is a combination of two identical machines operating in counterphase, both auxiliary chambers of which (in blue) are connected by a pipe. If the auxiliary chambers are filled with a gas at the pressure P_r needed in the receiver, they function as a large receiver without compression/expansion of the gas inside them. The auxiliary chambers can also be used for the lubrication, draining of working fluid leakages through the seals etc. An auxiliary chamber(s) can also be arranged in the compressor cylinder.

Having in mind the modified scheme, Figure 2, in the following analysis of the process efficiencies there is no need to distinguish single-acting and double acting schemes. For any given pressures in the compressor arbitrary low and upper pressures in the driver can be applied.

Efficiency of driving vapor use in basic vapor driven compressors

In this paper, the efficiency of vapor generation in the heat engine is not considered; it is assumed that driving vapor with a certain temperature and pressure is available. The focus of the study is on the efficiency of vapor use in the driver when it is applied for compression processes with different compression ratios either adiabatically or isothermally.

For the purpose of simple analysis, the following usual assumptions are made consistent with thermodynamic analysis:

- The process in the compressor is either adiabatic or isothermal.
- The process in the driver is adiabatic.
- The driver performs only useful work on the compression.
- Minimum volumes of the compression and driving cylinder are zero (no dead volume).
- Temperature and pressure of the fluids in the driver and compressor are uniform.
- Mechanical friction between moving and stationary parts in contact is negligible.
- Inertia of the pistons, piston rods and the fluids is negligible; this is justified for IE engines operating at low frequencies.
- Cross-sectional area of the piston rods is much smaller than the area of the pistons.
- Compressed and driving fluid are ideal gases; the ratios of the heat capacities at constant pressure and constant volume of the driving and compressed gases, γ_c and γ_d, were taken equal to 1.4 unless otherwise indicated.

To characterize the process we introduce a vapor use efficiency defined as the useful work performed by the driver per cycle per unit mass of the consumed vapor m_{d0}:

$$w = \frac{W_{c0}}{m_{d0}} \tag{4}$$

In Eq. 4 the useful work is designated as the work of the compressor per cycle, $W_{c0} = \oint P_c dV_c$, which is equal to the work of the driver, $W_{d0} = \oint P_d dV_d$, under the assumptions made.

The mass of the consumed vapor depends on its temperature at the end of compression stroke, T_{de}, and can be calculated as

$$m_{d0} = \rho_d(T_{de}, P_{dH})V_{d0} \tag{5}$$

where $\rho_d(T_{de}, P_{dH})$ and V_{d0} are the vapor density and its volume at the end of the compression stroke.

The specific work w can easily be calculated if the load on the driver piston is constant during the forward and back strokes of the piston. Such a constant load is realized in the limiting case when the compressor operates as an ideal pump. In this case the driver performs the maximum possible work, $W_{dmax} = (P_{dH} - P_{dL})V_{d0}$, because the pressure of the driving vapor is maximal during the compression piston stroke and minimal during the discharge piston stroke. From the energy balance equation it follows that temperature of the driving vapor at the end of the compression/pumping stroke is T_{dH}. Therefore, $m_{p0} = \rho_d(T_{dH}, P_{dH})V_{d0}$ and the specific work of the driver operating as an actuator of the ideal pump is

$$w_p = \frac{W_{dmax}}{m_{p0}} = \frac{P_{dH} - P_{dL}}{\rho_d(T_{dH}, P_{dH})} \tag{6}$$

w_p of Eq. 6 is taken as the benchmark for the comparison of the efficiencies of the driver in this paper. Thus the relative efficiency of vapor use is defined as

$$\alpha = \frac{w}{w_p} = \frac{\rho_d(T_{dH}, P_{dH})}{\rho_d(T_{de}, P_{dH})} \frac{W_{c0}}{(P_{dH} - P_{dL})V_{d0}} \tag{7}$$

α or T_{de} can be obtained from the energy balance for the compression stroke which in case of adiabatic process is:

$$\big(H_d(T_{dH}, P_{dH}) - U_d(T_{de}, P_{dH})\big)m_{d0} = W_{dc} \tag{8}$$

where H_d and U_d are specific enthalpy and internal energy of the driving vapor, and W_{dc} is the work of the driver in the compression stroke. Eq. 8 takes into account that the driving vapor enters the driver cylinder at temperature T_{dH} and pressure P_{dH}.

