



**QUEEN'S
UNIVERSITY
BELFAST**

Thermodynamic and heat transfer analysis of a Liquid Piston Gas Compressor (LPGC)

Khaljani, S., Mahmoudi, Y., Murphy, A., Harrison, J., & Surplus, D. (2019). Thermodynamic and heat transfer analysis of a Liquid Piston Gas Compressor (LPGC). In *The International Conference on Innovative Applied Energy (IAPE'19): Proceedings* [270] [http://iape-conference.org/Downloads/Proceedings/Articles%20\(Abstracts%20&%20Papers\)/a-1-Article-270.pdf](http://iape-conference.org/Downloads/Proceedings/Articles%20(Abstracts%20&%20Papers)/a-1-Article-270.pdf)

Published in:

The International Conference on Innovative Applied Energy (IAPE'19): Proceedings

Document Version:

Peer reviewed version

Queen's University Belfast - Research Portal:

[Link to publication record in Queen's University Belfast Research Portal](#)

Publisher rights

Copyright 2018 Springer.

This work is made available online in accordance with the publisher's policies. Please refer to any applicable terms of use of the publisher.

General rights

Copyright for the publications made accessible via the Queen's University Belfast Research Portal is retained by the author(s) and / or other copyright owners and it is a condition of accessing these publications that users recognise and abide by the legal requirements associated with these rights.

Take down policy

The Research Portal is Queen's institutional repository that provides access to Queen's research output. Every effort has been made to ensure that content in the Research Portal does not infringe any person's rights, or applicable UK laws. If you discover content in the Research Portal that you believe breaches copyright or violates any law, please contact openaccess@qub.ac.uk.

Thermodynamic and heat transfer analysis of a Liquid Piston Gas Compressor (LPGC)

Mansoureh Khaljani
South West College
Burn Rd, Cookstown BT80 8DN, United Kingdom
Queen's University Belfast
School of Mechanical & Aerospace Engineering
BT9 5AH, United Kingdom
(0)7568775305. +44
mansoureh.khaljani@swc.ac.uk
skhaljani01@qub.ac.uk

Adrian Murphy
Queen's University Belfast
School of Mechanical & Aerospace Engineering
BT95AH, United Kingdom
(0)28 9097 4095. +44
a.murphy@qub.ac.uk

Yasser Mahmoudi
Queen's University Belfast
School of Mechanical & Aerospace Engineering
BT9 5AH, United Kingdom
(0)28 9097 5495. +44
s.mahmoudilarimi@qub.ac.uk

John Harrison
South West College
Burn Rd, Cookstown BT80 8DN, United Kingdom
(0)28 9097 5495. +44
John.Harrison@swc.ac.uk

David Surplus
B9 Energy group
5 Willow bank Rd, Millbrook Industrial Estate Larne, Co.
Antrim, Northern Ireland, BT40 2SF, United Kingdom
(0)7545 696346. +44
d.surplus@b9energy.co.uk

ABSTRACT

A Liquid Piston Gas Compressor (LPGC) is a new concept to be deployed in Compressed Air Energy Storage (CAES) systems in order to tackle the challenge of unsteady energy supply by renewable energies such as wind and solar. The LPGC stores the energy as compressed air in the storage tank when renewable energies are available and reuses it when they are not available. This work aims to understand the thermodynamic and heat transfer characteristics of a LPGC system in the compression process. In addition, as LPGC are more efficient while working isothermally, further investigation is undertaken on the use of porous materials in the LPGC to increase the rate of heat transfer from the compressed gas to the porous medium in the compression phase. The porous medium is considered to be a row of narrow parallel plates attached to the top section of the LPGC. The results reveal that the peak air temperature reduces with the introduction of both the five plate (by 16 K) and the nine plate (by 21 K) configurations. Introducing more plates increases the heat transfer between the high-temperature air and the plates and thus increases the plate temperatures. Additionally, by inserting plates, the compression process becomes an isothermal process. Finally, the compression efficiency is used to compare the performance of case studies. The result reveals that there are an optimum number of plates to have the optimum compression efficiency.

