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Modelling of a new thermal compressor for supercritical CO_2 heat pump

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Abstract:

A new concept of thermal compressor has been designed by the boostHEAT company. This compressor uses thermal energy provided through the heater instead of mechanical energy to increase the pressure of the heat pump working fluid. The compressor is made up by the following parts: a cylinder with a displacer piston, a heater, a regenerator and a cooler. The heater is connected to the hot part of the cylinder on the one hand and to the regenerator on the other hand. The cooler is connected to the regenerator on the one hand and to the cold part of the cylinder on the other hand. The cold part of the cylinder is connected to the low pressure branch of the heat pump (evaporator) through an automatic inlet valve, and to the high pressure branch of the heat pump (gas cooler) through an automatic exhaust valve. The compressor is intended to replace the conventional mechanical compressor in a CO_2 heat pump for the residential heating or combined heat and power market. The main feature of the system is that the working fluid of the thermal engine for compression is the same as the working fluid of the heat pump.

The principle of the new thermal compressor and its advantages for heat pump application will be briefly presented. A model of the thermal compressor has been developed. Modelling results related to the regenerator, the piston rod diameter, the size of the adiabatic dead volumes and the working fluid leaks in the annular gap between the cylinder liner and the piston are presented.

Keywords:

CO_2, Heat pump, Thermal compressor.

1. Introduction

The building represents a major energy consumption. For example in France, this sector consumes 44% of the total energy and accounts for 25% of the greenhouse gas emissions. Heating alone accounts for 70% of the total energy consumption of the buildings [1]. Heating is therefore an important field of energy savings. Efforts focusing on improving the energy efficiency of heat production devices will have a decisive impact on the level of fuel consumption and on the reduction of greenhouse gases emissions.

Low temperature heat production devices from a heat source such as oil or gas combustion, combined with a heat pump effect (free energy taken from the environment) provide a significant improvement of the energy efficiency of the fuels. These technologies are very important issues for our societies. The underlying thermodynamic principles of these systems, generically known as trithermal systems, demonstrate an increased energy efficiency for applications in heat production compared to the conventional boiler. However the commercial use of trithermal systems remains very limited. Amongst the trithermal systems it is worth mentioning non-integrated devices such as the coupling of a gas engine with a conventional vapour compression heat pump, the coupling of a Stirling engine with a conventional vapour compression heat pump or with a Stirling heat pump, or integrated devices such as the absorption or the adsorption heat pumps or the Vuilleumier machines.

On the other hand environmental concerns have conducted to the banishment of some refrigerants and CO_2 tends to become more and more used as the best alternative [2-3]. The use of a transcritical cycle has emphasized the development of refrigerating machines or heat pumps using CO_2 as the working fluid [4].

The concept developed by the boostHEAT company and presented here is quite new and original. It consists in an integrated trithermal system combining a thermal compressor and a conventional vapour compression heat pump. The thermal compressor is identical to a gamma type Stirling engine of which the power piston and cylinder have been replaced by automatic inlet and exhaust valves.

2. The new thermal compressor

2.1. Principle of the thermal compressor

From the mechanical point of view, the thermal compressor is made up by the following components as it is illustrated in Fig. 1: a casing or cylinder, a displacer piston (D), a heater (H), a regenerator heat exchanger (R), a cooler (K), an inlet valve (IV) and an exhaust valve (EV).

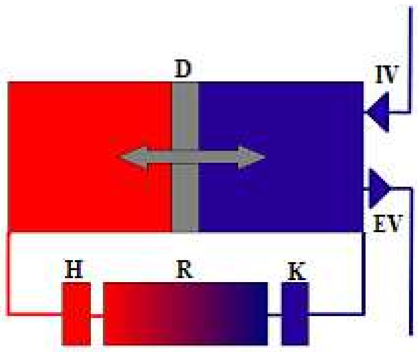


Fig. 1. The new concept of thermal compressor for supercritical CO_2 heat pump application.

The main advantage of the system is that despite its great structural simplicity, it ensures all functions equivalent to a system composed of a thermal engine with external heat supply, a power transmission and a mechanical compressor. The most important feature of the boostHEAT compressor is that the working fluid of the power cycle allowing the compression is the compressed fluid itself. The simplicity and performance of the heat driven compressor result from this essential feature. In particular, this feature allows the absence of any mechanical compression and power transmission device. The heat driven compressor performs a mechanical work of compression on a fluid (volumetric compression: suction-compression-discharge) without mechanical compression piston.

The displacer role is similar to that of the displacer (as opposed to power piston) of a gamma type Stirling engine, which is to ensure the movement of some fluid between the hot and cold parts of the cylinder through the exchangers (H-R-K). At any time the communication between the hot and cold parts of the system is open. The pressure remains uniform in the whole device (except for pressure losses), and the piston does not transmit mechanical power to compress the fluid.

The absence of mechanical compression device and the mechanical simplicity of the compressor will also provide an excellent mechanical efficiency compared to a system made up of engine / power transmission / compressor, particularly at the level of the piston rings / cylinder friction. Indeed the pressure differential is almost zero in the case of a displacer.