W_{dc} can be exressed through the work per cycle, $W_{d0} = W_{c0}$, and the discharge work of the driving vapor, $-P_{dL}V_{d0}$:

$$W_{dc} = W_{c0} + P_{dL}V_{d0} \tag{9}$$

From Eqs. 1, 5, 8, 9 and the fluid state equation ($\phi(\rho, P, T) = 0$) T_{de} and all other process characteristics can be obtained, if the work W_{c0} is known. Important qualitative conclusions can be drawn from these simple equations: for the highest specific work, Eq. 4, the internal energy at the end of the compression stroke $U_d(T_{de}, P_{dH})$ should be as low as possible, and the useful work W_{c0} as large as possible. Using Eq. 4 and taking into account that the highest rate of the pressure change with the volume occurs at adiabatic expansion, one can prove that the Rankine cycle has the maximum steam use efficiency.

The equations above can readily be treated if both the compresed and driving fluid are the ideal gasses. In this case enthalpy, internal energy, density and specific heats, c_p and c_v, are

$$H_d(T, P) = c_pT, \qquad U_d(T, P) = c_vT, \qquad \rho_d(T, P) = \frac{P\mu}{RT} \tag{10}$$

$$c_p = \frac{R\gamma_d}{\mu(\gamma_d - 1)}, \qquad c_v = \frac{R}{\mu(\gamma_d - 1)}, \quad \gamma_d = \frac{c_p}{c_v} \tag{11}$$

in which R is the gas constant and μ is the molecular mass of the driving vapor.

Combining these equations with Eqs. 4, 5, 8 and 9 the temperature of the gas at the end of the compression stroke and the efficiency of gas use can be obtained:

$$\frac{T_{de}}{T_H} = \frac{\gamma_d r_d}{\gamma_d + (r_d - 1)\left[\frac{\omega_{c0}}{r_c - 1}(\gamma_d - 1) + 1\right]}, \qquad \alpha = \frac{\omega_{c0}}{r_c - 1}\frac{T_{de}}{T_{dH}} \tag{12}$$

where $r_c = \frac{P_{cH}}{P_{cL}}$ and $r_d = \frac{P_{dH}}{P_{dL}}$ are the pressure ratios of the compressor and driver, and $\omega_{c0} = \frac{W_{c0}}{P_{cL}V_{c0}}$ is the dimensionless compressor work. In case of adiabatic and isothermal compressions the dimensionless compressor works are:

$$\omega_{c0,ad} = \frac{\gamma_c}{\gamma_c - 1}\left(r_c^{\frac{\gamma_c - 1}{\gamma_c}} - 1\right), \qquad \omega_{c0,isothermal} = ln(r_c) \tag{13}$$

where γ_c is the ratio of the heat capacities at constant pressure and constant volume of the compressed gas.

Figure 3 shows the calculated efficiency of driving gas use.

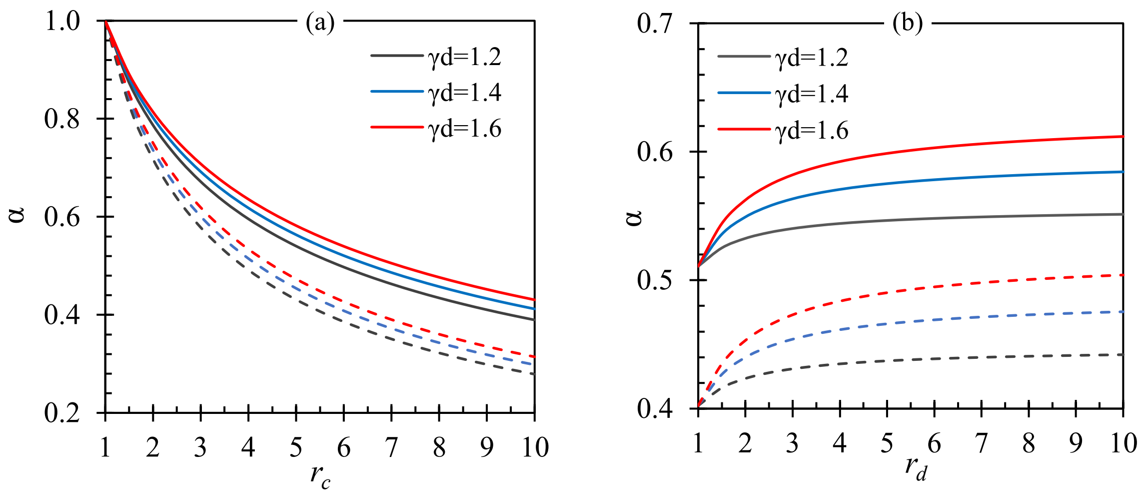

Figure 3. Efficiency of the driving gas use as a function of the pressure ratio in the compressor at r_d = 3 (a) and driver at r_c = 5 (b) and different γ_d; solid lines – adiabatic compression, dashed lines – isothermal compression, γ_c =1.4.

