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On The Strategies Towards Isothermal Gas Compression And Expansion

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On The Strategies Towards Isothermal Gas Compression And Expansion

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ABSTRACT

Isothermal compression/expansion is regarded as the most promising process in many applications and many researchers and inventors have tried different methods to achieve this goal. The current article first studies the gradual roadmap from adiabatic towards isothermal process from thermodynamics and heat transfer point of view. Different strategies are investigated to achieve this goal by evaluating different possibilities; the bottleneck of the problem is then identified in reciprocating pistons: most of the compression work (and hence heat generation) occurs at the end of the compression stroke, however the heat transfer surface is at its minimum by the time that a high overall heat transfer rate is most needed. Then increment of heat exchange surface (specially at this important time interval) is being focused on as the most promising solution. Based on such a concept, a novel kind of reciprocating air compressor is being developed, in which quasi-isothermal compression is achieved by increasing heat transfer through co-axial fins.
Finally a global map is illustrated to show the starting point, proposed solution standing point and the ideal situation and its requirement.

1. INTRODUCTION

When the gas is compressed rapidly with minimal heat transfer, significant energy is converted into increasing the gas temperature. As the compressed gas cools in the storage reservoir the energy content of gas decreases. The same argument is true during the expansion: in rapid expansion gas cools down and in absence of efficient heat transfer to the gas, only a part of its energy can be extracted. These shortcomings reduce the round trip efficiency of the cycle.

Current compressor and expander, provide poor heat transfer between mechanical boundaries and gas. The heat transfer is poor in these machines due to the high-speed operation and practical geometry requirements that dictate a poor surface area to volume ratio.

In the reciprocating piston compressors literature there exist many works that have investigated the heat transfer. Adair (1972) is one of the pioneers of such investigations, that has introduced one of the most well known relations in this regard. Generally, as Rao and Bardon (1985) indicate heat transfer in a compressor or expander is quiet complex because of the varying flow regimes within the cylinder and varying convection coefficient.
Also, Annand (1980) and Kornhauser (1989) point out that the other complexity is that heat transfer can be out of phase with the temperature difference between the bulk gas and the wall in a compressor and expander. The main reason that the bulk gas temperature does not closely follow the wall temperature is that the surface area to volume ratio in reciprocating piston systems is generally quite low.

There have been efforts to approach isothermal conditions using different techniques. One approach is to use multi-stage processes with intercooling. But the additional required equipments is relatively expensive and increase the size of the system considering the number of cylinders, valves and heat exchangers, which are required in multi-stage units. Furthermore, external heat exchangers introduce additional pressure drop and thermal difference.

These problems have motivated the researchers to develop compressors that perform the heat exchange in the working chamber. The machines designed for this purpose can be categorized in two groups:

- Machines that produce isothermal conditions using only a dry mechanism, which work only with pure air.
- Machine that involve liquid in compression or expansion process

Typical machines of first category include a small diameter cylinder with a long stroke or a large diameter cylinder with a short stroke.

Machines of the second category try to benefit from water as an external agent to absorb the heat of compression. Coney *et al*. (2002) developed a novel compressor using water injection. They claimed to reach quasi-isothermal compression benefiting from of increasing the heat transfer surface area of water droplets by spraying it through nozzles as well as the high density and heat capacity of liquid water. However, the drawback is the complexity of the system and additional equipments needed for water spray and separation of water and air at downstream.

Another example of the second category is liquid piston. As Van de Ven (2009) describes, in this method liquid column is utilized to directly compress the gas and is used as a medium to carry heat into an out of compression chamber. One of the industrial realized examples of liquid piston is the work by Lemofouet and Rufer (2005). In their system, the liquid piston is controlled with a valve by switching the inlet and outlet of the hydraulic valve between two chambers. In this arrangement, one liquid piston is emptying while the other is filling. This system will be discussed in more details in section 3. However, liquid piston has its own drawbacks: Firstly, in high pressures air can be solved in liquid and possibly causing cavitation in low-pressure areas of the system. Second, the liquid can leave the gas chamber when exhausting the gas through valves.

2. FROM ADIABATIC TO ISOTHERMAL COMPRESSION

To better understand a Compressed Air Energy Storage System (CAES) it is instructive to show the processes that comprise such a storage system. Figure. 1 shows the Pressure-Volume Diagram of a CAES system. At point 1 the air from atmospheric pressure enters the compressor, reaches to reservoir pressure at point 2 and at point 3 transfers to reservoir at the final pressure. During the expansion phase it enters an expander and reaches to point 4 and then to point 1 again. Figure 1 (a) is a representation of a classic compression/expansion system. If we assume compression and expansion to be “quasi-static“ processes, the area between the vertical axis and the compression curve (1-2-3) shows the work consumed by the compressor, while the area between the vertical axis and expansion curve (3-4-1) represents extractable work by expander. Since the ratio of the extracted work is relatively negligible compared to work provided to compressor, the overall round-trip efficiency is quiet low. In Figure 1 (b) heat transfer enhancement has made it possible to save a part of the input work during compression, and to increase the work output during expansion. In Figure 1 (c) the idea is to perform both compression and expansion as close as possible to isothermal process. Like this the work of expansion will get closer to that of compression and the round-trip efficiency will approach unity.

