ABSTRACT

RAMAKRISHNAN, KISHORE RANGANATH. Temperature Abatement Using Hollow Spheres in Liquid Piston Compressor for Ocean Compressed Air Energy Storage System. (Under the direction of Dr. Paul I. Ro).

A novel technique to reduce the temperature rise during compression in a liquid piston compressor used in Ocean Compressed Air Energy Storage (OCAES) system has been studied in this work. It involves floating of hollow spheres on the liquid column to achieve lower temperature of compressed air.

Initial 2D simulations were run using ANSYS Fluent, where the layer of spheres were modeled as a plate of thickness equal to the diameter of spheres. Then, experiments were run using high conducting hollow spheres made of Silicon Carbide (SiC) and outer diameter of 6.5 mm. Adding these spheres to the system increases the surface area available for heat transfer from hot high pressure air. Also, this extra layer of spheres promotes additional convective heat transfers between air and spheres and between spheres and water.

Further, material of spheres, diameter of spheres, and compression stroke time were identified as the key parameters which will affect temperature abatement using hollow spheres. Hollow spheres made of High Density Polyethylene (HDPE) and outer diameter of 6.5 mm, and Polypropylene (PP) and outer diameter of 9 mm were tested in the experimental setup to see the effect of size and material of spheres on temperature abatement. Compression stroke time was varied by varying pressure of air inlet to double acting pneumatic cylinders used to power the liquid column.

An analytical model was developed by considering convection between air and spheres, and spheres and water, and conduction in spheres to assess the excess heat transfer facilitated by the addition of hollow spheres. These three heat transfer mechanisms were taken to be in
series with each other to find overall heat transfer coefficient. Also, an empirical model was
developed based on 2D simulations to see the effect of diameter of spheres on temperature
abatement. A cubic not-a-knot spline was used to curve fit the data obtained from simulations
and later compare it to the experimental results for particular diameter of spheres.

Using ANSYS CFX 3D simulations were run to validate the accuracy of modelling the
layer of spheres as a plate of thickness equal to diameter of spheres. This model was based on
Conjugate Heat Transfer (CHT) method to account for the heat transfer at the air-solid-water
interface. It was observed that 2D simulations gave results similar to 3D simulations in
approximately 30 times lesser computational time, hence saving the cost of computation.
Temperature Abatement in Liquid Piston Compressor for Ocean Compressed Air Energy Storage System

by
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To my family and friends.
BIOGRAPHY

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1. Introduction

In this ever changing world everyday a new technology is invented, which is more efficient than its previous iteration in terms of time saving and energy consumption. However, at the same time we are seeing new digitalized versions of old technologies coming into our lives. Thus, an average human’s consumption of electricity for day to day activities has increased gradually over the years. Total annual energy production in the United States of America in 2015 is ~2.77 times the energy produced in 1949 [1]. The net energy available in the United States in 2015 was about 111.82 quadrillion BTU and the net consumption in that year was about 110.25 quadrillion BTU [2]. A major portion of the energy production is from conventional fossil fuel sources, which are responsible for a high amount of Carbon dioxide emission. In order to reduce the carbon footprint, many are taking the renewable energy path. There are many evolved technologies based on wind, solar, geothermal, and biomass, and a few pilot and prototypes to generate electricity from sources like tides and sewage. One of the main issues hurting the renewable energy industry is the lack of certainty about the amount of energy produced in systems based on wind, solar or tide. There is an inherent fluctuation in the availability of the required input for such systems. Also, if the renewable source is directly connected to the grid to supply to various end users, there are periods when the supply and demand requirements do not match.

In order to counter the fluctuations and avoid the loss of energy, researchers have come up with various Electrical Energy Storage (EES) devices which could be coupled with the renewable energy sources. Chen et al. have discussed in detail about the progress made in
several different EES systems like Pumped Hydroelectric Storage (PHS), Compressed Air Energy Storage (CAES), fuel cells, batteries, etc. [3]. Rogers et al.’s review paper on the economic and thermodynamic aspects of the CAES systems, they have compared the CAES system’s feasibility with respect to eight other technologies which are already in use or in pilot stage [4]. The authors of this paper have stated that even though the CAES systems are one of the best available energy storage option at a utility scale, its low efficiency and the problem with storing and delivering of air at constant pressure have been stated as one of the major reasons for not being adopted more in practical systems. At present, there are two CAES based systems running. The first and still functional CAES system is in Huntorf, Germany with a capacity of 290 MW and later increased to 321 MW in 2006 [5,6]. After the success of the Huntorf plant, a newer CAES plant was built at McIntosh, Alabama in 1991 with a capacity of 110 MW [7]. In both these plants, the compressed air is stored in salt caverns and they are used during the time of full load operation of the turbines. Over the past few years researchers have figured out that studying the thermodynamics of a CAES system is of utmost importance if the efficiency has to be improved.

1.1 CAES systems

A conventional CAES plant consists of the following parts:

1. Compressor train
2. Motor-generator unit
3. Gas turbine
4. Underground compressed air storage
Figure 1: Generic schematic of CAES system.

Figure 1 shows a generic CAES system’s schematic representation. The compressor train (a set of compressors) run by the motors by taking power from the grid during off peak hours (supply is more than demand) to compress the air and store it in the underground caverns. When the peak load period (demand is more than supply) starts, the compressed air from the cavern is used to combust the fuels and the hot combustion gases are expanded over a gas turbine to provide the peak load electricity [8].