The results obtained show that the efficiency of the driving gas use depends strongly on the properties of the gases, γ_c and γ_d, and on the pressure ratios in the compressor and driver. For practically interesting compression processes (r_c is not close to 1) the efficiency of vapor use is much less than the efficiencies of vapor use in vapor-driven pumps. The pressure ratio in the compressor r_c has the major effect on the efficiency. It drops rapidly with r_c and increases with the pressure ratio in the driver r_d. The reason for the low efficiency is that a significant fraction of the energy of the driving gas supplied to the driver is spent on the compression of the driving gas that is already present in the driver.

It should be noted that although the dependence of the efficiency on the driver pressure ratio r_d is not strong, operation with r_d approaching 1 is impractical, since in this case the work done approaches zero.

The relative position of the red, blue and grey lines in Figure 3 is explained by the fact that the temperature elevation of the driving fluid with larger γ_d during the compression stroke is higher and the density is lower. Accordingly, the consumption of the driving gas with higher γ_d is lower whereas the efficiency of its use is higher.

Isothermal compression is preferred over adiabatic compression in conventional compression processes because the compression work is lower. For vapor driven compression, this advantage is diminished, see Figure 3, because the driver piston is less loaded when compression is isothermal.

Based on the results obtained, several modified methods of vapor-driven compression can be proposed. These methods are presented below.

Multistage compression

When multiple compressor cylinders are connected in series and a gas or vapor is compressed in stages the arrangement is referred to as a multistage or stage compressor. If compression ratio is high, the compression is carried out in more than one step [6].

In almost all multi-stage applications the compressed gas or vapor will be cooled between stages because with intercooling, the compression more closely approximates an isothermal compression with resulting lower power requirement [6].

The simplest scheme of two-stage vapor driven compression with intercooling is presented in Figure 4. In this scheme each driver piston is actuated by the same vapor source. A difference compared to the conventional scheme is that the compressor pistons are actuated by a driving vapor and not by an electric motor or IC engine through a crank mechanism. In addition, it is important to note that the main reason for multistage vapor-driven compression is to improve efficiency of vapor use rather than a considerable temperature rise and a reduction of the compression work. Therefore processes without intercooling could also be of interest.

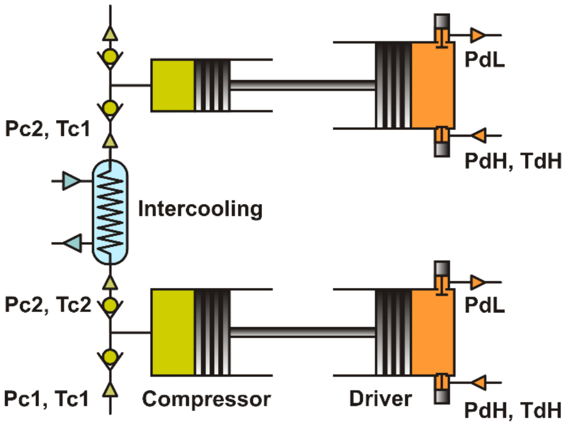

Figure 4. Scheme of two-stage vapor-driven compressor.

The multistage vapor driven compression process can be studied using the same approach as in the previous section. Figure 5 shows examples of efficiency of the driving gas use for one-, two- and three-stage compression. The results presented in Figure 5 were obtained for adiabatic compression under assumption that the compressor pressure ratio is the same in all stages.