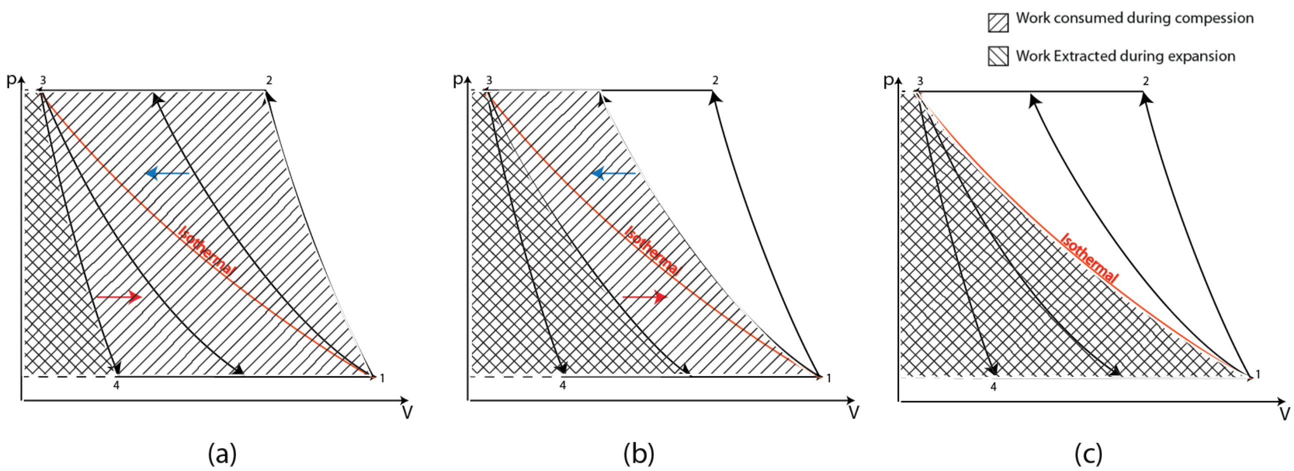

Figure 1: Pressure-Volume Diagram during compression and expansion phase.

Figure 2 shows the effect of heat transfer on work input to a compressor. According to the first law of thermodynamics, the power input to a compressor may be written as the sum of heat transfer rate rejected to ambient plus the enthalpy change difference:

$$\dot{W}_{in} = \dot{Q}_{out} + \Delta\dot{E} \quad (1)$$

Integrating over one cycle of compressor performance, one can calculate work input, heat rejected and difference of input and output enthalpy in the piston. If one changes the polytropic factor from 1.4 (adiabatic compression) to 1 (isothermal process) the parameters in equation (1) will change according to Figure 2 from right to left. It is observed that in the adiabatic case, however the heat transfer is zero, but the delta enthalpy is at its maximum. By decreasing the polytropic factor heat transfer increases but the decrease in enthalpy difference is more dominant, resulting in reduction in total work. The same argument is true in an expander, which means maximizing the heat transfer (isothermal expansion) can maximize the output work.

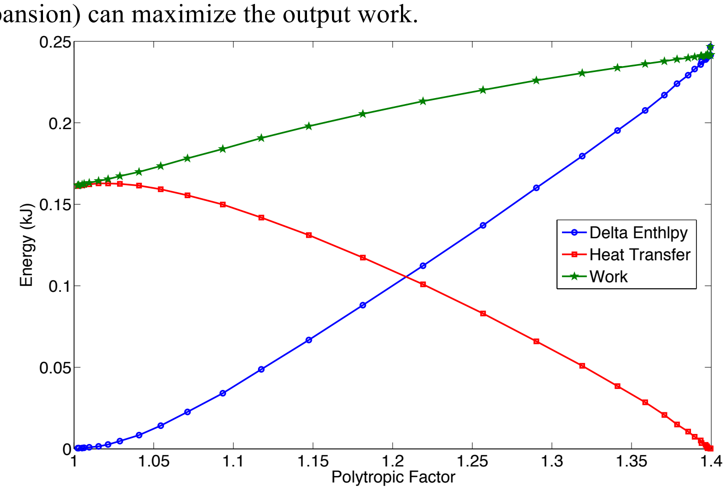

Figure 2: Delta Enthalpy, heat transfer rate and work change with Polytropic factor.

In a similar plot, Figure 3, Illustrates the change in key heat transfer parameters when the polytropic factor changes from 1.4 (Adiabatic) to 1 (Isothermal) process. The red line is the position of a classic piston compressor and the blue line is the position of an ideal isothermal compressor. One can observe that isothermal efficiency of a classical piston is only 78% because its average overall heat transfer coefficient is very low (1.42 W/K). This corresponds to the Nusselt number of 86.6. The average outlet temperature is very high (446 K). Thus the work input required will be 0.23 (KJ). The last plot has been seen already in Figure 2. By increasing the overall heat transfer coefficient (Heat transfer coefficient × Heat transfer Area) it is possible to increase the isothermal efficiency while reducing the outlet temperature and required input work. This corresponds to moving in the direction of arrow. One may notice that that while the values like isothermal efficiency, input work or average temperature change almost linearly, the overall heat transfer coefficient increases logarithmically. This is because when the process approaches isothermal, the temperature difference decreases and to compensate the overall heat transfer coefficient needs to increase drastically. The same trend is shown on a P-V and T-S diagram in Figure 4 and Figure 5. One can see in Figure 4 that as the process changes gradually from adiabatic (red) to isothermal (blue), the surface enclosed by P-V diagram (which represents the required work input) decreases. At the same time in Figure 5 the surface enclose by T-S diagram which is the energy wasted to heat generation decreases to almost zero for isothermal processes.