1.1.1 Types of CAES systems

a) Diabatic CAES (D-CAES)

The two current functional plants at Huntorf and McIntosh are D-CAES systems. These function in a similar way to a conventional gas turbine plant. However, the air compression process is decoupled from the gas turbine process. By doing this, the output of the turbine is
increased multifold. In a conventional gas turbine plant, energy from the turbine output is itself used to compress the air. In a D-CAES system the air compression is taken care by the off peak hour electricity production, thus saving the turbine output energy [9].

b) Adiabatic CAES (A-CAES)

Adiabatic CAES design is still in a conceptual stage. In this method, the heat generated due to rise in temperature of the compressed air is stored and used in the expansion process as a re-heat mechanism, thus eliminating the use of fossil fuels to heat the compressed air during the expansion process [10].

c) Isothermal CAES (I-CAES)

The above two methods of CAES systems use multi stage compression process with intercoolers between each stage to make sure that the compressed air does not go beyond a threshold temperature, making it practically impossible to store. In I-CAES, the compression is attempted in a single stage. For this, the heat transfer of the system has to be at its maximum to dissipate all the heat generated in the compression process. The work done by the compressor can be found from the pressure-volume (P-V) diagram. The closer the P-V graph of the compressor is to the isothermal P-V line, the lesser work it takes to compress. Hence, ideally the I-CAES system will be the most thermodynamically efficient CAES system [11].

1.2 Ocean Compressed Air Energy Storage (OCAES) system

In a conventional CAES, compressed air storage vessel is placed above ground or in salt cavern. When air is being removed from this vessel, the change in volume of air changes the
pressure of the remaining air in the vessel. Since there is no other force acting on the system to maintain the pressure constant, there is still a variation in the electricity produced by it.

Hence, one of the important issue to be tackled in a CAES plant is to control the change of pressure of stored air when it is being released to be expanded over the turbine. One of the new techniques that has been discussed by Park et al. [12] is the OCAES system. Figure 2 shows a schematic representation of the OCAES system proposed by Park et al. [13]. The high pressure air storage vessel is placed on the ocean bed and eliminating the danger of having an above ground or salt cavern storage vessel. This provides an added advantage of using the hydrostatic pressure of the water above the vessel to keep the air at a constant pressure even when the air is being released to be expanded over a turbine. Thus, overcomes the shortcomings of a conventional CAES system with regard to compressed air storage and release.
1.3 Isothermal compression/expansion

Apart from storage, the other major problem with the implementation of CAES system is the high temperature of compressed air. When the compressed air is stored in a vessel, it tries to reach thermal equilibrium with its surroundings. Thus, the thermal energy is wasted. Hence, much effort has been put to achieve isothermal compression/expansion process, as it is the key to achieve the highly efficient I-CAES design. Various researchers have come up with novel techniques to increase the heat transfer of the compressor for a CAES system.
1.3.1 Conventional reciprocating compressor

In the early years, researchers studied the average measured heat transfer rates and later the measured instantaneous heat transfer from the experiments and came up with their analytical correlations to understand the heat transfer phenomena in combustion chambers of internal combustion engines. Nusselt [14], Eichelberg [15], and Pflaum [16] were the first few to get a relation between the instantaneous heat transfer coefficient in terms of experimental parameters and measured values. Equation 1 shows the generic relation that they derived.

\[ h = f \left[ V_p, P_{g\text{ins}}, T_{g\text{ins}} \right] \]  

where \( h \) is the instantaneous heat transfer coefficient, \( V_p \) is the mean piston speed, \( P_{g\text{ins}} \) is the instantaneous gas pressure in the cylinder and \( T_{g\text{ins}} \) is the instantaneous gas temperature in the cylinder. The relations developed by the above mentioned three authors were debunked due to dimensional inconsistencies and the fact that they were wholly empirical in nature and hence could not be reliably used for all applications. Building on the work of Eichelberg, Elser built a modified relation and used better instrumentation in the experimental studies [17]. Although his relation reproduced the two stroke engine measurements quite well, it failed for the four stroke engine. Oguri [18] used the relation developed by Elser and made it more elaborate, however he had not checked for its accuracy for a four stroke engine, and subsequently failed to make a model which can be used for all purposes. All the above observations were made by Annand [19] in his work on the heat transfer study in cylinders of internal combustion engines. Annand [19] also notes that there is difference in phase between the temperature difference
between temperature difference and heat flux i.e. both these quantities are supposed to reach their zeros at the same point in time, however the experimental observation from Esler [17] shows that it does not happen.

Taking into consideration all the shortcomings of the various models suggested till 1963 and the fact that there exists a phase difference, Annand [19] came up with a model which would distinguish between convective and radiative heat transfers for any geometry of the compression chamber. He calls his relation the power law, which is given by Equation 2.

\[ Nu = a \times Re^b \]  

(2)

where \( Nu \) is the Nusselt number, \( a \) is the multiplying constant to account for convective heat transfer, \( Re \) is the Reynolds number, and \( b \) is an index relating \( Nu \) to \( Re \). All the correlations discussed above take into account the point at which the average bulk temperature of the gases surpass the temperature of the wall to account for the heat transfer across the wall. However, due to the complex unsteady physics within the chamber, the gases might form a boundary layer along the wall in which the temperature of the gas might be significantly different from the bulk temperature. Adair et al. [20] studied this anomaly and reported a relation similar to the power law provided by Annand, with an additional Prandtl number term multiplied to the existing Nusselt number correlation as in Equation 2.

Along with these studies carried out to understand the heat transfer within the chamber, researchers have come up with a few novel technologies which might be capable of a near-isothermal compression/expansion. Some of those ideas which have been worked on are liquid
piston, dry piston, and using an inter-fin heat transfer model [21]. Among these, liquid piston is seen as a very promising option to move towards near-isothermal compression/expansion. Companies like LightSail and venture capitalist funded startups are working on an unconventional idea of using a liquid piston compressor in place of the conventional compressors for CAES application.

