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The development of a sub-atmospheric two-phase thermosyphon natural gas preheater using a lumped capacitance model

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Abstract Natural gas (NG) for domestic and industrial use is distributed around the country at high pressure. To counteract the Joule-Thompson effect, pre-heating of natural gas is required prior to pressure reduction. Most preheaters used on national gas networks are in the form of water bath heaters, where a closed tank houses a burner which delivers energy to the fluid in situ, which in turn heats an exchanger through which the NG flows. These systems usually have a low efficiency, and as a result of thermal inertia have a long time lag to accommodate changes in NG mass flow rates.

A new preheating system has been designed utilising sub-atmospheric two-phase type thermosyphon theory. The main advantage of using a sub-atmospheric thermosyphon to preheat NG is the improved response time, due to the reduction in working fluid volume and the thermosyphon operating at lower temperatures resulting from partial vacuum.

A numerical model of the thermosyphon has been developed using a lumped capacitance method. Simulations of the thermosyphon preheater system over a 24 hour load cycle suggest that the thermosyphon system could reach an efficiency of approximately 84% over the period.

1. Introduction

Natural gas travels around distribution networks at high pressures, before the pressure is reduced for safe use for the end user. The gas must be pre-heated to overcome the Joule-Thompson effect, which causes the gas temperature to drop rapidly. If the gas is not preheated components within the reduction facility can freeze over. This can cause stresses on the pipes, fittings and any pressure reduction components. It can also cause the gas to hydrate which can result in ice forming within the pipeline [1].

Pre-heating the gas is common practice, and to date, within the UK, the majority of pressure reducing stations remain served by water bath preheaters. The technology is robust and proven to work, however, at a time when sustainability and energy saving are goals of all industry, this is one area which has not shown unanimous progress in the last few decades. One of the reasons for this is that the systems currently in service have demonstrated a long life span, often up to 30-40 years.

Thermosyphons are a form of heat pipe, where energy is added to the system at one location and extracted from another. The process involves the movement of a working fluid due to natural convection cycles due to the heat source and sink. The working fluid can undergo a phase change, to utilise high rates of heat transfer associated with condensation, a basic schematic is shown in Figure 1.

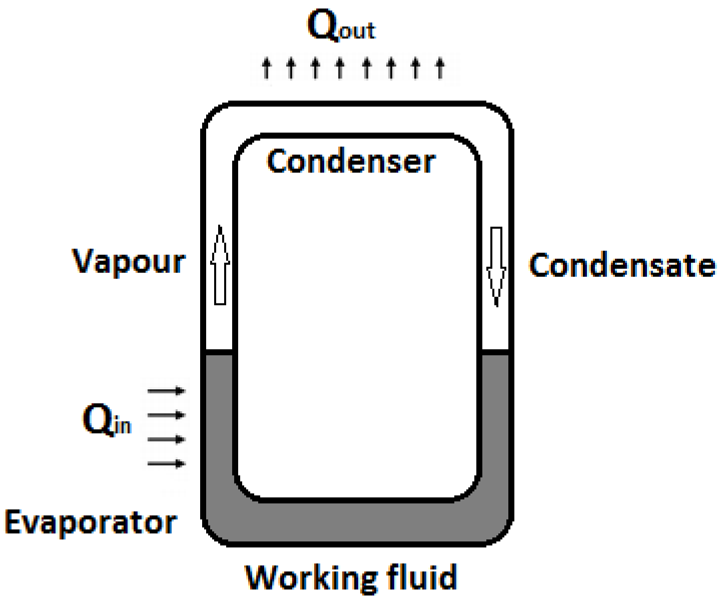

Figure 1. Sketch of a two phase closed loop thermosyphon

Heat is added to the working fluid within the evaporator section which undergoes a phase change, and travels up to the condenser as vapour. The vapour then condenses, rejecting its heat to the heat sink in the process. The working fluid then travels back to the evaporator as condensate. The movement of the fluid is achieved through natural convection cycles due to gravity and buoyancy effects; this means that there is no requirement for mechanical pumps to move the working fluid around the system. As such, they are cheap to construct and run, and are reliable due to containing no moving parts.