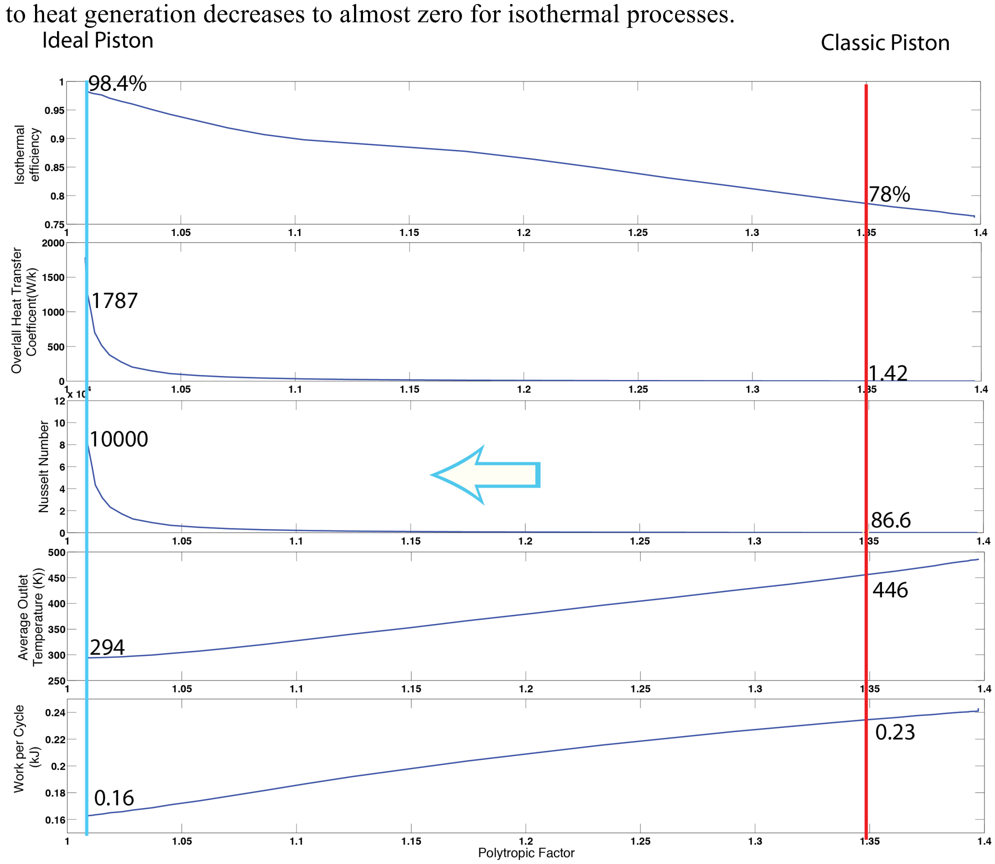

Figure 3: Gradual road map form adiabatic to isothermal.

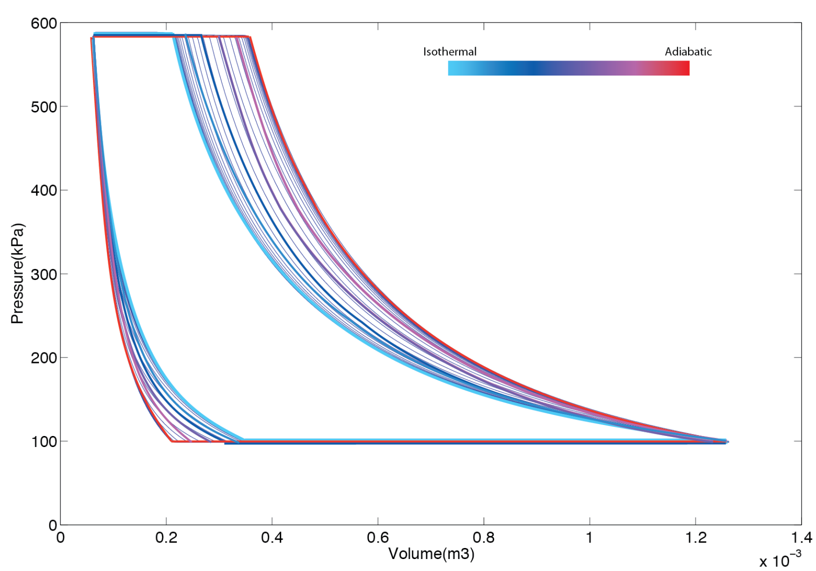

Figure 4: P-V diagram of a compression process while going from adiabatic to isothermal.

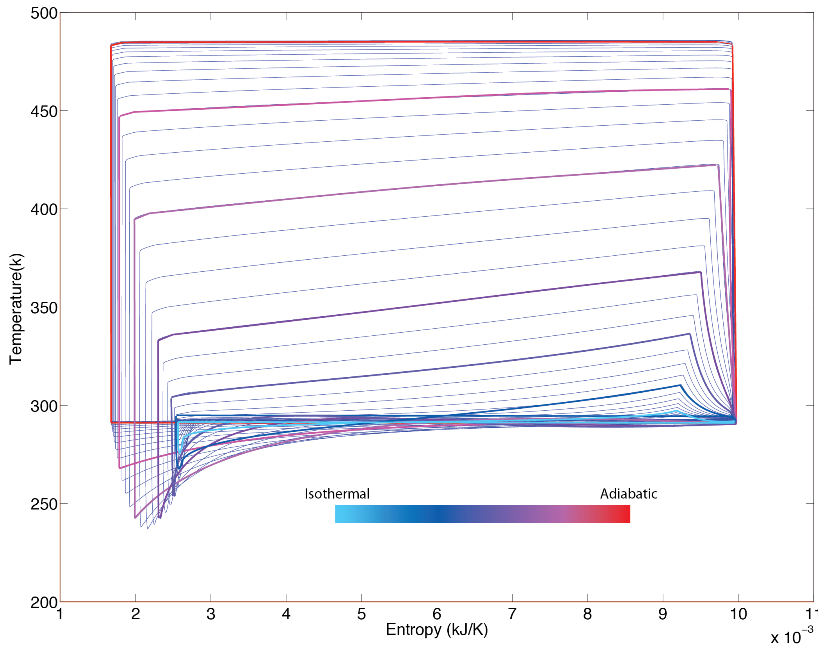

Figure 5: T-S diagram of a compression process while going from adiabatic to isothermal.

3. HEAT TRANSFER CONSIDERATIONS

As discussed in section 1 a general approach to model the heat transfer between gas and cylinder wall can be described as:

$$\dot{Q} = \mathrm{U}_1 A(T - T_w) = \mathrm{h}_1 A(T - T_w) \tag{2}$$

Where U_1 is the overall heat transfer coefficient and h_1 is the convective heat transfer coefficient that can be described by a well-known relation by Adair (1972)

$$h_1 = 0.053\frac{k_a}{D_h}(\mathrm{Re})^{0.8}(\mathrm{Pr})^{0.6} \tag{3}$$

Where $\mathrm{k_a}$ is the air conductivity coefficient and $\mathrm{D_h}$ is hydraulic diameter of cylinder.
In equation (3), Reynolds number in turn is a function of piston speed $\dot{\mathrm{x}}_\mathrm{p}$ and hydraulic diameter of cylinder $\mathrm{D_h}$ and air properties

$$Re = \frac{\rho \dot{x}_p D_h}{\mu} \tag{4}$$

Prandtl number is also dependent on air properties

$$Pr = \frac{\mu c_p}{k_a} \tag{5}$$

Similarly, one can represent the heat transfer using the heat transfer resistance. The Thermo-Electric Analogy for a reciprocating piston is discussed in details in another publication by Heidari *et al.* (2014b), but simply for a reciprocating compressor can be illustrated in Figure 6. In this analogy, heat is being generated as the gas pressure increases. This heat need to pass first a convection resistance $\mathrm{R_a}$, then it should pass through the conduction resistance R_b and finally will be convected to ambient air through resistance R_c. However since the heat transfer is not steady state a part of heat may be stored in the middle of the way in the cylinder wall body, which is represented by a parallel capacitor.