Such a heat driven compressor is thought to be very convenient to replace the mechanical compressor of a conventional transcritical CO_2 heat pump. So the fluid in the compressor will be CO_2. Modelling results show that for CO_2 heat pump applications a two stage thermal compressor is needed.

The thermodynamic cycle described by one stage of the thermal compressor is made up of four processes:

- An isochoric compression due to heat transfer. The valves are closed. The displacer is initially on the left dead centre and moves towards the right. Therefore the working fluid at initial pressure P_1 is transferred from the cold part to the hot part of the cylinder through the three heat exchangers K, R and H. The total volume of the system is constant. The fluid pressure gradually increases up to pressure P_2.
- An isobaric exhaust. At pressure P_2 the outlet valve opens and a mass of fluid is pushed out of the cylinder as long as the displacer moves towards its right hand side dead centre.
- An isochoric expansion by cooling. Both valves are closed. The displacer moves towards the left so that the working fluid flows from the hot side of the cylinder to the cold side of the cylinder. The total volume of the system is constant. The working fluid pressure gradually decreases from pressure P_2 up to pressure P_1.
- An isobaric sucking. At pressure P_1 the inlet valve opens. A mass of fluid is sucked into the cylinder as long as the displacer moves towards its left hand side dead centre.

2.2. The model

The working fluid is carbon dioxide (CO_2) which is modelled as a real gas with non-constant thermo-physical properties. In order to describe the behaviour of the thermal compressor, the coupled analysis usually applied for Stirling engine modelling is used [5-6]. This analysis is more accurate than the others but it requires a higher computing time.

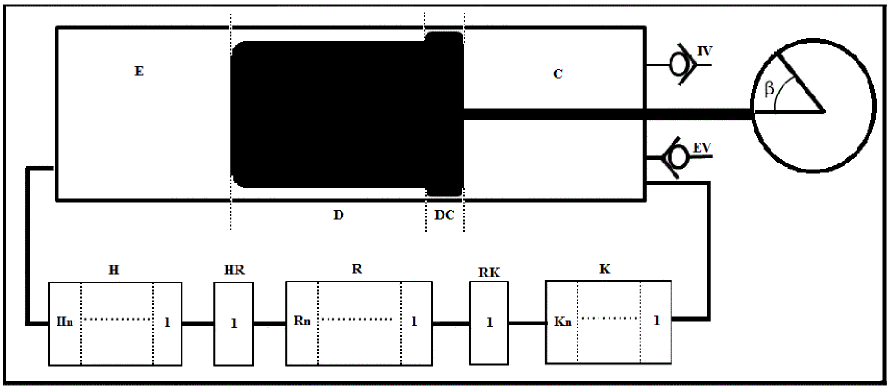


Fig. 2. Control volumes of the compressor.

The compressor stage is divided into several control volumes as shown in Fig. 2. Each cold and hot part of the cylinder acts as one control volume. The heat exchangers are divided in a lot of sub-volumes. The dead volume between the regenerator and the cooler is also divided in several control volumes. The number of control volumes depends on the temperature gradients in the heat exchanger considered and on the size of the heat exchanger. The number of control volumes is a compromise between accuracy and computing time [7].

The fluid physical properties at the outlet of each control volume are the ones at the inlet of the adjacent control volume. Some usual assumptions are used:

- There is no leak of the working fluid;
- The hot and the cold parts of the cylinder, and the hot and cold dead volumes are adiabatic;
- The instantaneous pressure in every control volume is the average of the pressure of the control volume interfaces;
- The temperatures of the wall of the heater and the cooler are constant;
- Fluid kinetic energy is negligible;
- Fluid flow is mono-dimensional.

According to the assumptions the coupled analysis is based on the equations for mass and energy balance on each control volume. The conservation of mass is given by:

$$\dot{m}_r + \dot{m}_l + m' = 0 \qquad (1)$$

$$m = V / v \qquad (2)$$

$$v = v(fluid; T; P) \qquad (3)$$

The total mass of fluid contained in the compressor is the sum of the masses contained in the compressor components. The mass of each component H, R and K is also the sum of the masses of the control volumes.

$$m_{tot} = m_C + m_K + m_{RK} + m_R + m_{HR} + m_H + m_E + m_D + m_{DC} \qquad (4)$$

The energy balance applied to the i[th] control volume is written as:

$$(mu)'_i + (\dot{m}_r h_r + \dot{m}_l h_l)_i = W_i + Q_i \qquad (5)$$

The volume variation of the hot and the cold part of the cylinder can be linked to the instantaneous crank angle β. In the control volumes of the heat exchangers the mechanical power is equal to zero. The characteristics of the real working fluid are taken from numerical tables and each parameter is function of two others. The internal energy and enthalpy are function of pressure and temperature like the specific volume in (3).