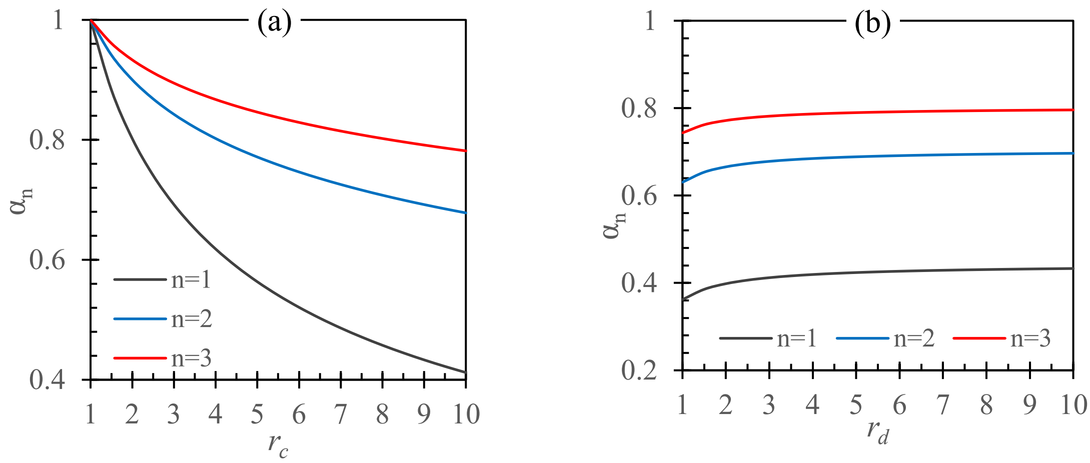

Figure 5. Efficiency of the driving gas use for one, two and three stage compression as a function of the compressor pressure ratio (a), r_d = 3 and driver pressure ratio (b), r_c = 10.

Significant improvement of the efficiency is observed when the number of stages increases, especially if two stage compression is used instead of single stage compression. The efficiency increases with the number of stages and in the limit of large number of stages its value approaches 1. In this case pressure changes in the compressor and driver at each stage are negligible and the process is the same as in an ideal pump.

The efficiency is the same for processes with and without intercooling because in case of the ideal gases it is determined by the pressure ratios in the compressor and driver and the heat capacities. However, intercooling reduces the driving gas consumption as the compression work is reduced. This effect is significant at high compressor pressure ratio.

It can be shown that, in addition to improving the efficiency, stage compression permits substantially reduce the total volume of the drivers, especially for processes with intercooling. For practically interesting pressure ratios the total volume of the drivers can be reduced several times. It is interesting that the total volume of the compressors and drivers can decrease with the number of stages at certain process parameters.

Compression with driving vapor reuse

Vapor in the driver at the end of the compression stroke is a valuable energy source. Its pressure is equal to the initial pressure P_{dH} and its temperature is higher than the initial temperature T_{dH}. Therefore, one of the ways to improve efficiency of the driving vapor use is to reuse the vapor that is in the driver at the end of compression stroke. A difference compared to the basic method, Figures 1 and 2, is that, the driving vapor at the end of the compression stroke is not discharged from the driver to the vapor circuit (specifically to the recuperator, Figure 1a) but used for initial, or intermediate, compression in another compressors. The vapor from the vapor circuit of the heat engine is used only in the final compression phase.

Many different schemes utilizing the driving vapor reuse principle can be proposed. As an example, a simple scheme of a single stage compression processes with vapor reuse is shown in Figure 6. It includes two drivers, two compressors and two 3-way valves. A description of the process can be started at the initial moment when driver 1 is at the end of the compression stroke and is filled with high pressure, hot driving vapor (P_{dH}, T_{dH}), Figure 6a. Driver 2 at this moment is in the end of its discharge stroke and does not contain driving vapor. Valve 1 is closed whereas valve 2 opens and communicate the drivers. As a result, driver 2 performs compression in compressor 2 until the pressure forces on the driver 2 and compressor 2 pistons equalize, Figure 6b. Thus, at this phase of the process, compression in compressor 2 occurs with vapor from driver 1. Then valve 2 closes and the driving vapor from its original source (heat engine) is delivered to driver 2 through valve 1 to accomplish the compression stroke in compressor 2, Figure 6c. Then, or simultaneously, driving vapor remaining in driver 1 is discharged and compressor 1 is filled with the fluid to be compressed, Figure 6d. After that process repeats with the interchanged functions of compressor 1 - driver 1 and compressor 2 - driver 2.