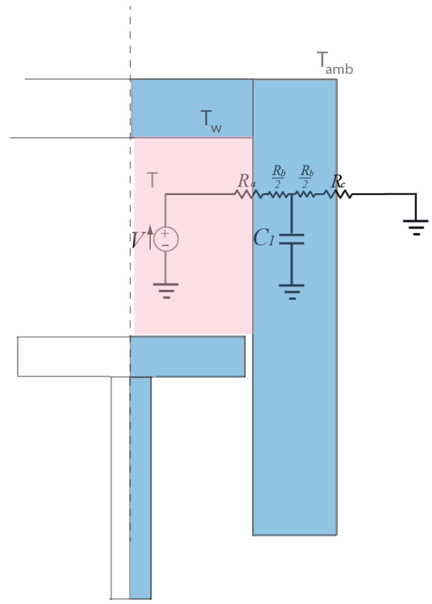

Figure 6: Thermo-Electric Analogy for a reciprocating compressor.

Considering the above mention analogy, the convection heat transfer resistance between the air and cylinder can be represented as:

$$R_a = R_{Conv} = \frac{1}{h_1 A} \tag{6}$$

Regarding Conduction in the cylinder wall, for the metallic cylinder in cylindrical coordinates with inner diameter of D_1 and outer diameter of D_2 and conductivity k_b, thickness l, the conduction resistance can be written as:

$$R_b = R_{Cond,Metal} = \frac{\ln(D_2/D_1)}{2\pi l k_b} \tag{7}$$

Also for convection between cylinder wall and ambient air, assuming $h_2 = 5\ \mathrm{W/m^2K}$ for natural convection for air, the convection resistance between metal and ambient air can be written as:

$$R_c = R_{Conv,amb} = \frac{1}{h_2 A} \tag{8}$$

The values of R_a, R_b and R_c are shown in Figure 7 for one cycles. Because the values of air velocity and the inside area of cylinder surface changes alternatively over a cycle R_a is variable. It is observed that R_b value is very low compared to R_a and R_c. Also R_c can be decreased further by applying a water jacket around the cylinder, so it seems the bottleneck in heat transfer enhancement is R_a.

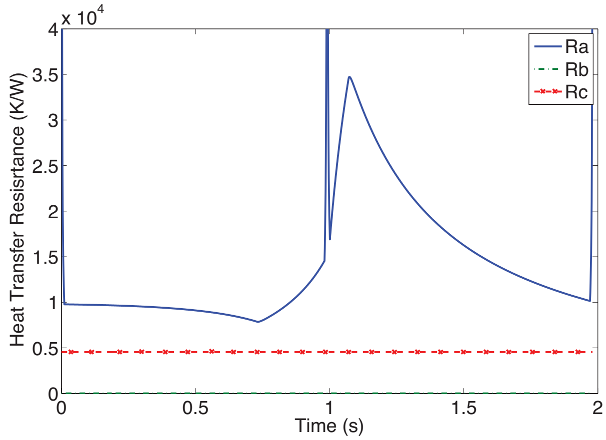

Figure 7: Thermal resistance change over three period of a cycle.

This fact has been shown in Figure 8. Similar to the idea illustrated by Cengel (2010), The passengers want to leave the island, and however there are enough ships ready in the harbor to take them to the mainland; the number of buses is not sufficient!

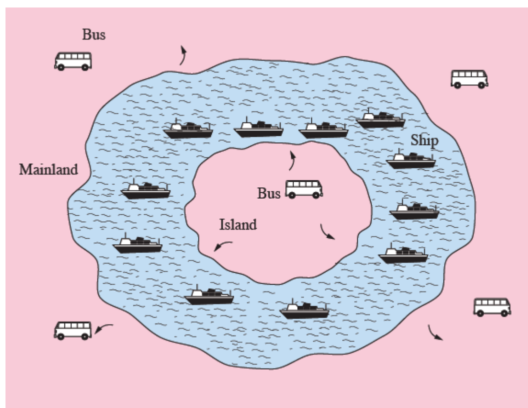

Figure 8: Analogy between low convection heat transfer and shortage of bus in the island.

The root of this problem is that in a classic reciprocating compressor (Figure 9), most of the heat generation (and hence temperature gradient) occurs at the end of the compression stroke, however the heat transfer surface is at its minimum by the time that a high overall heat transfer rate is most needed (Figure 10).
In section 4, a solution will be proposed to overcome this problem.