The fluid flow rates are assumed to be positive if the fluid moves from the cooler to the heater. At the interfaces of a control volume the fluid rate is assumed to be positive if it comes out and negative if it comes in. The flow rate in a control volume is given as a function of the two flow rates at the right and left interfaces:

$$\dot{m}_i = (\dot{m}_l - \dot{m}_r)_i / 2 \qquad (6)$$

The relation between pressures at control volume interfaces is expressed with the pressure drop (7). The pressure drop is defined as a function of friction factor, flow and geometric characteristics.

$$P_{l,i} = P_{r,i} - \Delta P_i \times TEST_i \qquad (7)$$

$$where\ (TEST_i = -1) \text{ if } (\dot{m}_i < 0) \text{ and } (TEST_i = 1) \text{ if } (\dot{m}_i > 0)$$

The heat power exchanged between the fluid and the wall of the heat exchangers H, R, K is expressed with the heat transfer coefficient, the heat transfer area and the temperature difference (8). The heat transfer coefficient is determined by specific correlations which differ in each exchanger.

$$Q_i = H_{T,i} A_i (T_{w,i} - T_i) \qquad (8)$$

In the case of the regenerator, the temperature of the matrix depends on heat transfer with fluid and on axial conduction through the matrix material [10]. The instantaneous variation of the temperature of the matrix of the i[th] control volume is expressed as:

$$\rho_{m,i} V_{m,i} Cp_{m,i} T'_{m,i} = -Q_i + (Q_{cond,l} + Q_{cond,r})_i \qquad (9)$$

$$Q_{cond,s,i} = (k_{m,i} A_{m,i}) / (L_{m,i} / 2)\,(T_{m,s} - T_m)_i \quad s = l \text{ or } r \qquad (10)$$

When the inlet or outlet valve is open, the total mass of fluid in the thermal compressor is modified (11) and (1) and (5) applied to the cold part of the cylinder become respectively (12) and (13):

$$\Delta m_{tot}^{t,t+\Delta t} = m_{in,out} \qquad (11)$$

$$\dot{m}_{r,K1} + \dot{m}_{in,out} + m'_C = 0 \qquad (12)$$

$$(mu)'_C + (\dot{m}_r h_r + \dot{m}_l h_l)_C + (\dot{m}_{in} h_{in} + \dot{m}_{out} h_{out}) = W_C \qquad (13)$$

2.3. Characteristics of the modelled thermal compressor

The bore of the cylinder is equal to 70.4 mm and the stroke is 44 mm. The operating frequency is 3.5 Hz. The dead volume in both parts of the cylinder is assumed to be 2% of the swept volume. Unless otherwise noted, the rod diameter is 16 mm. The heater (H) is an annular heat exchanger (V_H = 12.5 cm^3), while the cooler is a multi-tubular heat exchanger (V_K = 8.1 cm^3). The matrix of the regenerator is a stainless steel plain wire mesh with a wire diameter equal to 0.033 mm. The void volume of the regenerator is 125.5 cm^3, while the volume occupied by the fluid is V_R = 74.26 cm^3.

The wall temperature of the heater is T_H = 550 °C, while the wall temperature of the cooler tubes is T_K = 30 °C. The temperature of the working fluid at the inlet is equal to 1°C. A low pressure of 35 bar and a high pressure of 55 bar are considered respectively at upstream and downstream of the thermal compressor. A constant pressure loss of 0.5 bar is considered in the valves.

3. Results and discussions

Two cases are considered. In the first case, it is assumed that no fluid can flow through the annular gap between the displacer piston and the liner, which are represented by the control volumes D and DC (Fig. 2). In the actual prototype, there are no rings on the displacer and the sealing is achieved by mean of a contactless thin annular gap between the displacer and the liner. The length of the first part of the displacer D is 113.6 and the annular width of the gap is 0.15 mm. The length of the second part of the displacer DC is 29.2 mm and the annular width of the gap is 0.035 mm. So in the second case, fluid is allowed to leak in these very thin annular gaps. These control volumes are modelled as the others. The liner wall is assumed to be adiabatic. The heat transfer between the fluid in the annular gap and the displacer wall is taken into account assuming a constant linear temperature profile along each part of the displacer. Based on experimental measurements realized on the prototype, the temperature of the displacer is assumed to decrease linearly from 450 °C at the top of the displacer (side E, Fig. 2) up to 150 °C on the bottom of the first part of the displacer (D, Fig. 2) while it is assumed to vary from 72 °C up to 59 °C on the second part (DC, Fig. 2) of the displacer.

3.1. No leaks around the displacer

Fig. 3 shows the evolution of temperature in different parts of the compressor. In the cold and hot parts of the cylinder the temperature of the fluid is affected by the incoming gas flows and by the compression and expansion processes. The temperature gradient is more important in the regenerator and the temperature of the working fluid has a linear profile.

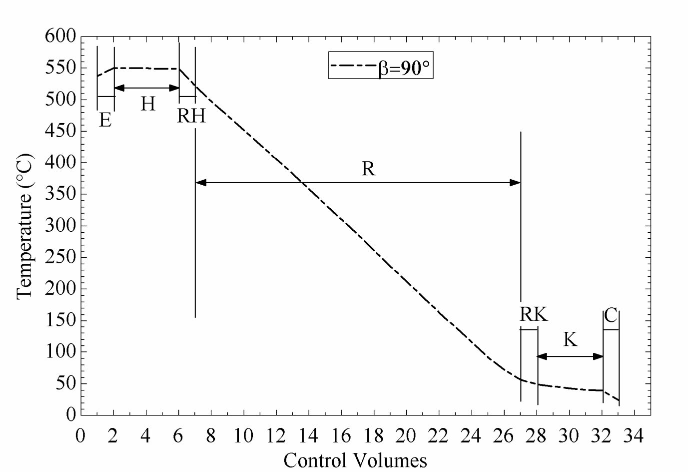


Fig. 3. Instantaneous temperature profile along the compressor.