Keywords

Liquid Piston Gas Compressor (LPGC), Thermodynamic, Heat transfer, Porous media

1. INTRODUCTION

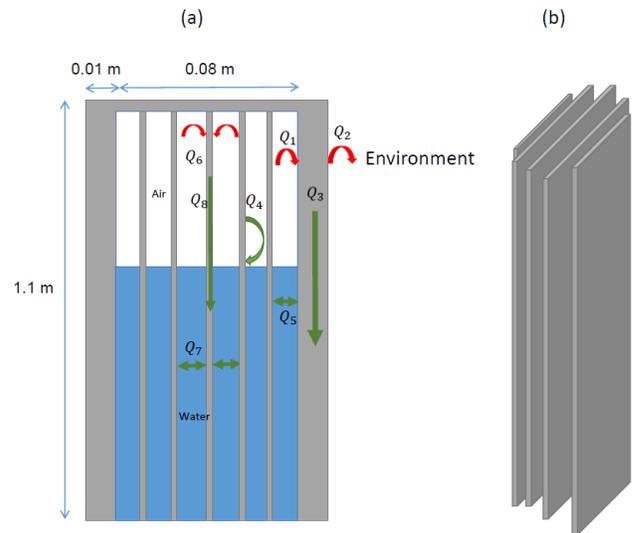
Today energy storage is a key element in the modern energy supply chain especially within the renewable energy sector as such technologies are unable to produce electricity steadily. Appropriate energy storage can make renewable energy resources more reliable, conserve fossil fuel resources and reduce the environmental impact of energy generation. Among different energy storage technologies, compressed air energy storage (CAES) is a technology that has a strong prospect of efficiency improvement.[1]. The solid piston air compressor is one of the components of a CAES system, which stores the electricity as compressed air. In this type of compressor, air is compressed in the adiabatic process so that the temperature of the compressed gas rises dramatically. After storage in a vessel tank, the high temperature compressed air is cooled at constant pressure and its thermal energy is wasted. In a CAES system, an external heat source is needed to heat the compressed air when it is needed to expand. The other drawbacks of solid piston air compressors are gas leakage and considerable friction loss during the compression phase [2]. To overcome the noted drawbacks of the solid piston air compressor process, the Liquid Piston Gas Compressor (LPGC) was introduced [3]. This type of compressor is new and the technology is still in development for commercial use [4], only a limited number of studies to date examine the heat transfer and thermodynamics of the process. In LPGCs, a liquid acts as a piston instead of a solid piston. The liquid enters the cylinder at high-pressure driven by a hydraulic pump, as the liquid fills the chamber a volume of air in the chamber is compressed to a desirable pressure.

In 2009, Van de Ven and Li [3] compared the performance of Liquid Piston gas Compressor (LPGC) and a reciprocating solid piston compressor in the same working conditions. In their study [3], the cylinder of LPGC was divided into small diameter bores to increase the heat transfer area. Results indicated that the liquid piston decreased the energy consumption by 19% over the reciprocating piston. Piya et al. [1] developed a numerical model to evaluate the influence of bore diameter and frequency on the heat transfer rate in the compression process in a liquid piston gas compressor. They considered the final gas temperature in the compression phase by studying different geometries, different gases and liquids as the working fluids. It was found that a small bore diameter cylinder, leads to the lower rise in the temperature of the working gas. However, in these studies, they did not consider the real properties of the gas and liquid. Considering these properties can thus lead to results far from reality. Other works [5-8] were conducted to improve the performance of LPGCs. In this regard, Sadat et al. [5] tried to find an optimal trajectory for a liquid piston compressor/expander. They inserted many small diameter tubes inside the chamber and used a general heat transfer model to investigate the power density and friction work as objective functions [5]. They found that the optimal profile includes an initial fast compression, then relatively slow compression and finally a rapid compression to have an optimum power density and lower friction work [5]. In another work, Zhang et al. [8] used the interrupted plate heat exchangers with different materials in a LPGC to increase the heat transfer rate from gas to the surrounding material during the compression in order to achieve a near-isothermal compression process. Results showed that the temperature distribution in an interrupted plate heat exchanger depends on the plate material and thickness. In another work, Zhang et al. [6] worked on a design for minimizing the temperature rise in the compressor during compression by using two types of porous media. They considered open-cell metal foams. Their CFD simulation results showed that the metal foam inserts are very effective in suppressing compressed air temperature rise. The metal foam reduces the air temperature to about 215 K compared to the base case in which the air temperature reach to 575 K. Although the heat transfer model was used in these studies, none of them used a detailed heat transfer model to calculate the heat transfer rate along the cylinder wall in the axial and radial direction or the heat transfer from the wall to the surrounding environment. Also, the simulations represented the ideal gas law. Zhang et al. [9] used a different approach to improve the performance of the LPGC. They focused on a design analysis of a shaped liquid piston compression chamber based on CFD. They studied the effects of varying the profile of the cross-sectional diameter along the axis of the chamber on compression efficiency. A quantitative design analysis showed that, in general, a large aspect ratio of the chamber (the length of cylinder to the maximum radius ratio) and a steep radius change of the chamber is preferred. The problem with this work is that the construction of a gourd-like shape cylinder is complicated, although this work had promising theoretical results on the improvement of the compressor. Arjimand Kermani and Rokni [10] developed a detailed heat transfer model to address the drawbacks of the previous studies mentioned above. They reported the details of heat transfer between the gas, water, wall, and surrounding. They declared that increasing the total heat transfer coefficient at the interface of the liquid and gas (10,000 times) or at the wall (200 times), leads to a 22% or 33% reduction of the hydrogen temperature, compared to the adiabatic case. They did not investigate approaches to increase the total heat transfer coefficient and how these affect the heat transfer parameters such as Reynolds and Nusselt numbers.

The main objective of this work is to investigate the detailed heat transfer and thermodynamic parameters in one stroke compression of air in a Liquid Piston Gas Compressor (LPGC) system. In order to evaluate the effect of using porous material in the LPGC on achieving a near-isothermal compression process, parallel plates are used as the porous medium. In the modeling of the systems with plates, the effects of plates on the flow regime and heat transfer parameters are considered. The results are presented and discussed in the form of thermodynamic parameters such as compression work, friction work, cooling work and compression efficiency are discussed.