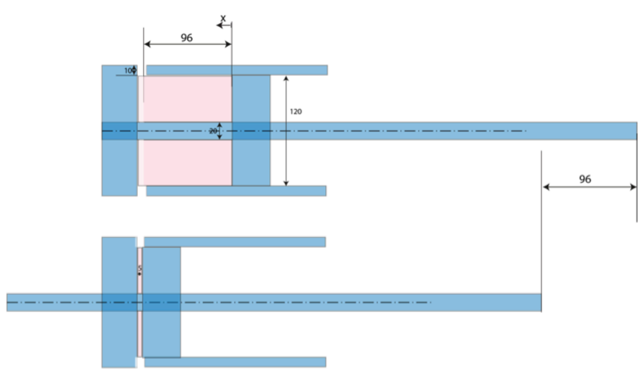

Figure 9: Classic piston positions at BDC and TDC

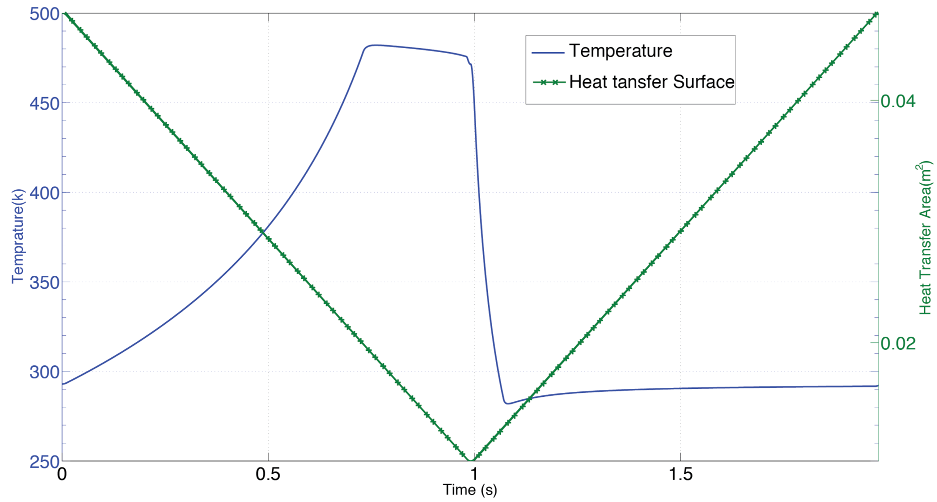

Figure 10: Gas temperature and heat transfer surface from gas to cylinder wall in classic compressor.

4. PROPOSING THE SOLUTION

Back to eq. (2), in order to maximize the heat transfer, one should increase each of the coefficients: $\mathrm{h_1}, A$ and $(\mathrm{T}-\mathrm{T_w})$. Among these factors, $(\mathrm{T}-\mathrm{T_w})$ naturally decreases if we approach towards isothermal conditions, so this parameter is out of our hands. In order to compensate this decrease the other two parameters ($\mathrm{h_1}$ *and* A) should increase drastically. So far, it is clear that heat transfer surface should increase. But if we look closely again to eq. (3) all the composing parameters are air or metal properties (ρ, μ, k_a, c_p) that are out of our hands, except for $\mathrm{D_h}$, and $\dot{x_p}$. $\dot{x_p}$ is initially a function of piston linear velocity, and however initially it seems that increasing piston velocity will lead to higher heat transfer, it is not the case in reality because higher velocity will not give the system enough time to reject heat to ambient sufficiently. But as it will be seen later the gas velocity can be increased by some techniques keeping the piston velocity low. So far the only remaining parameter is diameter. However $\mathrm{D_h}$ exists in the denominator of eq. (4), but finally heat transfer coefficient is proportional to inverse of $\mathrm{D_h}$ to the power of 0.2. Hence, if one can benefit from decreasing the hydraulic diameter to increase the heat transfer surface at the same time, the overall heat transfer coefficient can be increased drastically. This motivates us to compress the air in multitude of channels with smaller diameter, instead of compressing it in a single chamber with big diameter.

4.1 Liquid Piston

In order to implement such a concept, the first solution realized by LEI at EPFL was using a new concept of water-hydraulic gas compression/expansion called "liquid piston" (Figure 11 left). Because a liquid can conform to an irregular chamber volume, the surface area to volume ratio in the gas chamber can be increased using a liquid piston. This creates near-isothermal operation, which decreases energy lost to heat generation. A liquid piston eliminates gas leakage and replaces sliding seal friction with viscous friction. The liquid can also be used as a medium to carry heat into and out of the compression chamber. Besides introducing directly integrated heat exchangers eliminates the cost and size problems of external heat exchangers and the pressure drop and temperature difference associated with them. Moreover, water has a high heat capacity and density and thermal convection when it moisturizes the walls.

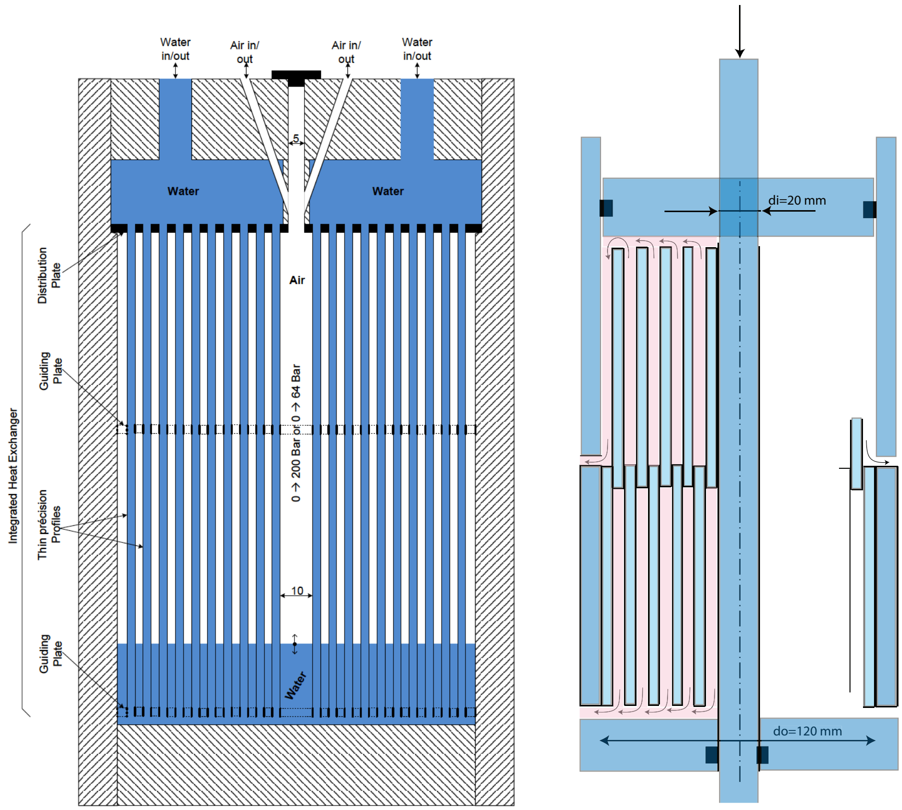

Figure 11: Liquid piston geometry (Left) and Dry (Finned) piston geometry (Right).