1.3.2 Liquid piston compressor

A liquid piston compressor utilizes a column of liquid to compress the gas, instead of a solid piston. The main advantage of using a liquid is the freedom of choosing the geometry of the chamber to maximize the surface area to volume ratio. Thus, making sure that we are maximizing the heat transfer through the walls. Also, using a liquid column replaces the friction between the seal and the wall with the viscous forces between the liquid and the wall, thus curbing the excess loss of energy. One of the initial efforts towards designing and understanding the heat transfer in a liquid piston was carried out by Van de Ven et al. [22]. The fluid-fluid interface between water and air acts an extra medium of convective heat transfer. In addition to these, the compression chamber can be split in multiple chambers of smaller diameter. However, there is a need for the number of smaller cylinders, their diameter and length have to be optimized to obtain a proper tradeoff between final temperature, pressure, and viscous losses. Van de Ven et al.’s work concluded that the compression efficiency of 83% for a liquid piston is a substantial improvement compared to a 70% of the conventional piston of the same size [22].
1.3.2.1 Techniques to improve heat transfer in a liquid piston

1.3.2.1.1 Porous inserts and interrupted plates

Zhang et al. came up with the idea of using either porous inserts (open cell metal foam structure) or interrupted plate inserts in the liquid piston chamber [23]. The whole point of using porous inserts to improve the heat transfer in the system can be done only in case of a liquid piston because both the compressing medium and the gas are fluids which can easily pass through the porous insert, thus increasing surface area for heat transfer. They have studied the effect of using these methods both numerically and experimentally. In the foam part, the authors have used foams of 40 pores per inch and 10 pores per inch models to study the effect of pressure drop with the structure of foam insert. For the interrupted plates, the effect of geometric parameters such as length and thickness of the plates, and the gap between successive plates on the heat transfer enhancement have been studied.

1.3.2.1.2 Spray cooling

Qin et al. have made use of the high specific heat property of water to reduce the temperature of the hot compressed air [24]. They have done a numerically study about this technique, where water droplets will be injected into the chamber at a particular instant during the compression process. As the liquid column compresses air in the chamber, the temperature of air increases. By spraying the water droplets into the core of the chamber, the heat from the bulk air is taken by the water droplets. This method
can be used successfully only in a liquid piston compressor and not a conventional compressor, where the piston is a solid which can get corroded by the use of water droplets. The authors have studied the effect of diameter of the droplet and ratio of mass of water sprayed to the mass of dry inlet air (mass loading) on the improvement of heat transfer in the system. They have reported an increase of the compression efficiency to 95% with a mass loading factor of 1 and 98% with a mass loading factor of 5, while the adiabatic efficiency of compression is 71% [24]. Depending on the size of the chamber, time of interaction between water and air might be very small for the heat transfer to be effective. Thus, making it necessary to increase the amount of water being sprayed into the system.

1.3.2.1.3 External gas impingement

Hari et al. [25, 26] studied another new technique for improving the heat transfer in the chamber by injecting high pressure and low pressure air in the chamber at a particular instant in the compression process. This excess air added to the chamber will reduce the bulk temperature of the air in the chamber by absorbing it, and also depending on the angle at which it is injected, it improves the mixing in the chamber and thus increases the heat transfer [24]. They have tried to analyze this problem analytically experimentally. In the experiments various factors like the point and angle of location, pressure of the injected air, pressure of the chamber at which the external air in injected,
and the duration of air injection were considered in seeing the extent of the effectiveness on temperature abatement.

All the three above mentioned methods discussed for temperature abatement either require more work during each compression process (porous insert and air impingement) or will accumulate substantial amount of water over the course of time and will reduce the effective chamber length (spray cooling). The porous medium causes the compressed air passing through it to have pressure drop while moving between the pores, thus making the compressor put in more work to compress the air to the desired final pressure. Similarly, with the air impingement method the external high pressure air added during compression increases the mass of air in the chamber and increases the input work for the process.

This present work focuses on a novel technique for temperature abatement in liquid piston compressor, without increasing the work input required for compression. As a part of this master’s thesis the effect of floating hollow spheres made of different materials and sizes on the liquid column on temperature abatement. This method was studied analytically, numerically and experimentally.

1.4 Hollow spheres

The use of hollow spheres gives rise to a new medium and interface for heat transfer. Instead of just having convective heat transfer between water and air at the interface and between air and the walls of the chamber, the spheres floating on water surface will absorb heat from the air, conduct it with in itself, and then convect it on to water. In addition to this, this also
increases the surface area available for heat transfer. As mentioned before, the main advantage of this method is that it requires the same amount of work to compress air as the regular method without hollow spheres. This can be verified by the fact that the pressure curve for both the cases follow the same trajectory and take the same time to reach the desired pressure ratio. This work aims at identifying the physical parameters of the hollow spheres which might affect the heat transfer in the system.
2. Analytical Model

This analytical model has been developed with the sole aim of trying to see the factors that will affect the heat transfer component added to the system by introducing the hollow spheres. In the following sections, the heat transfer calculated from the analytical model has been compared to the value obtained from the 3D simulation of the system.

The air in the compression chamber is assumed to be an ideal gas in nature because the process here takes place at a very low pressure. Starting with Equation 3, the polytropic equation of state [27]:

$$Tp^{\frac{1-n}{n}} = constant$$

(3)

where \( T \) is the absolute temperature of air, \( p \) is the absolute pressure in Pascal, and \( n \) is the polytropic index of the compression process.

Differentiating the polytropic equation involving temperature and volume with respect to time, we get

$$\frac{dT}{dt} = (n - 1) \frac{T}{V} \frac{dV}{dt}$$

(4)

where \( T \) is the absolute temperature of air, \( V \) is the volume of the compression chamber, and \( n \) is the polytropic index of the compression process. Substituting for the rate of change of volume in the above Equation 4 as the product of the cross sectional area of the compression chamber
and the speed of the water front compressing the air, we get the following relation shown in
Equation 5 for temperature of air at any time instant after integration.

\[ T_a = T_o t^{n-1} \]  

(5)

where \( T_a \) is the temperature of air at any time \( t \), and \( T_o \) is the initial temperature of air.