Thermosyphons are used across a broad range of many industries, from the nuclear industry, the oil transportation industry, personal computing and now pre-heating. Much literature exists on thermosyphons due to extensive research over an extended period of industrial application.

There has been a great deal of experimental research, such as the research of Noie et al [2] which considers investigations on a vertical two-phased closed thermosyphon under vacuum. Various parameters (input heat transfer rates, filling ratio of the working fluid and the evaporator lengths) were studied to check the heat transfer performance. Their observations concluded that along the evaporator section, the temperature decreases with the increase of the filling ratio to a critical value at a constant evaporator length. As the evaporator length is increased further, the critical filling ratio is decreased. They also concluded that the maximum heat transfer for each evaporator length occurs at a different filling ratio.

Jouhara et al [3] investigates the performance of a vertical thermosyphon with four different working fluids; water, FC-84, FC-77 and FC-3283. In their analysis they considered seven nucleate pool boiling heat transfer correlations, and concluded that overall they all compared well to the water-charged thermosyphon results. The water-charged thermosyphon outperformed the other working fluids in terms of working resistance and maximum heat transport capabilities, except when considering low operating temperatures, due to the other working fluid saturation temperatures being lower.

A detailed review on existing experimental studies for small dimension closed loop two-phase thermosyphons (CLTPT) has been reported by Franco and Filippeschi [4], where some interesting and sometimes contradictory elements were observed. It was found that the aims of the experimental research were often different, and therefore provided a different outcome. The lack of general results causes difficulty in the application of CLTPTs in different operating conditions. The complexities of conducting experimentation due to the highly sensitive nature of equipment and measurement errors, which the presence of air can cause, were shown.

There has also been research into the numerical modelling of heat pipes and thermosyphons such as Ferrandi et al [5] simulating the transient and steady state operation of a sintered heat pipe, using a lumped capacitance method. They validated their code with previously published data.

Ziapour and Shaker [6] conducted a numerical study on transient and steady state behaviours of a TPCT represented by a closed vertical container. The lumped capacity in conjunction with the thermal network model was employed using a modified Runge-Kutta method to enhance the time integration. It was shown that the condenser surface temperature in the steady state case was independent of the convective effects.

Angelo et al [7] consider the use of a thermosyphon type method to pre-heat natural gas in Brazil, where the evaporator and condenser were split into separate vessels, with connecting pipes. The natural gas runs through a heat exchanger which uses steam as the second phase for heat extraction. Water was used as the working fluid.

The design adopted in the current work is similar to that of Angelo et al, with two fundamental distinctions; the preheater runs at a partial vacuum which was deemed to improve the overall efficiency of the heat transfer, and a water/ glycol mixture is used as the working fluid.

The theory behind the current design is discussed in the next section, followed by a description of the lumped capacitance model which has been developed to assist the design. Finally, results for a thermosyphon preheater using the lumped capacitance model for a typical 24 hour loading requirement of natural gas through a preheater site are shown.

2. Design of Thermosyphon Preheater

The preheater was designed to have the evaporator and condenser in separate locations, connected by piping. The evaporator was designed to have a burner with a serpentine coil within the bottom half, whilst the condenser had a tightly packed heat exchanger coil running through a small space. The working fluid was water with a 30% glycol content to stop freezing, and the whole system was evacuated to an absolute pressure of 0.02MPa. A sketch of the design idea is shown in Figure 2.

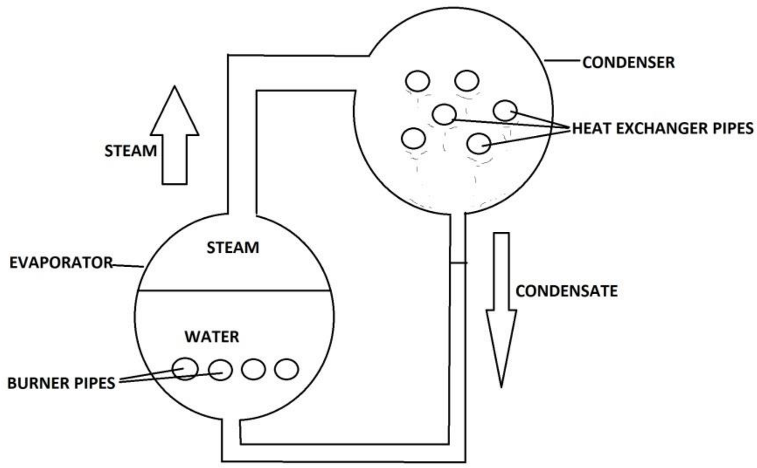

Figure 2. Sketch of preheater in 2D.