However, there are a few complexities of the liquid piston gas compression system that need to be addressed. First, because a gas and a liquid are in direct contact at high pressures, a portion of the gas will become entrained in the liquid, causing cavitation in low pressure areas of the system.
Another possible issue that can arise from a liquid piston gas compression system is liquid leaving the gas chamber when exhausting the gas through valves.

4.2 Dry (Finned) Piston

To solve the problems associated with liquid piston technology like solution of high-pressure air in liquid and the complexity of the design, a new concept has been proposed recently in LEI, called "Dry Piston" to eliminate the water and complexities and equipments associated with it. In this design, a series of concentric metal annuli's are used to fit into each other to compress the air. These to sets of fines are sized so that the two sets nest and mesh with each other during the operation (Figure 11 right). The details of the design are disclosed by Heidari and Rufer (2013) in details. The advantage of such a design is double: not only the hydraulic diameter of the compression chamber has been decreased, but also the entire heat transfer surface is usable during the whole cycle. This fact has been shown in Figure 12. This shows that the heat transfer surface of a finned piston is 5 times more at BDC and 18 times more at TDC compared to a classic piston. Also one can compare the heat transfer surface of liquid piston to these two. However, the liquid piston has an equal heat transfer surface in TDC, but when it is important for heat transfer in BDC, its surface decrease to great extent. This shows the advantage of finned (dry) piston over liquid piston from this point of view. Another advantage of using the finned piston is that the hydraulic diameter will be 0.005 m (Figure 9-b) instead of 0.120 m (Figure 9-a) for a classic piston.

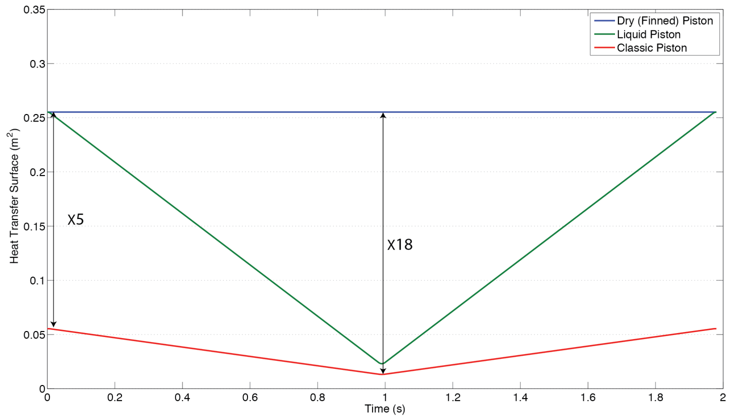

Figure 12: Heat transfer area comparison during one cycle

5. INTER-FIN HEAT TRANSFER

Figure13. shows the finned piston with symbolic dimensions (The space between the fins is represented bigger than it is). The top and bottom sets of chambers are designed in a non-symmetric way, since the radial collecting channel is placed differently. This fact leads to a pressure difference between the top and bottom chambers (Figure 14). As described by the authors in another publication, (Heidari and Rufer, (2014a)) this causes a mass flow between mass flow between the two chambers, that can help in convection heat transfer in the very small gap between the fins (0.1 mm). Since the surface area of this gap is very small, this induces a relatively high speed flow as seen in Figure 15.

What makes the inter-fin heat transfer more advantageous is that hydraulic diameter will decrease to 0.0001 m (=0.1 mm), which increases the heat transfer coefficient (according to Eq. 3) even more.
This flow becomes more and more important as we reach the end of the compression stroke and becomes the dominant mode of heat transfer between $t = 0.8$ to $t = 1$. (Qd_{12} and Qd_{21} are heat transfer in the chambers, Qd_{14} and Qd_{24} are inter-fin heat transfer and Qd_{23} is the axial heat transfer in Figure 16). This phenomenon can enhance the total heat transfer to a great extent (Figure 17).

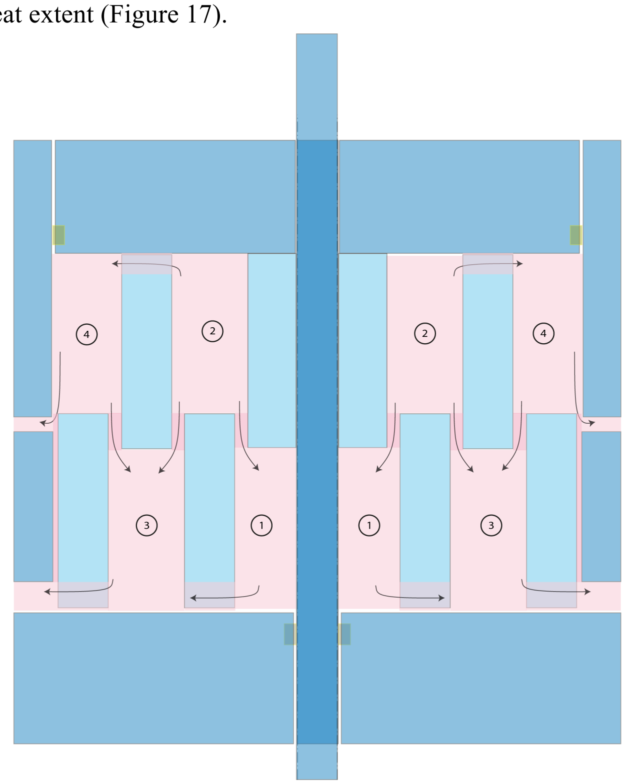

Figure 13: Fluid flow direction in dry (finned) piston

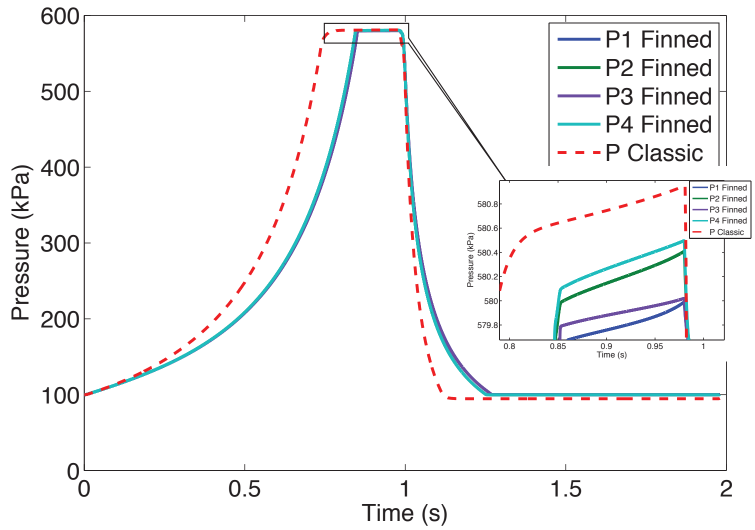

Figure 14: Pressure evolution comparison.