As the temperature of air varies, the properties of air like viscosity, density, specific heat, and
conductivity also vary. The variation of viscosity is accounted for by using Sutherland’s
formula:

\[ \mu = \mu_o \left( \frac{T}{T_o} \right)^{1.5} \left( \frac{T_o + 110.4}{T + 110.4} \right) \]  

(6)

where \( \mu_o \) is the dynamic viscosity of air at absolute temperature \( T_o \) (273K) in kg/m-s, \( T \) is the
temperature at which the dynamic viscosity has to be found.

The variation in density with change of temperature, which in turn is a function of time is found
by using Equation 3 and Equation 5, as shown below.

\[ \rho = \rho_o t^{n(1-n)} \]  

(7)

The variation of specific heat and conductivity of air with temperature were obtained from
literature [28].

\[ c = 1002.5 + 275 \times 10^{-5}(T - 200)^2 \]  

(8)
\[ k = 0.02624 \left( \frac{T}{300} \right)^{0.8646} \tag{9} \]

where \( c \) is the specific heat of air at absolute temperature \( T \), and \( k \) is the conductivity of air at absolute temperature \( T \).

Since the thermal conductivity of air is very low, the spheres can be assumed to be at a constant temperature. In this model, three thermal resistance have been taken into account.

1) Convection between air and solid.
2) Conduction in solid.
3) Convection between solid and water.

All the above obtained parameters were substituted into the Nusselt number and Rayleigh number equations for free convection [29].

\[ Ra_D = \frac{g \rho^2 c}{T k \mu} (T - T_s)D^3 \tag{10} \]

\[
\bar{N}u_D = 2 + \frac{0.589 Ra_D^{0.25}}{1 + \left( \frac{0.469}{Pr} \right)^{9/16}} \left[ \frac{h D}{k} \right]^{4/9} \tag{11}
\]

where \( Ra_D \) is the Rayleigh number for the sphere of diameter \( D \), \( \bar{N}u_D \) is the average Nusselt number for the free convection over a sphere of diameter \( D \), and \( Pr \) is the Prandtl number of air.
From the above equations, the heat transfer coefficient between air and sphere, and sphere and water can be found by substituting the properties of the respective fluids in Equation 10 and Equation 11.

\[
q'' = \frac{(T_a - T_w)}{\left(\frac{1}{h_{a-s}} + \frac{D_{eff}}{k_s} + \frac{1}{h_{s-w}}\right)}
\]  

(12)

where \( q'' \) is the heat transfer per unit area, \( T_a \) is the absolute temperature of air, \( T_w \) is the absolute temperature of water, \( h_{a-s} \) is the heat transfer coefficient between air and solid, \( h_{s-w} \) is the heat transfer coefficient between solid and water, \( D_{eff} \) is the effective diameter of the sphere taking into account the number of spheres covering the water surface, and \( k_s \) is the thermal conductivity of the material of the sphere.

The excess heat transfer in the system due to the addition of hollow spheres (Equation 12) is derived as a function of air and water temperature, and overall thermal resistance in the system. The denominator of Equation 12 is the reciprocal of effective overall heat transfer coefficient at the air-solid-water interface.

It is observed that the temperature of spheres remain almost constant, and hence the conduction component can be neglected. However, the size of the sphere and the increase in surface area that it provides will be taken into account while calculating the heat transfer coefficients between air and spheres, and spheres and water. In later section, the results from analytical model is compared with heat flux found by simulation.
3. Simulation

In this work both 2 dimensional and 3 dimensional models were simulated using commercial Computational Fluid Dynamics (CFD) packages. The 2D model was an axis-symmetric model simulated using ANSYS Fluent package, and the 3D model was simulated using ANSYS CFX package.

There are three major methods which commercial CFD packages use to solve the underlying Navier Stokes system of equations, viz.

1) Finite Difference Method (FDM).

In Finite Difference Method (FDM), the domain of the flow is divided into a number of grid points. The derivatives appearing in the Navier Stokes equation are numerically discretized using Taylor’s series expansion as:

\[
\frac{dB}{dx_{x=i}} = \frac{B(i+1) - B(i)}{\Delta x} + O(\Delta x)
\]  

(13)

B is any function whose value depends on spatial coordinates as well as time, such as velocity, pressure or density and O (\(\Delta x\)) is the error produced due to approximating a derivative by its finite difference. This is termed as first order approximation since error is proportional to (\(\Delta x\)). Higher order approximations result in errors (\(\Delta x\)) \(m\), where \(m\) is the order of the method and hence are more accurate. As the number of grid point increases, grid spacing \(\Delta x\), reduces, leading to a more accurate estimation of the derivative. By replacing all the derivatives in the equations with their corresponding
finite differences, a set of algebraic equations, relating the value of the unknown field variable at a node to its neighboring node, result. The solution of the resulting equations give the required flow quantities in the domain [30].

2) Finite Element Method (FEM).

Finite Element Method (FEM) is a numerical technique for solving (PDE). Its essential characteristic is that the domain, is subdivided into cells, called elements. The elements can be rectilinear or curved. In addition, the FEM does not look for the solution of the PDE itself, but looks for a solution of an integral form of the PDE. The most general integral form is obtained from a weighted residual formulation. The important advantage of FEM, not shared by the FDM, is FDM needs a structured grid, whereas with curved cells and unstructured grids, FEM can handle complex geometries with ease. With respect to accuracy, the FEM is superior to the FVM, where higher order formulations of the PDE are quite complicated. [31]

3) Finite Volume Method (FVM).