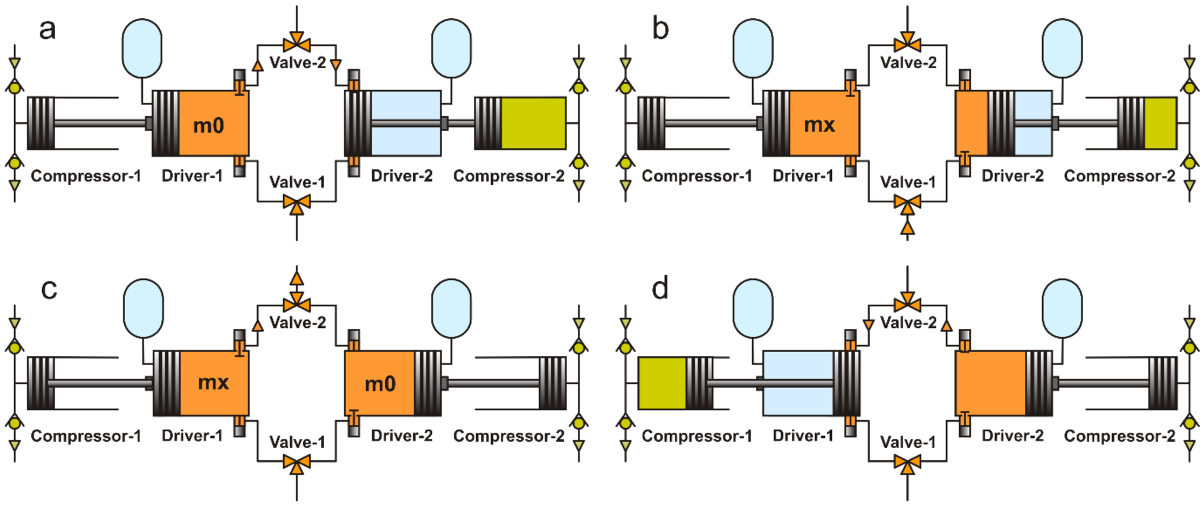

Figure 6. Scheme of single stage vapor driven compression with vapor reuse; different phases of the process are shown.

Taking into account the advantages of multistage vapor driven compression a logical step is to apply the idea of vapor reuse for multistage compression processes. Figure 7 shows an example of two-stage scheme with vapor reuse. At first stage the fluid is compressed from the low pressure P_{cL} to some intermediate pressure in compressor 1. Then partly compressed fluid is displaced to compressor 2 where it is compressed till the desired pressure P_{cH}. The drivers operate in the same way as in the scheme shown in Figure 6. The fluid from compressor 1 can be displaced to compressor 2 directly or through an intercooler as in Figure 7.

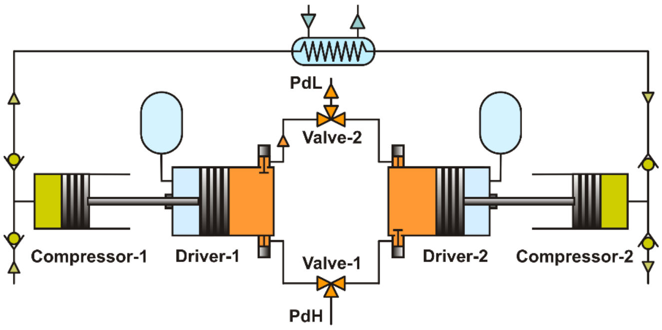

Figure 7. Scheme of two-stage vapor driven compression with vapor reuse.

Figure 8 shows the calculated efficiencies for different processes: without driving gas reuse (basic scheme), and with driving gas reuse in the one- and two-stage compression as a function of the pressure ratios in the compressor and driver.

The results presented were obtained using the same approach as in the previous sections, although with slightly more complex mathematics. A significant increase in efficiency is seen due to the reuse of driving gas. Two-stage compression with reuse of the driving gas offers the efficiency very near to the efficiencies of vapor driven pumping process. Remarkably that at low compressor pressure ratio the efficiency becomes even higher than in case of operation as the ideal pump.