The cyclic variation of the temperatures of matrix and working fluid in the cold and hot ends control volumes of the regenerator are presented respectively in Fig. 4. The curves of temperatures of the matrix and the fluid in the middle control volume are superposed to those of the hot and cold ends control volumes. It is clear that the temperature amplitudes of the fluid and matrix in the cold and hot ends control volumes are higher than the temperature amplitudes of the fluid and the matrix in the central control volumes. This higher temperature amplitude at the ends of the regenerator control volume is caused by the incoming gas flows whose temperatures differ significantly from the temperatures of the fluid and the matrix in the ends of the regenerator. Anderson et al [8-9] and Byun et al [10] found that the temperature oscillations in a regenerator matrix is significant in the ends of the regenerator and that less steep matrix temperature gradients occur in the central part of the regenerator.

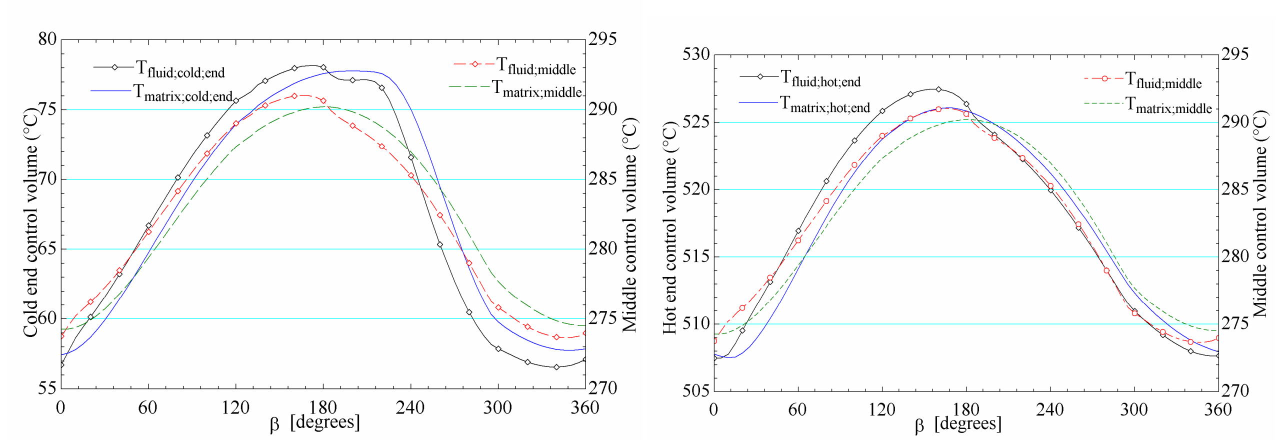
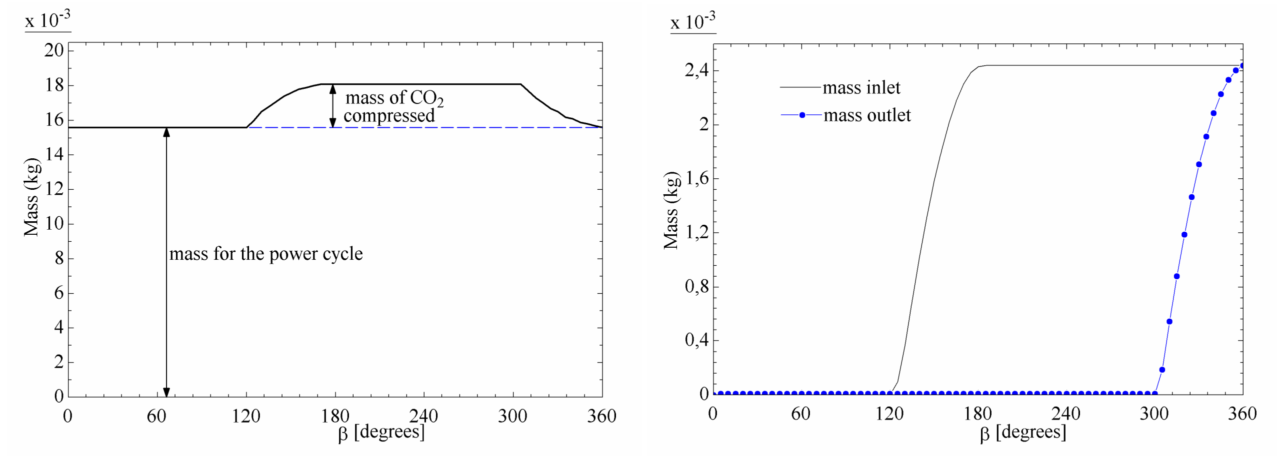


Fig. 4. Temperatures amplitudes of the fluid and the matrix in the central and ends control volumes of the regenerator.