In the Finite Volume Method, the domain is divided into smaller volumes called control volumes. The governing partial differential equations are recast in a conservative form, and then solved over discrete control volumes. The field variables viz. density, pressure and velocity are calculated at the center of the control volumes. This discretization guarantees the conservation of fluxes through a particular control volume. The finite volume equation yields governing equations in the form,
\[
\frac{\partial}{\partial t} \iiint Q \, dV + \iint F \, dA = 0
\]  

(14)

Q is the vector of conserved variables, with mass, momentum and energy as its components. F is the vector of fluxes, with mass, momentum and energy fluxes as components. V and A are the volume and area of the fluid control volume respectively [30].

Even though FEM is better compared to FDM and FVM, FVM is the most used method in CFD since it is computationally superior and easier to implement than FEM, and has better geometrical flexibility than FDM. Both Fluent and CFX are FVM based solvers.

3.1 2D Simulation setup

As stated earlier, an axis-symmetric model was developed and studied using ANSYS Fluent package. ANSYS Fluent is a commercially available CFD software by ANSYS Inc. which is suitable for the simulation of low subsonic, laminar or turbulent, viscous or in viscid, non-reacting flows.

Since the geometry is very simple it was designed using the ANSYS design modular as shown in Figure 3 and Figure 4 for the case without and with spheres respectively. The computational domain for the case without spheres was split into two domains viz. water and air, and for the case with spheres it was split into three domains viz. water, solid (spheres) and air. In this setup, the layer of spheres has been approximated as a plate of thickness equal to the diameter of the sphere.
Figure 3: 2D geometry without spheres.
Figure 5 and Figure 6 show the mesh for the geometry of the two cases which were generated in the ANSYS Workbench meshing tool, and Table 1 gives the mesh details for both the cases in 2D simulation. It is observed that the skewness for both the cases is close to 0 and orthogonal quality close to 1. Both these are considered the main measures to understand the quality of mesh in the simulation. If the skewness which can range from 0 to 1 is between 0 and 0.25, the mesh is considered to be excellent and similarly if the orthogonal quality which also ranges from 0 to 1 is close 1, the mesh is considered excellent [32].
Figure 5: Mesh for 2D setup without spheres.

Table 1: 2D simulation mesh details.

<table>
<thead>
<tr>
<th></th>
<th>Number of mesh elements</th>
<th>Type of mesh elements</th>
<th>Skewness</th>
<th>Orthogonal Quality</th>
</tr>
</thead>
<tbody>
<tr>
<td>Without spheres</td>
<td>8336</td>
<td>Quadrilateral</td>
<td>1.58E-3</td>
<td>0.99</td>
</tr>
<tr>
<td>With spheres</td>
<td>13593</td>
<td>Quadrilateral</td>
<td>2.92E-3</td>
<td>0.99</td>
</tr>
</tbody>
</table>
To accommodate for the flow of water into the compression chamber, a pressure inlet was given. And the pressure difference was created by moving the interface (water-air or water-sphere-air) toward the top end of the compression chamber. Fluent supports the interfacing of a user defined function (UDF) written using C language. One such UDF was written (attached in Appendix) to make the interface move at a given speed and compress air. A moving and re-meshing technique called layering has been used in the piston movement. This method can be given inputs for it to collapse two cells into one cell based on the height ratio of the cell which

Figure 6: Mesh for 2D setup with spheres.
is breaking apart and the next cell that the moving front will encounter. Figure 7 demonstrates this re-meshing technique at some arbitrary time \( t \) s and \( t + \Delta t \) s.

Figure 7: 2D mesh at (a) \( t \) s; (b) \( t + \Delta t \) s.

The thickness of the chamber walls and material properties were given to account for the heat transfer out of the chamber wall. Also, the material properties of the spheres, their initial temperature and initial temperature of the liquid (water) column was specified to be used for the energy equation.

Semi Implicit Method for Pressure Linked Equations Consistent (SIMPLEC), a pressure based solver, was chosen since it has a good convergence rate and required accuracy [33]. First order upwind scheme was chosen to solve the Reynolds Averaged Navier Stokes (RANS) equations over the higher order schemes in order to save the computation time.
3.2 3D Simulation setup

Similar to the 2D setup, even the 3D setup was split into multiple zones. The number of zones depended on the case whether it was without or with spheres. The major difference here in terms of modelling is in the case with spheres. Instead of using a plate approximation, actual spheres have been introduced into the simulation domain. Figure 8 and Figure 9 show the 3D simulation model for the case without and with spheres.

*Figure 8: 3D geometry without spheres.*
Figure 10 shows the top view of the layer of spheres floating on the surface of water. Meshing was done using the ANSYS Workbench meshing tool. Figure 11 and Figure 12 show the mesh for the cases without and with spheres, and Table 2 gives the details about the mesh. Even in this case, the mesh is observed to be of good quality when the skewness and orthogonal quality are checked [32].

Figure 9: 3D geometry with spheres.
Figure 10: Top view of layer of spheres in 3D simulation setup.

<table>
<thead>
<tr>
<th></th>
<th>Number of mesh elements</th>
<th>Type of mesh elements</th>
<th>Skewness</th>
<th>Orthogonal Quality</th>
</tr>
</thead>
<tbody>
<tr>
<td>Without spheres</td>
<td>54122</td>
<td>Tetrahedron</td>
<td>0.21</td>
<td>0.87</td>
</tr>
<tr>
<td>With spheres</td>
<td>398754</td>
<td>Tetrahedron</td>
<td>0.24</td>
<td>0.85</td>
</tr>
</tbody>
</table>
Figure 11: 3D mesh without spheres.
In this setup the mesh movement was achieved by keeping the layer of spheres at a fixed position and moving the chamber downward at the same speed at which the water front would move in the actual experiment; the physics was still consistent. A total energy model was used for the fluid domains and was combined with the conservative flux model for the spheres (solid domain). This is called the Conjugate Heat Transfer (CHT) method, where the whole Navier Stokes system of equations are solved for fluid regions, and just the energy equation for the
solid region without any flow in it [34]. One of the objectives for trying 3D simulation was to see how the spherical surface and the gap between the spheres affect the heat transfer.