2. System description

The compression phase of a Liquid Piston Gas Compressor with and without plates is simulated using one dimension heat transfer as well as thermodynamic laws in the EES Software [11]. The analysis is conducted in order to calculate the heat transfer exchange between the air, liquid, plates and cylinder wall. Figure 1 shows the schematic of a simple LPGC with plates (as the porous medium). The model is developed based on the properties of air and water as the working gas and liquid piston, respectively. According to the figure, the hydraulic pump pushes the high-pressure water into the cylinder. While the high-pressure water moves up into the cylinder, the air is compressed at the top of the cylinder.



- Q_1 : Convection heat transfer from air to the cylinder wall
- Q_2 : Convection heat transfer from wall to the surrounding
- Q_3 : Conduction heat transfer in the wall
- Q_4 : Conduction heat transfer from air to the water at the interface
- Q_5 : Convection heat transfer from water to the piston or piston to the water
- Q_6 : Convection heat transfer from air to the plates
- Q_7 : Convection heat transfer from water to the plates or from plates to the water
- Q_8 : Conduction heat transfer in the plates

Figure 1. The cross-sectional schematic of (a) compression chamber and heat transfer mechanisms between the gas, liquid, wall, plates and surrounding (b) the arrangement of plates.

In this work, the air is compressed from 8 bar to 40 bar. The hydraulic pump operates at a constant speed and supplies the constant required mass flow rate for water. It is assumed that

initially 10% of the cylinder volume is filled by water to avoid the penetration of air into the hydraulic system. The dimension of the cylinder and plates and other information is presented in Table 1.

Table 1. The dimension of compression chamber, plates and input variables in simulation

| Parameters | Value | Unit |
|---------------------------------|-------------------|------|
| Inside diameter of cylinder | 0.08 | m |
| Height of cylinder | 1.1 | m |
| Material of cylinder and plates | Stainless_AISI316 | - |
| Pump efficiency | 70 | % |
| Height of plates | 1.1 | m |
| Thickness of plates | 0.001 | m |
| Thickness of cylinder | 0.01 | m |

3. Porous inserts

A porous medium is a material containing pores. The skeletal material is usually solid, but structures like foams are often analyzed using the concept of porous media [12]. Porous media can be used within the liquid-piston compression chamber to absorb the heat from the air as it heats during compression [6]. The aim of using porous media is increasing the heat transfer surface area to increase the heat transfer rate between the high-temperature air and the surrounding structure. In the present work, a number of plates are used as a porous medium to analyze the effect of plates on the heat transfer rate of the high-temperature air during the compression phase. The flow resistance and resulting pressure drop and friction are the disadvantages of porous materials [13] which are modelled and discussed in detail in this work.

4. Thermodynamic analysis

The amount of air remains constant during the compression process because the input air valve is closed. By considering the compression chamber as the control volume, the mass balance and energy balance for the air and water are as follows [14]:

$$\frac{dU_{gas}}{dt} = \frac{\delta Q}{\delta t} - \frac{\delta W}{\delta t} \quad (1)$$

$$\frac{dU_{liq}}{dt} = \frac{\delta Q}{\delta t} - \frac{\delta W}{\delta t} - \frac{dH_{in}}{\delta t} \quad (2)$$

$$\frac{dM_{liq}}{dt} = \sum \dot{m}_{liq} \quad (3)$$

Where, $\frac{dU_{gas}}{dt}$ and $\frac{dU_{liq}}{dt}$ in are the internal energy variations of air and water during compression, respectively. The term $\frac{\delta W}{\delta t}$ is the compression work on the gas, $\frac{dH_{in}}{\delta t}$ is the enthalpy variation enters to the cylinder, and $\frac{dM_{liq}}{dt}$ is the mass flow rate of liquid. In the energy balance equation (2), $\frac{\delta Q}{\delta t}$ is the heat transfer rate of air and water with the cylinder wall and plates, and the heat transfer rate at the interface of the water and air - which is defined as follows:

$$\dot{Q}_{liq/gas} = \sum_{i=m,j=0}^{i=n,j=d} \pm \dot{Q}_{w,i,j} \pm \dot{Q}_{p,i,j} \pm \dot{Q}_{interface,i,j} \quad (4)$$

In Eq. (4), for \dot{Q}_{liq} , m is equal to 1 and n corresponds to the last node where water is in direct contact with the wall. However, for \dot{Q}_{gas} , m is equal to the node where air is in direct contact with the water and n corresponds to the total discretization number. Also, j shows the number of ducts from 0 for the base case to d which shows the final number of ducts in the chamber. This description satisfies all following equations.

5. Heat transfer analysis

In the heat transfer analysis of the air compression process, the following assumptions are considered:

- The liquid level in all ducts was considered at the same level.
- The hydraulic diameters of all ducts were considered as $D_h = 2 L_z$ [15]. L_z is the distance between plates.
- The convective heat transfer coefficient of the compression chamber surrounding environment is considered as $100 \text{ W/m}^2 \text{ K}$ [10].

The heat transfer exchange between air and liquid through their interface is as [16]:

$$\dot{Q}_{interface} = UA_{interface}(T_{liq} - T_{gas}) \quad (5)$$

$$UA_{interface} = \frac{1}{\sum R_{total}} = \sum \frac{K_{liq}A_{i,j}}{L_{liq}} + \frac{K_{gas}A_{i,j}}{L_{gas}} \quad (6)$$

In equation (6), A_i is the cross sectional area of each duct in LPGC-porous media. In the base case without porous media, A refers to the cross sectional area of the cylinder. K is the thermal conductivity, and L is the height of liquid or gas within the chamber.