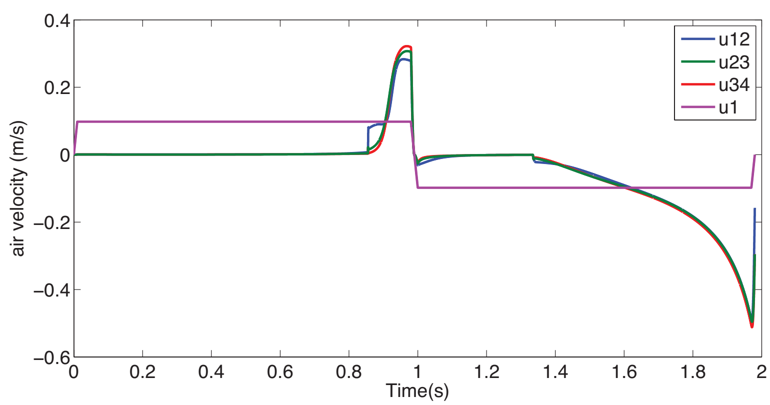

Figure 15: Air Velocity.

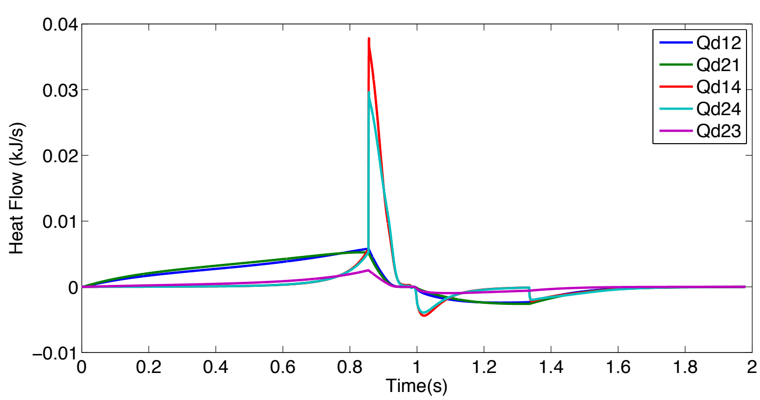

Figure 16: Heat transfer rate comparison.

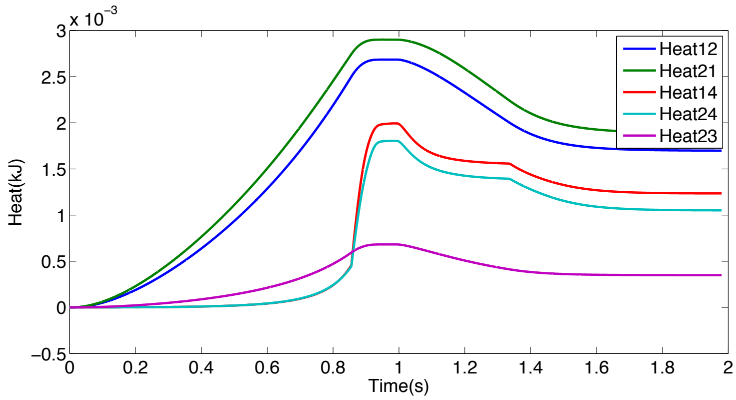

Figure 17: Heat transfer comparison.

Benefiting from an intelligent design and taking into account the enhanced heat transfer the finned piston can be placed to a point with $n = 1.14$. This corresponds to an average overall heat transfer coefficient of 16.8 W/K which is almost 15 times more than a classic piston. Noticing the second plot in Figure 18 shows that approaching toward higher isothermal efficiencies is very difficult, because the overall heat transfer coefficient increases logarithmically toward isothermal, and an ideal piston needs almost 100 times more overall heat transfer coefficient. This will mean 100 more time fins in the same volume, which is practically impossible. On the other hand the viscous dissipation in such a configuration will also increase dramatically. This balance of heat transfer to viscous forces and also construction costs requires a design optimization, which is beyond the scope of this work and will be investigated in future.