Both the 2D and 3D simulations were run using Shear Stress Transport (SST) and k-ε models. Both these are two equation models. In SST, the solver uses the k-ω model to solve the flow near the wall, and k-ε model for the bulk of the fluid. Previous works have shown that these models do not differ much for certain cases [35].
4. Experiment

The experimental setup used in this work is a slightly modified version of the setup made use by Park et al. [13] and Hari [25]. Figure 13 below shows the experimental setup, with various parts numbered on the image. Table 3 gives the description of these parts.

Figure 13: Experimental setup.
Table 3: Description of components in the experimental setup.

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Power source for running the programmable logic controller (PLC) to control the valves</td>
</tr>
<tr>
<td>2</td>
<td>Programmable logic controller (PLC)</td>
</tr>
<tr>
<td>3</td>
<td>Data acquisition (DAQ) system</td>
</tr>
<tr>
<td>4</td>
<td>Outer polycarbonate chamber for water jacket</td>
</tr>
<tr>
<td>5</td>
<td>Double acting pneumatic cylinder</td>
</tr>
<tr>
<td>6</td>
<td>Polycarbonate compression chamber</td>
</tr>
</tbody>
</table>

In this experiment, one pressure sensor placed before the outlet valve and one thermocouple placed near the top of the compression chamber are used to measure pressure and temperature respectively. The DAQ system is an interface between the signal source and the computer. Here, the signal sources are the data from pressure and temperature sensors. Both these analog signals (voltage) are converted into digital format to log into a file in the computer using the DAQ system.

Figure 14: Location of pressure sensor.
Figure 14 shows the placement of the pressure sensor and Figure 15 shows the placement of the thermocouple. Figure 16 shows the K-type thermocouple used in this experiment, whose tip is at a distance of 30 mm from the top of the compression chamber.

*Figure 15: Location of thermocouple.*

*Figure 16: K-type thermocouple.*
The double acting pneumatic cylinders are used to push the water in and out of the compression chamber. The two valves at the inlet and outlet, on the either side of the pressure sensor as shown in Figure 14, and the piston-cylinder movement are controlled by a LabVIEW program which is interfaced with the PLC using the DAQ system.

Experiments were run with and without spheres. The spheres used are hollow in structure made of materials like Silicon Carbide (SiC), High Density Polyethylene (HDPE), and Polypropylene (PP). Thermal conductivity of the materials and the outer diameter of the spheres are mentioned in Table 4 below. Figure 17 shows the various spheres.

*Table 4: Material properties and diameters of hollow spheres.*

<table>
<thead>
<tr>
<th>Material</th>
<th>Thermal Conductivity (W/mK)</th>
<th>Diameter (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>SiC</td>
<td>120</td>
<td>6.5</td>
</tr>
<tr>
<td>HDPE</td>
<td>0.5</td>
<td>6.5</td>
</tr>
<tr>
<td>PP</td>
<td>0.22</td>
<td>9</td>
</tr>
</tbody>
</table>
Figure 17: Hollow spheres made of (a) SiC; (b) PP; (c) HDPE.

The effect of using spheres with variation in the speed of the compression process was also studied using this experimental setup. The compression stroke time was varied by changing the pressure input to the double acting pneumatic cylinder. The input pressure was varied from 206.84 kPa to 551.58 kPa, by adjusting the pressure control valve.

The results from the experiment have been presented and compared to the simulation in the next section.
5. Results

5.1 Effect of using hollow spheres

Before running experiments, the simulation were run to see if floating hollow spheres on the surface of water column compressing air would bring down the temperature of air. As shown in chapter 3, the layer of sphere was approximated as a plate of thickness equal to the diameter of the spheres.

When the hollow SiC spheres were used during compression, both the experimental and 2D computational methods show 13K reduction in the temperature of the compressed air, for a 2:1 compression ratio this can be observed in Figure 18 (a). It can be seen from Figure 18 (b) that the pressure profiles are the same for both with and without spheres case. This shows that there is no extra work needed to compress air with spheres on the surface of water.
Figure 18: Comparison of experimental and computational results for 2:1 compression ratio (a) Temperature v/s Time; (b) Pressure v/s Time.
5.2 Effect of material of the spheres

The experiments were carried out using hollow spheres of the same size as that of the SiC spheres, but made of HDPE. It was found that the temperature abatement by both these materials are similar.

Similarly, when PP hollow spheres of a higher outer diameter was used, the temperature abatement was not as effective as the smaller SiC or HDPE hollow spheres. Figure 19 shows the comparison of the temperature reached at a particular pressure for the various spheres used.

![Figure 19: Comparison of experimental results without and with spheres of different materials and sizes.](image)

When spheres of bigger diameter are used, the number of spheres required to cover the surface of the water column reduces. Also, there are bigger and more gaps between the spheres in that
layer. Thus, the effective surface area available for heat transfer is lesser than that created by a layer of smaller spheres. This can be attributed as the main reason for reduction in the temperature abatement process.

In order to understand the effect of the diameter of the sphere on temperature abatement, 2D simulations were run by varying the diameter from 4.5mm to 13.5mm in steps of 1mm and the material was assigned to be SiC. An empirical model was developed to quantify this effect.

5.2.1 Empirical model

A ‘Not-a-knot’ cubic spline was used to fit the data. The maximum temperature $(T_{\text{max}})$ was taken from the simulation at the same position where the thermocouple is placed in the experimental setup. Polytropic index $(n)$ is calculated using Equation 15 which was obtained by solving the polytropic equation of state for the polytropic index (Equation 3).

$$n = \frac{\ln(Pr)}{\ln(Pr) - \ln \left( \frac{T_{\text{max}}}{T_{\text{atm}}} \right)}$$

(15)

where $Pr$ is the pressure ratio ($Pr = 2$), and $T_{\text{atm}}$ is the atmospheric temperature or the inlet temperature of air ($T_{\text{atm}} = 296K$).