Moreover, the heat transfers from high-temperature air into the wall (plates or cylinder wall) through convection, in which a part of the heat is then transferred from the wall to the water, and the rest is transferred to the surrounding environment. The heat transfer relationship between the wall and water or air and heat transfer are as [10, 16]:

$$\dot{Q}_{w,i} = UA_{w,i}(T_{w,i,j} - T_{liq/gas}) \quad (7)$$

$$\dot{Q}_{p,i} = UA_{p,i}(T_{p,i,j} - T_{liq/gas}) \quad (8)$$

$$UA_{w,i} = \frac{1}{\sum R_{total w,i,j}} \quad (9)$$

$$\sum R_{total w,i,j} = \sum \frac{1}{h_{i,j}A_{i,j}} + \sum \frac{\ln\left(\frac{D}{\frac{D}{2} + \frac{t_w}{2}}\right)}{2\pi L_{i,j}K_w} \quad (10)$$

$$UA_{p,i,j} = \frac{1}{\sum R_{total p,i,j}} \quad (11)$$

$$\sum R_{total p,i,j} = \sum \frac{1}{h_{i,j}A_{i,j}} + \frac{t_p}{A_{i,j}K_p} \quad (12)$$

where, U is the overall heat transfer coefficient and R refers to heat transfer resistance of the wall or plates. In equations (10) and (12), t_w and t_p are the thickness of the cylinder wall and plates, respectively. Meanwhile, K is the thermal conductivity of the wall or plates.

The heat transfer along the wall length (axial heat transfer in the walls) and convection heat transfer from the wall to the surrounding environment are defined as:

$$\dot{Q}_{axial\ w,i} = \sum K_w A_{w,i} (T_{w,i+1} - T_{w,i}) / L_i \quad (13)$$

$$\dot{Q}_{axial\ p,i,j} = \sum K_p A_{p,i,j} (T_{p,i+1} - T_{p,i}) / L_{i,j} \quad (14)$$

$$\dot{Q}_{sur,i} = UA_i (T_{amb} - T_{w,i}) \quad (15)$$

$$UA_{air} = \frac{1}{\sum R_{t,i}} \quad (16)$$

$$\sum R_{t,i} = \frac{1}{h_{sur}\pi(D + 2t_w)L_i} + \frac{\ln(\frac{D}{2} + t_w) / (\frac{D}{2} + \frac{t_w}{2})}{2\pi L_i K_w} \quad (17)$$

In order to solve these equations, the convective heat transfer coefficient should be known a priori. It can be calculated based on flow regime, defined as Reynolds and Nusselt numbers. The general formulation of Nusselt number is as follows, where the coefficients vary with flow regime and geometry of the container. In this work, the following equations are used for the circular chamber and the chamber with plates [16].

$$Re_i = \frac{\rho_{liq}\ V D_h}{\mu_{liq}} \quad (18)$$

$$D_h = \frac{4A}{P} \quad (19)$$

In equations (18) and (19), ρ and μ are the density and viscosity of air and water, respectively. Also D_h , V, A and P are the hydraulic diameter, velocity, cross-sectional area and perimeter of each ducts or a simple circular cylinder for the base case. The Nusselt number for turbulent flow and laminar flow is then calculated as [10, 16]:

$$Nu = A Re^a Pr^b \mu^c / \mu_0 \quad (20)$$

The value of A, a, b and c coefficients depend on the flow regime. For turbulent flow ($Re > 3000$) these coefficients are 0.026, 0.8 and 0.3 for the liquid region [16] and 0.75, 0.8 and 0.6 for the gas region [17]. Also, these coefficients for gas and liquid flows for the laminar flow regime ($Re < 2000$) are taken to be 0.664, 0.5 and 0.3 [10].

As explained above one of the main parameters to be used to evaluate the performance of a LPGC systems is the compression efficiency. Compression efficiency is defined as the following equation [1, 3]:

$$\eta_{Com} = \frac{E_s}{W_c + W_f + W_{cool}} \quad (21)$$

In this equation, E_s is the maximum energy storage for an isothermal expansion. The first step of this expansion is a constant pressure pumping of the gas from the reservoir to the expansion chamber and the next step is the expansion phase. The stored energy is defined as follows [3]:

$$E_s = P_c V_{final} \ln |P_c / P_{initial}| \quad (22)$$

where V_{final} is the final volume of the compressed gas after cooling to the ambient temperature at the constant pressure, P_c is the gas pressure at the compression ratio and $P_{initial}$ is the initial pressure of the gas.

W_c in equation (21) is the compression work to compress air from initial volume to the final volume for the desired pressure ratio as follows:

$$W_c = \int_{V_{initial}}^{V_{final}} P \, dv \quad (23)$$

W_f in equation (21) is the energy loss due to friction which is calculated by the integral of pressure drop from an initial volume of air to the final volume of air at the end of compression:

$$W_f = \int_{V_{initial}}^{V_{final}} \Delta P \, dv \quad (24)$$

where, ΔP describes the pressure drop for fully developed, steady, incompressible flow as [18]:

$$\Delta P = f \frac{L}{D_h} \frac{\rho v^2}{2} \quad (25)$$

In this equation, f is the friction factor, L is the height of liquid in the cylinder, D_h is the hydraulic diameter of the duct or circular cross-sectional chamber, ρ is the density of the liquid, and v is the velocity of the liquid column. Friction factor for different flow regimes is as [19]:

$$\begin{aligned} f &= \frac{64}{Re} & Re \leq 2000 \\ f &= a Re + b & 2000 \leq Re \leq 3000 \\ \frac{1}{\sqrt{f}} &= -0.869 \ln \left(\frac{\epsilon}{3.7} + \frac{2.523}{Re \sqrt{f}} \right) & Re \geq 3000 \end{aligned} \quad (26)$$

Where, $\frac{\epsilon}{d}$ is the relative roughness and a and b are constants that specify relative roughness [19].