The design of the preheater was conducted around the heat transfer required for the natural gas, in a top down method. The maximum volumetric flow of natural gas through a pressure reducing station was the critical design point. Heat transfer and flow rate calculations were developed to design the heat exchangers for any site conditions. This paper will focus on a particular site on a gas distribution network with a bespoke design for specific diurnal and yearly long loading requirements, but the methodology could be adopted to design a system for any site. Each site will have a maximum volumetric flow rate and maximum pressure, and the required pressure and temperature after choke (pressure reduction). There are also certain design requirements from the heat exchanger coil such as maximum velocity of gas within the pipes and maximum pressure drops.

3. Theory

The heat required for the pre-heating of the natural gas was calculated by considering the conditions of the natural gas flowing into the preheater and also the conditions required after choke. The velocity, temperature and pressure before and after pressure reduction must be considered, and the heat required can be calculated. The enthalpy of the natural gas is used to calculate the heat required from inlet to outlet, by matching the enthalpy of the natural gas after choke to the enthalpy at the outlet of the preheater. The difference in enthalpies from the inlet and outlet can then be used, with the mass flow through the heat exchanger, to calculate the power required to raise the enthalpy. Therefore, the site conditions and material properties of the natural gas drive the calculations, mainly through the enthalpy. The enthalpy for natural gas is found for the pressure and temperature conditions after choke. This is then used as the enthalpy for the heat exchanger outlet, and with the pressure at the outlet, the temperature can be determined. The heat required was calculated from [8, 9]

$$Q_{NG} = \dot{m}\Delta H \tag{1}$$

where, $\dot{m}$ is the natural gas mass flow, and

$$\Delta H = H_{out} - H_{in} \tag{2}$$

where, H is the enthalpy, and subscripts in and out represent the conditions into and out of the heat exchanger respectively.

From the heat addition required, and maximum velocity allowed, the heat exchanger can be designed. The internal heat transfer coefficient was calculated from [10]

$$h_{in} = \frac{k_{NG}}{D_{int}}\left(0.0214\,(Re^{0.8} - 100){Pr_{av}}^{0.4}\right) \tag{3}$$

where k_{NG} is the average thermal conductivity of the natural gas from inlet to outlet, D_{int} is the internal diameter of the heat exchanger, Re is the Reynolds number, Pr_{av} is the average Prandtl number.

Concerning the condensation, the heat transfer for a horizontal tube with diameter D_{ext} is defined from the Nusselt analysis for a laminar regime [11] as

$$h_c = 0.729\left[\frac{g\rho_l(\rho_l - \rho_g)h'_{fg}{k_l}^3}{\mu_l(T_{sat} - T_s)D_{ext}}\right]^{1/4} \tag{4}$$

where ρ is the density of the working fluid with subscripts l and g corresponding to the liquid and vapour forms respectively, μ_l the dynamic viscosity of the condensate film, k_l the thermal conductivity of the film, D_{ext} is the external diameter of the heat exchanger coil, T_{sat} is the saturation temperature of the working fluid, T_s is the temperature of the pipe surface and is calculated as the average of T_{sat} and T_{in}, where T_{in} is the natural gas temperature in. The modified latent heat of vaporisation, h'_{fg}, is calculated as [8, 9]

$$h'_{fg} = h_{fg} + \left(0.68\, c_{p,l}(T_{sat} - T_s)\right) \quad \textbf{(5)}$$