Table 5 below shows that the polytropic index increases as the diameter of the sphere is increased from 4.5mm to 11.5mm and it stabilizes beyond 11.5mm. Figure 20 shows the not-a-knot cubic spline fit for the data obtained from the simulations. It is observed that when hollow spheres of lower diameter are used the polytropic index is low compared to that when
larger diameter hollow spheres are used. Lower polytropic index indicates that final temperature of compressed air is low and the efficiency of compression process is high.

*Figure 20: Empirical model.*
Table 5: Comparison of diameter of the hollow sphere and the polytropic index.

<table>
<thead>
<tr>
<th>Diameter (mm)</th>
<th>Polytropic Index</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.5</td>
<td>1.153</td>
</tr>
<tr>
<td>5.5</td>
<td>1.165</td>
</tr>
<tr>
<td>7.5</td>
<td>1.190</td>
</tr>
<tr>
<td>8.5</td>
<td>1.204</td>
</tr>
<tr>
<td>9.5</td>
<td>1.209</td>
</tr>
<tr>
<td>10.5</td>
<td>1.219</td>
</tr>
<tr>
<td>11.5</td>
<td>1.221</td>
</tr>
<tr>
<td>12.5</td>
<td>1.222</td>
</tr>
</tbody>
</table>

The resulting curve was used to compare the values from the experiment with the model prediction for maximum temperature for 6.5 mm and 9 mm spheres from Figure 19. Table 6 shows this comparison. It can been seen that the model predicts the final temperature of compressed air for the experiments with ±0.4% accuracy.
Table 6: Comparing the maximum temperature from the empirical model and experiment.

<table>
<thead>
<tr>
<th>Diameter (mm)</th>
<th>Maximum Temperature (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Empirical model</td>
</tr>
<tr>
<td>6.5</td>
<td>328.54</td>
</tr>
<tr>
<td>9</td>
<td>333.38</td>
</tr>
</tbody>
</table>

5.3 Effect of compression stroke time

Simulations were run to assess the effect of these spheres at a higher compression ratio of 5:1, and it was observed that when compression is done at a slower speed (Figure 21) the temperature reduction was about 7K and for a faster compression (Figure 22) it was about 32K.
In experiments for 2:1 compression ratio the inlet air pressure for the pneumatic cylinders were varied from 206.84kPa to 551.58kPa in steps of 68.95kPa to vary the compression stroke time between 7.5s and 4.7s. The first observation is that the temperature of compressed air increases as the stroke time is reduce, as expected. When the compression process is faster, the time available for the hot air in the chamber to convect the heat across the chamber wall and the spheres is less. Thus, the air leaving the chamber is still at a high temperature.

It was observed during the experiments when multiple runs were done for the same experiment, the value of pressure recorded in each of those runs were not the same. That is to say that the pressure passes through the same values in each run, but due to the response time of the sensor

Figure 21: Comparison of computational results for a slow 5:1 compression ratio of temperature without and with spheres.
the pressure values logged into the system will not be at the same instant in each of those runs. Hence, it is not possible to quantify error in measurement or getting an average value of temperature for a particular value of pressure.

![Temperature v/s Time](image)

*Figure 22: Comparison of computational results for a fast 5:1 compression ratio of temperature without and with spheres.*

To tackle this issue, both temperature and pressure values have been normalized for each experiment and those values have been compared and plotted. Figure 23, Figure 24, and Figure 25 show the comparison of experimental normalized pressure versus normalized temperature plot for stroke time of 7.5s, 6.3s, and 4.7s for cases without and with spheres respectively. Table 7 gives the account of final normalized pressure and temperature of compressed air, and change in the polytropic index without and with spheres at various stroke times. Normalized
pressure is observed to reach \( \sim 2.2 \) and normalized temperature is seen to be low for slow processes. Polytropic index \( n \) is calculated from normalized pressure and temperature data using Equation 15. Figure 26 shows the comparison of polytropic index with and without spheres for various stroke times. Polytropic index for both without and with spheres are low for slower process compared to a faster one. The difference between the polytropic indexes for a particular compression stroke time is seen to be higher for a faster process.

![Figure 23: Comparison of normalized pressure versus normalized temperature with and without spheres for stroke time of 7.5s.](image)

All the pressure-temperature plots above are observed to have more than one peak. When the air is getting compressed it heats us. But, the air close to the wall and the spheres are at a lower temperature. Since the experimental temperature data is collected at just one point, as the water
front moves closer to the top end of the compression chamber the bulk hot air in the center moves past the thermocouple giving rise to the first peak. The compression process still continues to happen and now the air close to the moving water front heats up and moves past the thermocouple to give the next peak. And the final dip is because of the small amount of air very close to the water surface which is pushed out when the outlet is opened.

Figure 24: Comparison of normalized pressure versus normalized temperature with and without spheres for stroke time of 6.3s.
Figure 25: Comparison of normalized pressure versus normalized temperature with and without spheres for stroke time of 4.7s.
Table 7: Comparison of polytropic index without and with spheres for various compressions stroke times.

| Stroke Time (s) | Without Spheres | | | With Spheres | | |
|---|---|---|---|---|---|
| | Normalized | Normalized | n | Normalized | Normalized | n |
| 7.5 | 2.216 | 1.077 | 1.102 | 2.214 | 1.068 | 1.090 |
| 6.3 | 2.223 | 1.086 | 1.115 | 2.220 | 1.071 | 1.094 |
| 5.5 | 2.218 | 1.090 | 1.121 | 2.222 | 1.072 | 1.096 |
| 5.3 | 2.217 | 1.099 | 1.136 | 2.222 | 1.074 | 1.098 |
| 5  | 2.218 | 1.102 | 1.139 | 2.208 | 1.075 | 1.099 |
| 4.7 | 2.215 | 1.117 | 1.162 | 2.216 | 1.076 | 1.101 |
5.4 Effect of gaps between the spheres

3D simulations are run to access the authenticity of modelling the layer of spheres as a plate of thickness equal to the diameter of the spheres. With the plate assumption, the spheres are considered to be tightly packed on the top surface of the water column compressing air. In the 3D simulation the spheres are modeled as it is.