W_{cool} in equation (21) is the consuming work while the compressed air cools to the ambient temperature at the constant pressure in the storage tank [20] and is given by:

$$W_{cool} = (P_f - P_0) \left(V_f - V_0 \frac{P_f}{P_0} \right) \quad (27)$$

V_f and P_f are the final volume and pressure of the gas at the end of compression, respectively. V_0 and P_0 are the initial volume and pressure of gas at the end of compression, respectively.

6. Results and discussion

In the present work, we first compare the results of the modelling against those reported in [10]. In reference [10] the LPGC for the hydrogen storage application was analysed from the heat transfer point of view. By considering air and fluid as real fluids, the thermodynamic parameters of air and water such as enthalpy, internal energy, viscosity, density and thermal conductivity are temperature and pressure dependent - modelled herein as described in Section 5. Figure 2 shows the hydrogen gas temperature predicted in the present work against those reported in reference [10]. As expected a perfect agreement is observed for the gas temperature as a function of liquid-gas interface displacement.

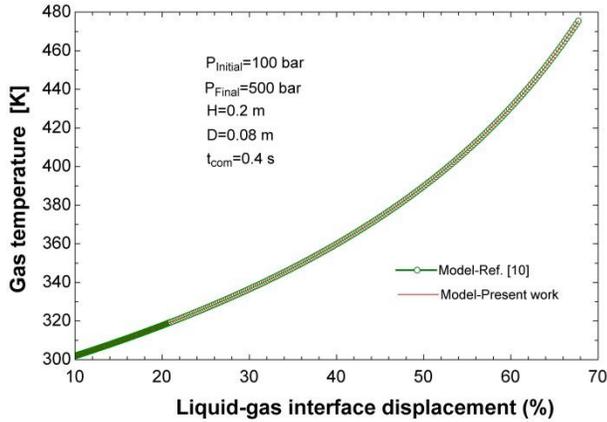


Figure 2. Comparison of the gas temperature as a function of liquid-gas interface displacement predicted by the present model against those reported in reference [10].

In the following, the thermodynamic and heat transfer parameters such as the temperature profile of air, wall and porous medium, friction work, compression work and compression efficiency for the LPGC are discussed.

Figure 3 illustrates the temperature profile of air as a function of the liquid-gas interface displacement for the LPGC system with and without the plates with five and nine plates. In order to compress air from 8 bar to 40 bar, the liquid moves up inside the cylinder to some 73% of the cylinder volume in the base case and to 80% of the cylinder volume in the LPGC with five plates and to 83% of the cylinder volume in the LPGC with nine plates. The air temperature rises from 298.2 to 430 K for the base case and from 298.2 K to 419 K and 298.2 K to 414 K for a cylinder with five and nine plates, respectively. This figure clarifies that using plates the final compressed air temperature in the LPGC with plates is significantly lower than that of the LPGC with no plates. Thus, in this case the introduction of plates may be used to create a near-isothermal compression system.

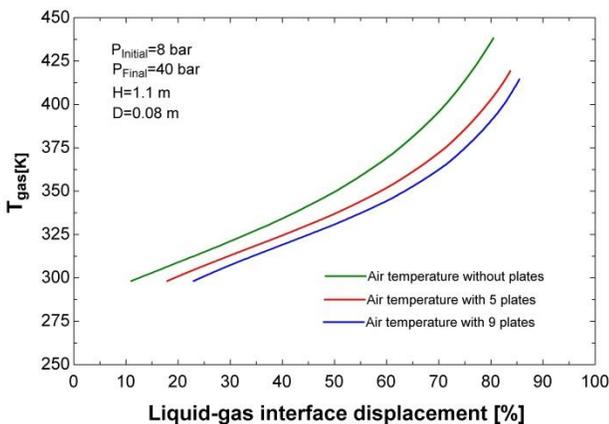


Figure 3. Air temperature as a function of liquid displacement in the LPGC with and without plates.

The variations of cylinder wall temperature as a function of distance from the bottom side of the chamber with and without plates are shown in Figure 4. It is seen that for each case, after the compression process, the wall temperature where it is in contact with the liquid remains almost constant in all cases. It is because

the water enters into the cylinder continually with ambient temperature during the compression phase. While, because of heat transfer between high-temperature air and the cylinder wall, the wall temperature boosts from 298.2 K to 394.2 K for the base case, from 298.2 K to 339 K in a cylinder with five plates and from 298.2 K to 324 K in the case of a cylinder with nine plates. This figure further shows that using plates reduces the wall temperature at the top section of the cylinder wall. It is because of the change in flow regime from turbulent to laminar by inserting plates into the chamber. The Reynolds number depends on the hydraulic diameter which declines with the addition of the plates. As a result, the Nusselt number and convective heat transfer coefficient between the wall and air drops compare to the base case. This results in a lower heat transfer rate between the air and the cylinder wall and consequently lower wall temperature. Figure 3 also shows that increasing the number of plates from 5 to 9 does not have a noticeable effect on the wall temperature.