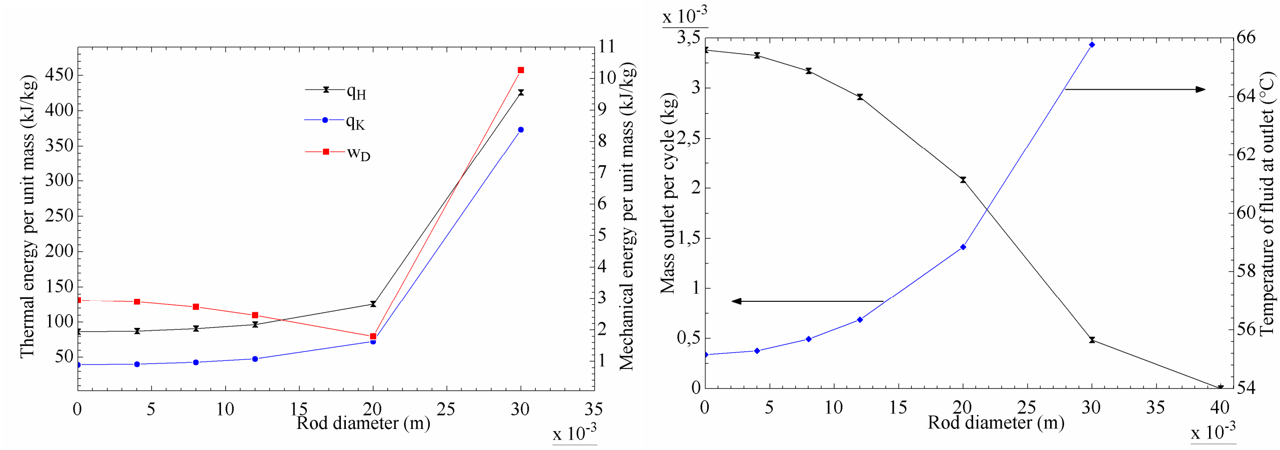
The presence of valves permits the outlet of compressed fluid mass at high pressure and the inlet of fluid mass at low pressure. Figure 5 presents the quantities of mass for the power cycle and the compressed fluid. The mass of the compressed fluid is about 15% of the mass of fluid needed for the power cycle.



(a) (b)

Fig. 5. Inlet and outlet mass of fluid per cycle and mass for the power cycle.

The instantaneous volume of the cold part of the cylinder is affected by the presence of the rod. Figure 6 (a) presents the influence of the diameter of the rod on the thermal energies exchanged in the heater and cooler and the mechanical energy of the displacer per unit mass delivered. The mass of compressed CO_2 delivered per cycle and its temperature are also plotted (Fig. 6 (b)). The energy supplied by the heater per unit of mass of fluid increases slowly with the increase of the rod diameter up to a value of 20 mm for the outlet pressure of 55 bar (Fig. 6 (a)). Above this value the heater energy supply increases significantly, due to the important decrease of the quantity of CO_2 delivered. For a rod diameter of 40 mm, no CO_2 is delivered (Fig. 6 (b)). The minimum mechanical energy per unit mass needed for an outlet pressure of 55 bar is obtained with a rod diameter of 20 mm (Fig. 6 (a)). Figure 6 (b) also shows that the temperature of the fluid flowing out of the compressor increases with the increase of the rod diameter.



(a) (b)

Fig. 6. Evolution of the energies exchanged per unit mass delivered (a), and the mass and temperature of the CO_2 delivered per cycle.

3.2. Impact of leaks around the displacer

The impact of fluid flow through the annular gap between the displacer piston and the liner is considered. Figure 7 shows the evolutions of temperature and mass flow rate at the left (side E) and right interface (side C) of the displacer. Temperatures levels at the left and right interfaces of the displacer annular gap are closed respectively to those in the hot and cold parts of the cylinder. The mass flow rate in the annular gap between the displacer and the liner is very low compared to those at the left side of the expansion space (E) and at the right side of the compression space (C).