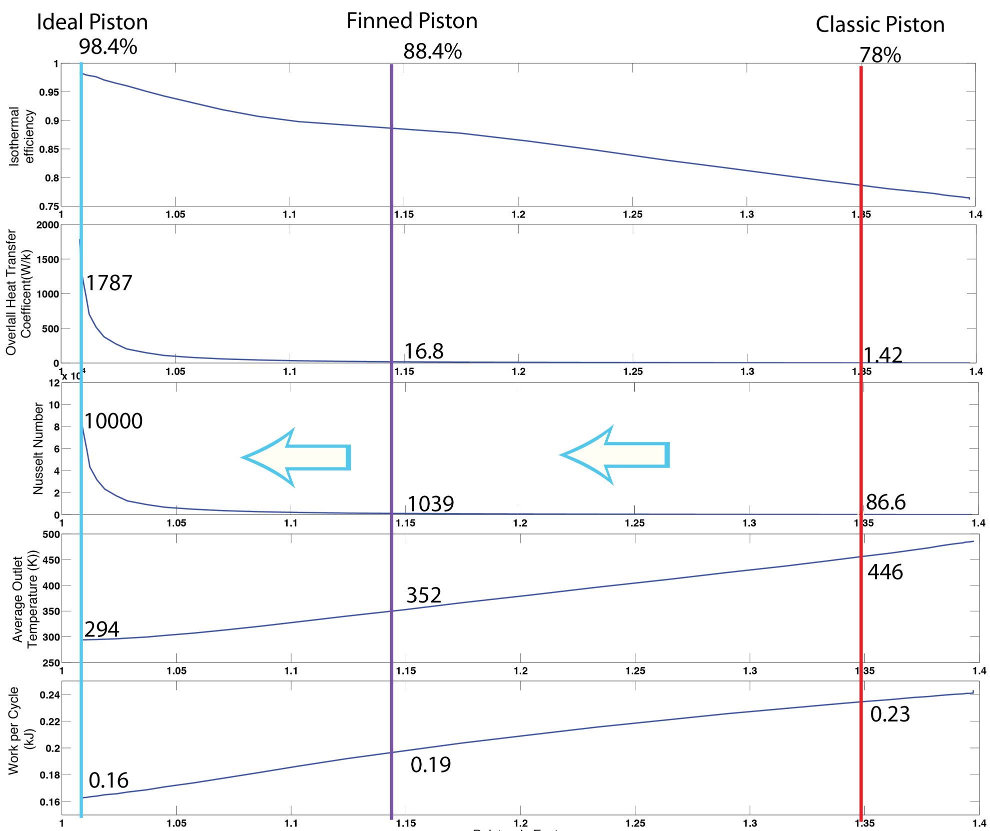

Figure 18: Placement of Finned piston in the roadmap.

6. CONCLUSIONS

The gradual improvement from adiabatic to isothermal process in compression and expansion is discussed from different aspects. The heat transfer in such devices was studied and the bottleneck of low heat transfer was revealed. Heat transfer area and chamber diameter were identified as the parameters that can increase the heat transfer and a design that benefits from the positive influence of both factors is introduced: Finned piston.

- Dry (finned) piston exploits the entire heat transfer surface throughout the whole compression/expansion process without using water and equipment's associated with it and operates simply like a normal compressor.
- The design of both dry (finned) and liquid piston requires a careful balance between convection heat transfer coefficient and the viscous dissipation. Decreasing the diameters (gaps) of the cylinder increases convective heat transfer while negatively impacts the viscous pressure drop.
- Further approach toward isothermal conditions is very difficult since for increasing isothermal efficiency or decreasing work linearly, the overall heat transfer coefficient needs to be increased logarithmically.

NOMENCLATURE

TDC	Top Dead Center	(-)
BDC	Bottom Dead Center	(-)
U	Overall heat transfer coefficient	(W/ k)
Q	Heat Transfer	(kJ)
$\dot{Q}$	Heat Transfer rate	(kJ/s)
W	Work	(kJ)
$\dot{W}$	Power	(kJ/s))
E	Enthalpy	(kJ)
$\dot{E}$	Enthalpy flow	(kJ/s)
m	Mass	(kg)
$\dot{m}$	Mass flow	(kg /s)
V	Volume	(m^3)
$\dot{V}$	Volume change	(m^3 /s)
T	Temperature	(k)
p	Pressure	(kPa)
ρ	Density	(kg/m^2)
D	Diameter	(m)
t	Time	(s)
h	Heat transfer coefficient	(W/m^2.k)
Re	Reynolds number	(-)
Pr	Prandtl number	(-)
μ	Viscosity	(kg/m.s)
c_p	Specific heat at constant pressure	(kJ/kg.K)
R_g	Gas constant	(kJ/kg.K)
k	Heat conductivity	(W/m.k)
A	Heat transfer Area	(m^2)
ϕ	Electric potential	(V)
I	Electric current	(A)
R	Resistance	(Ω)
C	Capacitance	(F)
l	Length	(m)
X	Position	(m)
$\dot{X}$	Speed	(m /s)

Subscript

in	inlet
out	outlet
a	air
b	body of metal

p	piston
w	wall
h	hydraulic
amb	ambient
cond	conduction
conv	convection

REFERENCES

Adair, R.P., Qvale, E.B. and Pearson, J.T., 1972. Instantaneous heat Transfer to the Cylinder Wall in Reciprocating Compressors, Proc. of the *Purdue Compressor Technology* Conf, West Lafayette, IN, USA. 521-526.

Annand WJD, Pinfold D., 1980, Heat transfer in the cylinder of a motored reciprocating engine. *SAE preprints (800457).*

Benson GM, 1984, Isothermalizer system. United States patent: 4446,698.

Çengel, Y. A. and Ghajar, A., 2011, *Heat and Mass Transfer,* McGraw-Hill, pp. 135–293.

Coney, M.W., Stephenson, P.L., Malmgren, A., Linnemann, C., Morgan, R.E., Richards, R.A., Huxley, R. and Abdallah, H., 2002, Development Of A Reciprocating Compressor Using Water Injection To Achieve Quasi Isothermal Compression, International Compressor Engineering Conference, Purdue University.

Heidari, M., Rufer, A., 2014a, Fluid Flow Analysis of a New Finned Piston Reciprocating Compressor Using Pneumatic Analogy, *FDTT2014*, Antalya, Turkey.

Heidari, M., Rufer, A. and Thome, J. R., 2014b, Thermoelectricity Analogy Method For Computing Transient Heat Transfer In A New Reciprocating Finned Piston Compressor, 15th International Heat Transfer Conference, *IHTC-15*, August 10-15, Kyoto, Japan.

Kornhauser AA. 1989, *Gas-wall heat transfer during compression and expansion*. PhD thesis, Massachusetts Institute of Technology, Cambridge, MA.

Lemofouet S, Rufer A., 2005, Hybrid energy storage systems based on compressed air and supercapacitors with maximum efficiency point tracking. *11th European conference on power electronics and applications, Dresden, Germany.*

Rao VK, Bardon MF, 1985. Convective heat transfer in reciprocating engines. *Proc Inst Mech Eng*, Part D: Transport Eng;199:221–6.

Van de Ven, J.D., Li, P.Y., 2009, Liquid piston gas compression, Appl Energy, doi:10.1016/j.*apenergy*.2008.12.001