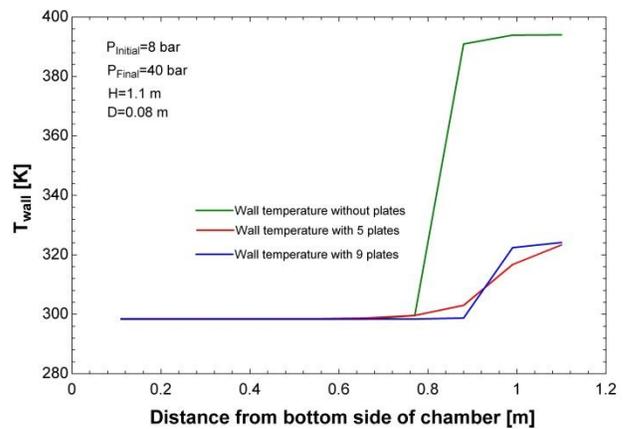


Figure 4. Wall temperature as a function of distance from the bottom side of the chamber.

Figure 5 reveals that the temperature profile of the plates depends on the distance from the bottom side of the chamber for an LPGC with and without plates. The trend is similar to the temperature profile of the cylinder wall (figure 4). In the case of the cylinder with 5 plates, the plates temperature varies from 298.4 K to 334.2 K, and for the case with 9 plates it varies from 298.4 K to 323 K. It is clear that the lowest temperature is a section of the plate in contact with the liquid and the highest value is at the top of the plate in contact with the high-temperature air.

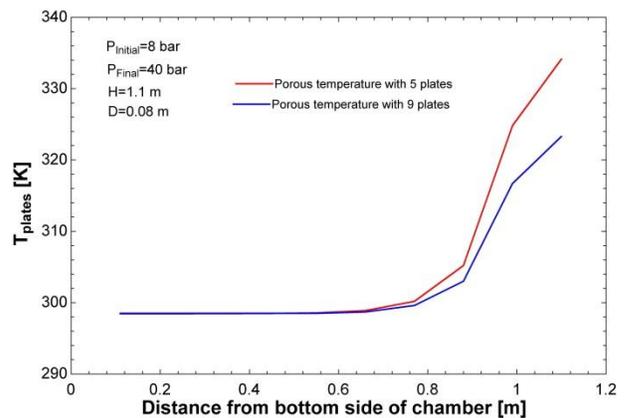


Figure 5. Temperature of the plates as a function of distance from the bottom side of the chamber.

It is expected that increasing the rate of heat transfer from the compressed gas and thus moving toward an isothermal compressor using parallel plates, is achieved by an increase in pressure drop due to friction loss. Figure 6 shows the friction work variation as a function of the compression time. This figure shows that the friction work rises from 0 J to 0.28 J where there are 5 plates in the chamber and from 0 J to 0.5 J for the case with 9 plates. The friction work in the base case is negligible.

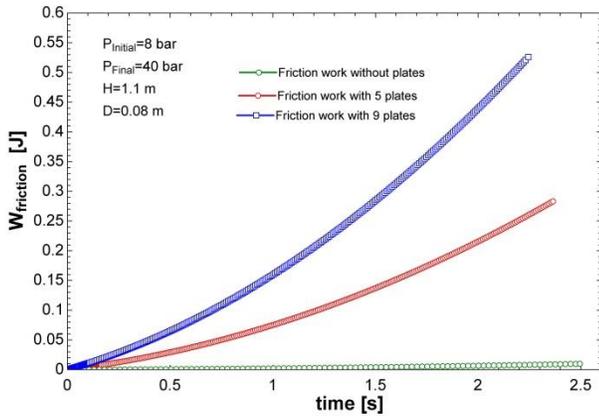


Figure 6. Friction work as a function compression time.

The variation of compression work as a function of compression time is shown in Figure 7. This work is needed to compress air from 8 bar to 40 bar. In all cases, there is an increasing trend in pressure work. In the base case in the compression time of 2.49s, the pressure work increases from 0 W to 5912 W. In case of 5 plates LPGC, it rises from 0 W to 5577 W during a compression time of 2.36 s, and for the LPGC with 9 plates it rises from 0 W to 5275 W with a 2.24s compression time. The value of pressure work at the end of compression declines by inserting plates into the chamber. This is because a portion of cylinder volume is occupied by plates and a lower volume of air is thus compressed compared to the base case. Meanwhile, the time of compression and mass flow rate of the required water to pressurize the air is lower than the base case.

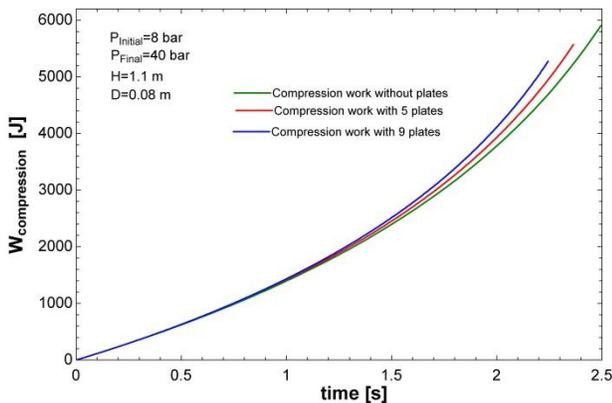


Figure 7. Compression work as a function compression time.