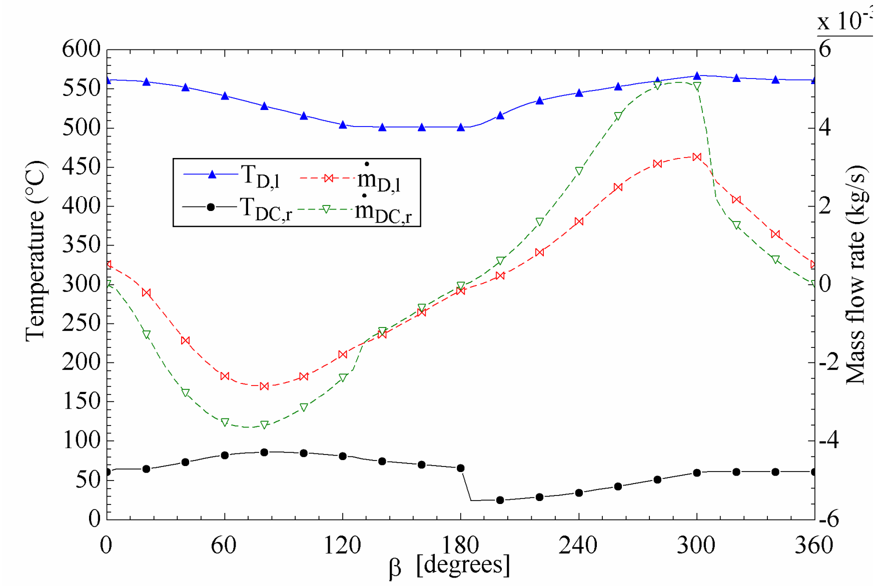


Fig. 7. Temperature and mass flow rate of the fluid at the left and right hand side of the displacer annular gap interface.

Figure 8 shows the superposition of temperature curves, with and without leaks, in the hot and cold parts of the cylinder. The fluid temperatures have similar evolutions with and without leaks, but the temperature level is always higher in the case of leaks in the annular gap. The temperature difference is more important on the cold hand side (C) of the cylinder.

According to the French regulation, the factor of conversion of mechanical energy into primary energy is set to 2.58. So the primary energy needed to run the compressor is computed by adding the mechanical power consumed (or in some cases delivered) by the displacer multiplied by 2.58 to the thermal power supplied by the heating device.

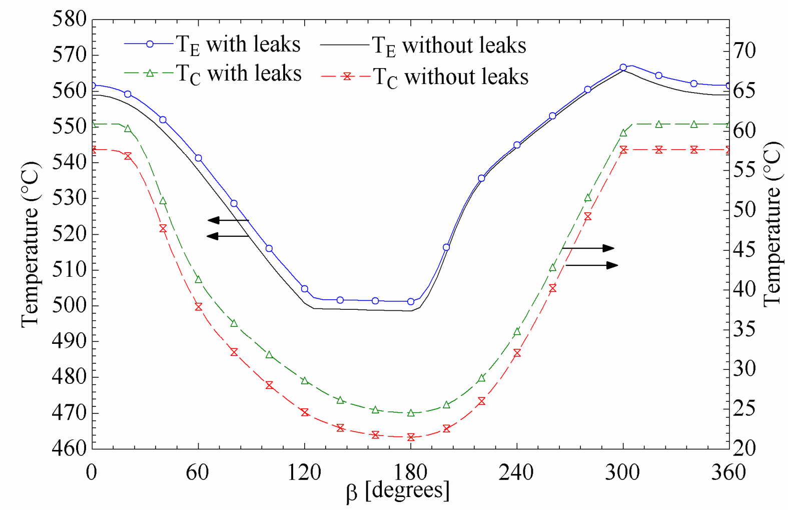


Fig. 8. Temperature of the fluid in the hot and cold parts of the cylinder, with and without leaks.

Table 1 presents a comparison of the cycle averaged energies exchanged by the fluid per unit mass of CO_2 delivered, with or without leaks. The mass of CO_2 delivered per cycle and the temperature of the CO_2 at the exhaust are also presented. In the case of leaks, the heat per unit mass supplied to the fluid in the heater q_H is a bit lower, but the heating device has also to supply heat to the displacer wall which, in turn, will heat the fluid flowing in the annular gap. The net heat supplied by the heating device, which is ($q_H + q_D + q_{DC}$) is thus much higher in the case of leaks. Moreover, the mechanical energy supplied by the displacer to the fluid is also higher in the case of leaks. The primary energy needed to compress one kilogram of CO_2 is thus much higher in the case of leaks. This supplementary energy is partially given to the environment in the cooler (q_K is higher in the case of leaks), but also as sensible heat to the compressed fluid (T_{out} is higher in the case of leaks). Finally, it can be seen that the quantity of fluid compressed per cycle is lower in the case of leaks. Therefore it can be said that even if the relative mass flow rate of CO_2 in the annular gap is very low, the consequence of this flow on the compressor performance is very important.

Table 1. Comparison of results with and without leaks

	q_H (kJ/kg)	q_K (kJ/kg)	w (kJ/kg)	q_D (kJ/kg)	q_{DC} (kJ/kg)	q_{prim} (kJ/kg)	m_{out} (kg)	T_{out} (°C)
Without leaks	110.61	–59.93	2.03	0.00	0.00	115.84	$2.44\ 10^{-3}$	57.7
With leaks	108.91	–74.543	2.296	28.86	–8.51	135.18	$2.19\ 10^{-3}$	60.9

3.3. Impact of the dead volumes

In addition to the heat exchangers which are considered as dead volumes, some other unswept dead volumes can exist when the thermal compressor is made up. These volumes that do not even take part in the heat transfer processes permit the connection between the different heat exchangers and the spaces of the cylinder. The designer of this kind of systems can have the possibility to distribute the dead spaces in different localisations of the thermal compressor (Fig. 9).