From Figure 27 it is observed that the assumption of simulating the layer of the sphere as a plate of thickness equal to the diameter of the spheres gives a very close approximation. The computation time taken for the 2D simulation was ~8 hours and the time taken for the 3D simulation was about ~10 days. It can be observed that both 2D and 3D simulations results
predict similar final temperature for compressed air. Since 2D simulations take ~30 times lesser amount of time to compute the results, it is better to run 2D simulations from the point of view of computational cost.

![Image](image1.png)

**Figure 27:** Comparison of final temperature of compressed air for 2:1 compression ratio in (a) 2D simulation and (b) 3D simulation.

5.5 Comparison of analytical model and simulation model

The heat transfer per unit area from the simulation and the theoretical model have been compared and tabulated in Table 8 for the cases without spheres and with spheres. It can be seen that the results from simulation and analytical model are quite similar. It can be observed from Figure 28 that the change in temperature of the spheres is very minimal. The term
accounting for conduction within the solid domain in overall heat transfer coefficient term in the analytical model is the only term which accounts for the conductivity of the material of the spheres. Since the change in temperature is negligible so is the conduction heat transfer. Thus, concluding that the presence of the solid medium alone helps in increasing the heat transfer in the system and the thermal conductivity of the spheres do not play a significant role in temperature abatement.

*Table 8: Comparison of Heat Transfer per unit area without and with spheres.*

<table>
<thead>
<tr>
<th></th>
<th>Heat transfer per unit area (W/m²)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Without spheres</td>
</tr>
<tr>
<td>Simulation</td>
<td>257.82</td>
</tr>
<tr>
<td>Analytical model</td>
<td>292.74</td>
</tr>
</tbody>
</table>
Figure 28: Final temperature of spheres for 2:1 compression ratio.
6. Conclusions and scope for future work

6.1 Conclusion

The aim of this thesis was to study the effect of hollow spheres on temperature abatement in liquid piston compressor used for OCAES system. Various parameters like material of the spheres, size of the spheres, and the stroke time of the compression process were identified to be affecting the temperature abatement using hollow spheres. Experimental and computational studies have been conducted based on the existing experimental setup to understand the effect of these above mentioned parameters on temperature abatement. An analytical model and an empirical model have been developed to help us understand the temperature abatement phenomena using hollow spheres better.

- Initial experiments were conducted using high conductivity hollow spheres made of Silicon Carbide and outer diameter 6.5mm. It was observed that the polytropic index of compression process for a 2.2:1 compression ratio was 1.08 with spheres when compared to 1.15 without spheres. Thus, improving the overall efficiency of the OCAES system by about 10%. [36, 37]

- The material and size of the hollow spheres were varied. Materials with conductivity much lower (~250-500 times) than that of SiC and diameter of 6.5 mm and 9 mm have been tested in this setup. It was observed that the material of the spheres did not have a significant effect in temperature abatement. However, the smaller diameter (6.5 mm)
spheres are seen to be more effective in temperature abatement than the larger diameter (9 mm) spheres.

- Stroke time of compression process was varied by varying pressure inlet air (206.84 kPa to 551.58 kPa) to the double acting pneumatic cylinders used to operate the liquid piston compressor. Temperature abatement was seen to be more pronounced at faster compression stroke times (high pressure air input to the cylinders).

- Analytical model was developed by assuming air to be an ideal gas to assess the excess heat transfer provided by addition of a layer of hollow spheres into the liquid piston compressor. The model and 2D simulations have predicted similar values for the heat flux for both without and with sphere cases.

- Empirical model was developed based on the data from 2D simulations in which diameter of spheres were varied from 4.5 mm to 11.5 mm in steps of 1 mm. From the simulation, the final temperature and pressure of air was taken to calculate the polytropic index of compression process. A cubic not-a-knot spline was used to fit the data and the polytropic index from the resulting curve was compared with the experimental temperature for 6.5 mm and 10 mm diameter spheres and are seen to be a close match.

- 3D simulations were run to assess the authenticity of modelling the layer of spheres as a solid plate in 2D. It was seen that both the models produce similar results and the 3D simulations took ~10 days to compute the results, whereas the 2D simulations took ~8 hours.
6.2 Scope for future work

- Since it is observed that the material of the spheres do not have as significant an effect as the size of the spheres, further studies can be done of assessing the exact mechanism of heat transfer mechanism at the interface of air-solid-water.

- All the experiments have been run for a 2.2:1 compression ratio in the small experimental setup. The same method can be tried experimentally in a larger setup and higher pressure ratios and see how effective hollow spheres are at high pressure.

- In the empirical model spheres of diameter in the range 4.5 mm to 11.5 mm have been studied. It can be seen that the model is almost linear from 4.5 mm to 8 mm and non-linear above 8 mm. This model cannot quantify the behavior of spheres below 4.5 mm. In future, experimental studies can be conducted to see the effect of diameters below 4.5 mm and study the behavior of the model below 4.5 mm. It can also be extended to understanding the effect of suspending micro and nano particles in the chamber.
References


[34] Francesco Balduzzi, Giovanni Ferrara, Alberto Babbini, Riccardo Maleci, “Reciprocating compressor cylinder’s cooling: a numerical approach using computational fluid dynamics with