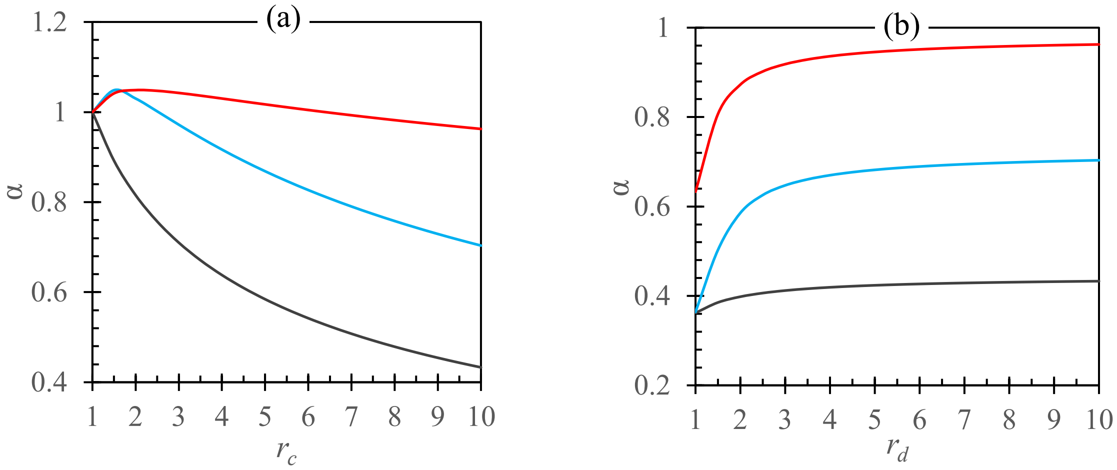

Figure 8. Efficiency of the driving gas use as a function of the compressor (a) and driver (b) pressure ratio: grey lines - basic compression scheme, blue lines - single-stage with gas reuse, red lines - two-stage with gas reuse; r_d = 10 (a), r_c = 10 (b).

Vapor driven compression with force transmission

From the analysis of the basic vapor driven compression processes it follows that the rising pressure of the driving vapor during compression stroke is the reason of low vapor use efficiency. One of the possible methods to avoid significant pressure rise in the driver and improve its efficiency is to connect the piston rods through a mechanical linkage (kinematic transmission) to manage forces and movement of the pistons.

A massive flywheel in combination with a crank gear is well-known linkage mechanism used in different machines e.g. in reciprocating piston compressors driven by an electric motor or IC engines. This method can also be applied in vapor driven compressors as e.g. is shown in Figure 9a. In this case, some part of power delivered by driver in an initial period of the compression cycle is accumulated by a flywheel (shown in blue in Figure 9a) and then is used in a final period of the compression piston stroke. If the intake valve of the driver is open and does not create a resistance to the driving vapor flow the pressure in the driver will be constant and equal to its maximum pressure P_{dH}. As a result, the vapor use efficiency will be the same as in case of an ideal pump.

Many mechanisms for force transmission can be proposed, which help to increase the efficiency of vapor use without using the effect of energy storage. One of such methods was proposed in [31] to equalize the variable forces between the steam actuated power piston and the saline water pump piston in a reverse osmosis system.

Mathematically the presence of a mechanical linkage can be expressed by an equation relating the forces $F_c(x)$ and $F_d(y)$ on the compressor and driver piston:

$$F_d(y) = Tr(x)F_c(x) \tag{14}$$

In these equation x and y are positions of the pistons in the compressor and driver and the transmission function $Tr(x)$ is expressed as a function of x. Obviously, choosing $Tr(x)$, one can achieve any desired balance of the forces.

We will briefly illustrate the concept of force transmission using as an example a simple lever mechanism shown in Figure 9b. The transmitting function in this case expressed through the angle between the driver piston rod and the lever is $ctg(\varphi)$.