The dead spaces reduce the amplitudes of pressure variations. They have a negative effect on performance of the thermal compressor. However, some other effects which are difficult to apprehend have to be taken into account. In the case of Stirling engines, additional dead volumes in the compression and expansion spaces can increase both the work and the efficiency of the engine, due to a good effect on the temperature oscillations in the heat exchanger and on the modification of the phase of the pressure variations [8].

The presence of dead volumes has complex consequences which are difficult to apprehend. The purpose of this section is to bring out this complexity when a particular design of the thermal compressor is considered. So different supplementary adiabatic dead volumes (KC, RK, HR, EH) are now considered. Unlike the heat exchangers (K, R, H) which are divided into several control volumes, each adiabatic dead space (KC, RK, HR, EH) is modelled as only one control volume. The equations of mass (1) and energy (5) balance presented previously are also applied to each dead control volume.

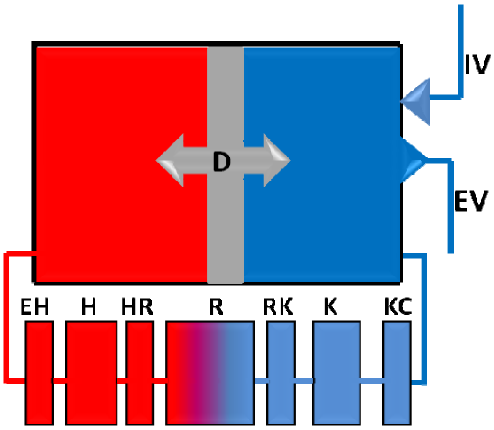
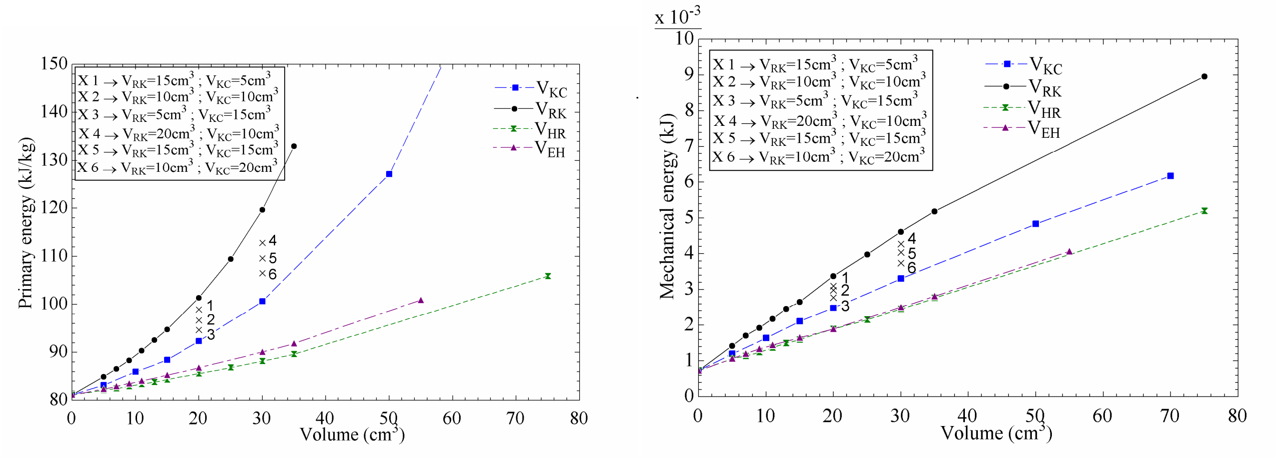


Fig. 9. The thermal compressor with possible supplementary dead volumes placements.

Figures 10 and 11 represent the effect of additional dead volumes on primary energy consumption per unit of mass delivered, on mechanical energy used to move the displacer, on the quantity of the fluid compressed per cycle and on the discharge temperature. In these figures, the curves V_{EH}, V_{HR}, V_{RK} and V_{KC} correspond to supplementary dead volumes considered alone. For instance, the curve V_{EH} is related to simulation results for which V_{HR}, V_{RK} and V_{KC} are zero. In addition 6 points have been added corresponding to combinations between dead volumes V_{RK} and V_{KC}.