Because all thermodynamic, physical and heat transfer parameters are changed by inserting plates into the chamber, we need a non-dimensional parameter to compare the performance of all case studies. In this regard, compression efficiency can be used as a non-dimensional parameter. Figure 8 illustrates the compression efficiency as a function of the number of plates. It can be seen from this figure that in the base case the compression efficiency is about 87%. After adding 5 plates into the chamber, the efficiency increases up to 88% and with 9 plates the compression efficiency increases to 89%. It can be observed that inserting more plates has a diminishing effect, comparing the efficiency between LPGC with 5 plates and 9 plates, the efficiency only improved by 1%.

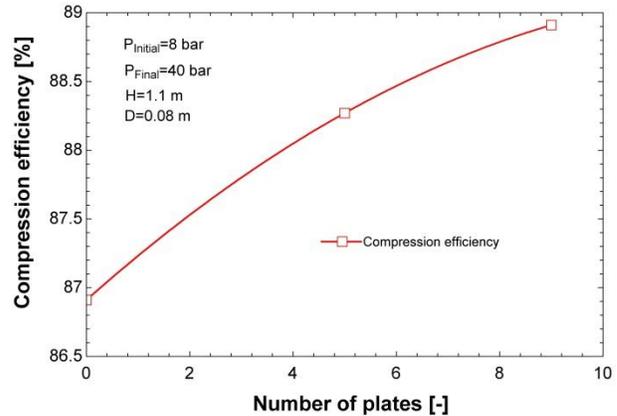


Figure 8. Compression efficiency as a function of the number of plates

Table 2 compares important parameters in the compression process in LPGC with and without plates. Since 10% of the chamber is assumed to be filled initially with water, then adding plates to the chamber will result in a lower amount of mass of gas and water. It is observed that more plates in the chamber lead to lower required cooling work (equation (27)). That is because more plates leads to the lower final temperature of the air at the end of compression so that lower power is needed to cool the air to the ambient temperature. Also, the stored energy and compression work decreases because of the drop-in mass of air by inserting plates, but the friction work increases by adding plates into the cylinder. Finally, the compression efficiency which is the ratio of stored energy and input work improves by inserting plates into the chamber.

Table 2. Important parameters in compression process in LPGC with and without porous media

| Parameters | LPGC | LPGC-5 plates | LPGC-9 plates | Units |
|-----------------------|---------|---------------|---------------|-------|
| M-air | 0.04662 | 0.04338 | 0.0409 | kg |
| M-liquid | 3.483 | 3.302 | 3.133 | kg |
| Input work | 7476 | 6841 | 6423 | J |
| Stored energy | 6497 | 6038 | 5711 | J |
| Required cooling work | 1564 | 1263 | 1148 | J |
| Friction work | 0.009 | 0.28 | 0.52 | J |

| | | | | |
|------------------------|-------|-------|-------|---|
| Pressure work | 5912 | 5577 | 5275 | J |
| Compression efficiency | 86.91 | 88.27 | 88.91 | % |

7. Conclusion

In this work, thermodynamic and heat transfer analysis was conducted to evaluate the efficiency and other pertinent parameters of a LPGC in order to evaluate the use of porous materials as an approach to achieve a near-isothermal compression system. The porous medium is modelled as metal parallel plates inserted in the chamber. The main results of the present modeling are as follows:

- The final temperature of the air at the end of the compression phase with the inclusion of five and nine plates reduces by 15 K and 20 K respectively, compared to the base case.
- By adding plates into the chamber, a significant difference was observed in the final wall temperature compared to the base case, due to change in flow regime and a reduction in heat transfer coefficient between the air and the wall.
- As expected, by adding plates into the chamber of LPGC, the friction work rises with compression time.
- The compression time and the pressure work are reduced in the case of LPGC with porous media because of the reduced air volume of the chamber. This leads to the lower amount of required water to compress the lower amount of air compared to the base case.
- By inserting plates into the cylinder, the compression efficiency improves but there are optimum numbers of plates to have an optimum compression efficiency.

Future work will include sensitivity analysis and optimizing the size of plates and the number of plates to find the optimum configuration of plates in the compression phase.

8. ACKNOWLEDGMENTS

The work is supported by the European Union's INTERREG VA Programmed, managed by the Special EU Program Body (SEUPB), with match funding provided by the Department for the Economy and Department of Jobs, Enterprise and Innovation in Ireland.