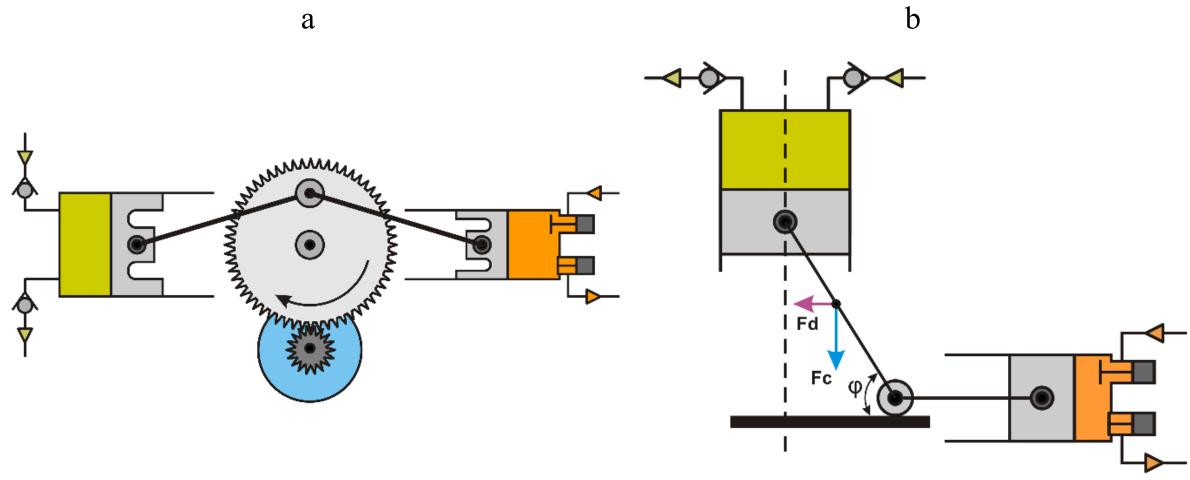

Figure 9. Schemes of vapor driven compressor with mechanical linkages between the pistons.

Figure 10 shows an example of the calculated vapor use efficiency in the case of double acting piston-driver arrangements when the angle φ changes in the range 20° – 70°.
Gain in the efficiency due to the transmission is clearly seen. An additional advantage of the transmission is that the improved efficiency is achieved with smaller driver volume as the pressure span in the driver becomes narrower, close to the maximum pressure.
Generally the analysis of the systems with a transmission is more complicated than without it. Not every transmission can provide a balance of the forces and uniform piston movement. In the example above even a minor force on the driver piston will result in acceleration of the both

pistons if the angle φ approaches 90º and $ctg(\varphi)$ tends to zero. Also the ambient pressure or pressure in the receiver become an additional process parameters in case of single acting scheme. These important details are beyond the scope of this paper.

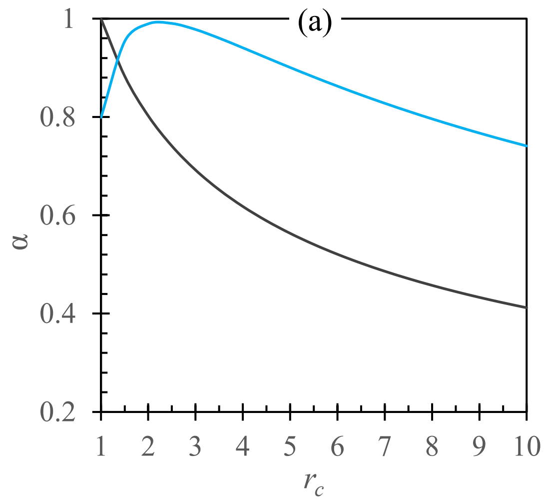

Figure 10. Efficiency of the driving gas use without (grey lines) and with (blue lines) transmission as a function of the pressure ratios in the compressor, r_d = 10 (a) and driver r_c = 10 (b).

Discussion and conclusions

In the known steam driven compressors [18] the steam was supplied to the compressor from a boiler and exhausted to atmosphere. Therefore thermal efficiency of the compressor was determined by the steam use efficiency introduced in this paper, Eq. 4 or 7, because the supplied heat was directly proportional to the mass of the generated steam.

The results obtained in this paper for the simplest schemes might explain why such steam driven compressors are not used. The use of closed steam cycles, shown in Figure 1a, will not help to improve their thermal efficiency because heat regeneration in case of steam is very poor as explained e.g. in [32]. However, if a regenerative heat engine is used and the steam is replaced by a fluid with thermodynamic properties allowing efficient heat regeneration, see [26], it can be expected that processes with low vapor use efficiency might be feasible.

Inefficient use of the driving vapor manifests in the higher temperature of vapor exhausted from the driver. In the case of ideal gases this temperature is determined by Eq. 12. The increased temperature offers additional potential for improved heat recovery in the recuperator of the IE engine. This problem requires further detailed investigation accounting for thermodynamic properties of the driving fluid.

This paper presents how, if necessary, the disadvantages of the schemes with low vapor use efficiency can be eliminated with the proposed above methods – multistage vapor driven compression, vapor reuse, force transmission or combination of thereof – and other methods. With these methods the vapor use efficiency can become comparable with the efficiency in pumping processes. Which of them is preferable depends on technical complexity and eventually on the economics.

Overall, the results presented in this paper show the significant potential of the heat driven compression concept and the need for further research. This concepts is especially attractive because it allows the of low-grade heat sources (<100 ºC) which cannot be effectively used with existing technologies.

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