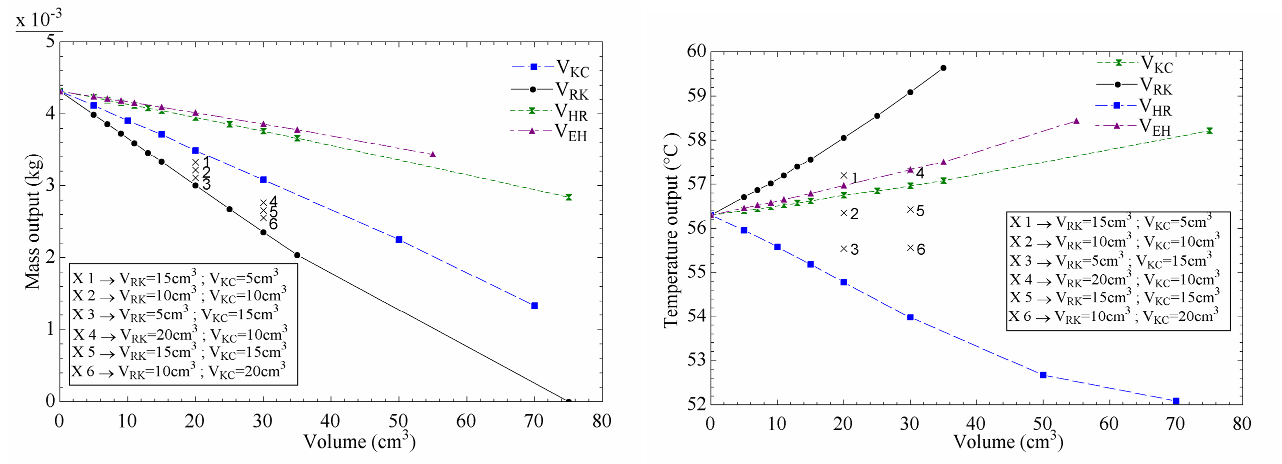


(a) (b)

Fig. 10. Energy consumed: a) primary energy per mass of compressed fluid, b) mechanical energy of the displacer per cycle.

The primary energy consumed to compressed one kilogram of CO_2 increases with the increase of the size of the dead volumes as it is shown in Fig. 10 (a). However, the negative effect of the dead volume is different according to its placement in the thermal compressor. As it can be seen, the thermal compressor consumes less primary energy when dead volumes are placed in the hot part (V_{EH}, V_{HR}) than in the cold part (V_{RK}, V_{KC}). In the hot part of the thermal compressor, there is little difference in primary energy consumption if the dead volume is placed between the hot part of the cylinder and the heater (V_{EH}) or between the heater and the regenerator (V_{HR}). In the cold part, a dead volume placed between the regenerator and the cooler (V_{RK}) has a more negative effect than a dead volume placed between the cooler and the cold part of the cylinder (V_{KC}).

Figure 10 (b) presents the mechanical energy per cycle needed to move the displacer. The presence of dead volumes also increases the consumption of mechanical energy. As previously, this consumption is lower if dead volumes are placed in the hot part than in the cold one of the thermal compressor. Almost the same quantity of mechanical energy is needed if a dead volume is placed between the heater and the hot part of the cylinder (V_{EH}) or between the heater and the regenerator (V_{HR}).



(a) (b)

Fig. 11. Characteristics of the output: a) mass of fluid delivered, b) temperature of the fluid delivered.

Figure 11 (a) shows that the mass of fluid delivered per cycle decreases as the size of the dead volumes increases. Since dead volumes placed in the hot part of the thermal compressor contains a lower quantity of fluid, its negative effect is less significant than dead volume placed in the cold part. On the hot side of the compressor, it is worth noting that the dead volume localization (V_{HR}) which causes the more important decrease of the mass output is not the localization (V_{EH}) which implies the greatest primary energy consumption per unit mass delivered.

The temperature of the fluid at the discharge changes with the size and the placement of the dead volume as illustrated in Fig. 11 (b). The discharge temperature decreases as the size of the dead volume only in the case where the dead volume is placed between the compression space C and the cooler K (V_{KC}). In the other cases, the temperature of the discharge fluid increases as the size of the dead volumes increases.

4. Conclusion

A new concept of thermal compressor is presented. The thermal compressor is identical to a gamma type Stirling engine of which the power piston and cylinder have been replaced by automatic inlet and exhaust valves. The working fluid of the power cycle is the same as the fluid to compress. The thermal compressor is especially suited for supercritical CO_2 heat pump applications. Some modelling results are presented with a special attention to the regenerator and the influence of the piston rod diameter on the energies exchanged and the quantity of compressed CO_2 delivered. The impact of the leak in the thin annular gap between the displacer and the cylinder wall is emphasized and it is shown that, even if the relative mass flow rate of CO_2 in the annular gap is very low, the consequence of this flow on the compressor performance is very important. Finally the complex influence of the dead spaces size and distribution on the thermal compressor performance is presented.

Acknowledgments

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Nomenclature

A heat transfer area, m^2

Cp specific heat at constant pressure, J/(kg K)

h enthalpy, J/kg

H_T heat transfer coefficient, $W/(m^2 K)$

k thermal conductivity, W/(m K)

L length, m

m mass of fluid, kg

$\dot{m}$ mass flow rate, kg/s

P Pressure, Pa

Q heat power, W

q thermal energy per unit mass, J/kg

T Temperature, °C

u internal energy, J/kg

V volume, m^3

v specific volume, m^3/kg

W mechanical power, W

w mechanical energy per unit mass, J/kg

Greek symbols

β crank angle, degrees

ρ density, kg/m^3

Subscripts

C compression space

cond conduction

D upper part of the displacer (side E)

DC lower part of the displacer (side C)

E expansion

EV exhaust valve

H heater

i i^{th} control volume in H, R or K ($1 \leq i \leq n$)

in entering the thermal compressor

IV inlet valve

K cooler

l left

m matrix

n number of control volume

out flowing out of the thermal compressor

prim primary (energy)

r right

R regenerator

RH dead space between regenerator and heater

RK dead space between regenerator and cooler

tot total

w wall

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