Nomenclature

| | |
|----------------|--------------------------------------|
| U (J) | Internal energy |
| Q (J) | Heat transfer |
| W (J) | Work |
| H (J/kg) | Enthalpy |
| M (kg) | Mass |
| R (m^2/KW) | Heat transfer resistance |
| K (W/m K) | Conductivity |
| A (m^2) | Surface area |
| D (m) | Diameter |
| t (m) | thickness |
| h (W/m^2K) | Convective heat transfer coefficient |
| Re (-) | Reynolds |
| D_h (m) | Hydraulic diameter |

| | |
|-----------------|----------------|
| V (m/s) | Velocity |
| Nu (-) | Nusselt number |
| ΔP (Pa) | Pressure drop |

Greek symbols

| | |
|---------------------|------------------------|
| η (-) | Compression efficiency |
| ρ (kg/m^3) | Density |

Subscripts and superscripts

| | |
|------|-----------------------|
| liq | Liquid |
| in | Inlet |
| p | Plate |
| w | Wall |
| i | Displacement position |
| j | Number of ducts |
| amb | Ambient |
| f | Friction |
| cool | Cooling |
| c | Compression |

9. REFERENCES

- [1] Piya, C., Sircar, I., Van de Ven, J. and Olinger, D. 2009. Numerical modeling of liquid piston gas compression. *ASME International Mechanical Engineering Congress and Exposition. American Society of Mechanical Engineers.*
- [2] Heidari, M., Lemofouet, S and Rufer, A. 2014. On The Strategies Towards Isothermal Gas Compression And Expansion. *International Compressor Engineering Conference. Purdue University.*
- [3] Van de Ven, J.D. and Y. Li, P. 2009. Liquid piston gas compression. *Applied Energy*, 86(10): p. 2183-2191. DOI=<https://doi.org/10.1016/j.apenergy.2008.12.001>
- [4] Hadjipaschalis, I., Poullikkas, A. and Efthimiou, V. 2009. Overview of current and future energy storage technologies for electric power applications. *Renewable and Sustainable Energy Reviews*. 13(6-7): p. 1513-1522. DOI=<https://doi.org/10.1016/j.rser.2008.09.028>
- [5] Saadat, M., Y. Li, P. and Simon, T.W. 2012. Optimal trajectories for a liquid piston compressor/expander in a compressed air energy storage system with consideration of heat transfer and friction. *In American Control Conference (ACC), (pp. 1800-1805). IEEE.*
- [6] Zhang, C., Yan, B., Wieberdink, J., Y. Li, P., Van de Ven, D., Loth, E., Simon, T.W. 2014. Thermal analysis of a compressor for application to compressed air energy storage. *Applied thermal engineering*. 73(2): p. 1402-1411. DOI=<https://doi.org/10.1016/j.applthermaleng.2014.08.014>
- [7] Zhang, C., Simon, T.W and Y. Li. P. 2013. Optimization of the axial porosity distribution of porous inserts in a liquid-

- piston gas compressor using a one-dimensional formulation. In *ASME 2013 International Mechanical Engineering Congress and Exposition. American Society of Mechanical Engineers*.
- [8] Zhang, C., Wieberdink, J., W. Simon, T., Y. Li, P., Van de Ven, J.D., Loth, E. 2014. Numerical Analysis of Heat Exchangers Used in a Liquid Piston Compressor Using a One-Dimensional Sub model. In *ASME International Mechanical Engineering Congress and Exposition. American Society of Mechanical Engineers*.
- [9] Zhang, C., Y. Li, P., Van de Ven, J.D., W. Simon, T. 2016. Design analysis of a liquid-piston compression chamber with application to compressed air energy storage. *Applied thermal engineering*, 101: p. 704-709. DOI=<https://doi.org/10.1016/j.applthermaleng.2016.01.082>
- [10] Kermani, N.A. and Rokni, M. 2015. Heat transfer analysis of liquid piston compressor for hydrogen applications. *International journal of hydrogen energy*, 40(35): p. 11522-11529. DOI=<https://doi.org/10.1016/j.ijhydene.2015.01.098>
- [11] Klien, S. and Alvarado, F. 1997. *EES engineering equation solver: user guide. F-chart software, Wisconsin*.
- [12] Su, B.-L., Sanchez, C and Y. Yang, X. 2012. Hierarchically structured porous materials: from nanoscience to catalysis, separation, optics, energy, and life science. *John Wiley & Sons*.
- [13] Yan, B., Wieberdink, J., Shirazi, F., Y. Li, P., W. Simon, T., Van de Ven, JD. 2015. Experimental study of heat transfer enhancement in a liquid piston compressor/expander using porous media inserts. *Applied Energy*, 154: p. 40-50. DOI=<https://doi.org/10.1016/j.apenergy.2015.04.106>
- [14] Sonntag, R.E., Borgnakke, C., Van Wylen, G. 1998. Fundamentals of thermodynamics. Vol. 6. *Wiley New York*.
- [15] Kreith, F., Manglik, R.M and Bohn, M.S. 2012. *Principles of heat transfer, Cengage learning*.
- [16] Cengel, Y.A. and Pérez, H. 2004. *Heat transfer: a practical approach. transferencia de calor*.
- [17] Liu, R. and Zhou, Z. 1984. Heat transfer between gas and cylinder wall of refrigerating reciprocating compressor. *International Compressor Engineering Conference*.
- [18] Schetz, J.A. and Fuhs, A.E. 1999. Fundamentals of fluid mechanics. *John Wiley & Sons*.
- [19] Streeter, V.L., Wylie, E.B. and K.W. 1998. *Bedford, Fluid mechanics. WCB, McGraw-Hill*.
- [20] Zhang, C., Simon, T.W. and Y. Li, P. 2012. Storage power and efficiency analysis based on CFD for air compressors used for compressed air energy storage. In *ASME International Mechanical Engineering Congress and Exposition. American Society of Mechanical Engineers*.