The overall heat transfer coefficient was calculated as [8, 9]

$$U_{TOT} = \left[\frac{1}{h_c} + \frac{D_{ext}\ln(D_{ext}/D_{int})}{2 \times k_{pipe}} + \frac{1}{h_{int}}\frac{D_{ext}}{D_{int}}\right]^{-1} \quad \textbf{(6)}$$

where D_{int} and D_{ext} refer to the internal and external diameters of the heat exchanger coil, and k_{pipe} is the thermal conductivity. The external surface area was then found from

$$A_{s,req} = \frac{Q_{NG}}{U_{TOT}\left(\Delta T_{lm} - T_{foul}\right)} \quad \textbf{(7)}$$

where ΔT_{lm} is the log mean temperature difference, and T_{foul} is a temperature allowance for fouling of the heat exchanger coil.

The heat required for the temperature raise in the natural gas can be used to design the evaporator. The burner must first raise the bulk temperature of the volume of the fluid before vaporisation can occur. Due to the glycol content of the working fluid, the specific heat capacity was greater than that of water, and due to the vessel running under a partial vacuum, the saturation temperature of water was reduced.

The burner can be designed to meet the heat addition required for the natural gas flow. Simply, there must be enough steam to raise the natural gas by the required amount,

$$\dot{m}_s = \frac{Q_{NG}}{h'_{fg}} \quad \textbf{(8)}$$

where $\dot{m}_s$ is the mass flow of steam. The burner must first heat the working fluid up to the saturation temperature

$$Q_{rise} = m_l C_{p,l}(T_{sat} - T_1) \quad \textbf{(9)}$$

where Q_{rise} is the power required to raise the temperature of the working fluid from the initial working fluid temperature, T_1, to the saturation temperature, T_{sat}, m_l is the mass of the working fluid and $C_{p,l}$ is the specific heat of the working fluid.

The burner must deliver enough power to overcome the losses to atmosphere associated with each vessel, and account for the efficiency of the burner to deliver the power to the working fluid.

$$Q_{burner} = Q_{rise} + Q_{loss} + Q_{NG} \quad \textbf{(10)}$$

To analyse the transient behaviour of the two-phase closed loop thermosyphon considered in this study, a numerical model was required.

3.1. Thermosyphon Lumped Capacitance Transient Model

Figure 3 shows the lumped capacitance model in a resistance network form. Convection heat transfer is present between the following parts: burner outer surface and the working fluid, working fluid and inner evaporator chamber, working fluid and evaporated fluid, evaporated fluid and inner evaporator chamber (and transporting-pipes), evaporated fluid and the natural gas (NG) condenser outer surface, and finally, the NG inside pipe surface and the NG-fluid. The conduction heat transfer is represented in the burner and condenser pipes, evaporator and condenser chamber walls, and the fluid transport pipes.

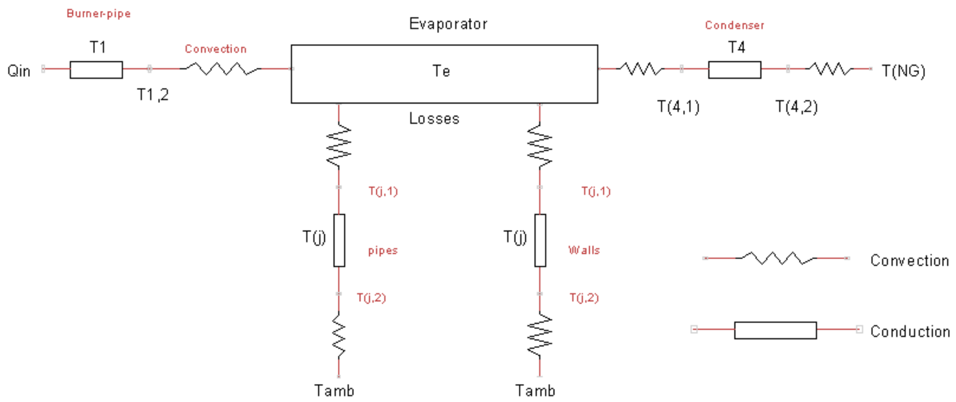

Figure 3: Schematic diagram of the network model

The input power, Q_{in}, is defined as the power delivered by the combustion of natural gas, rather than the calculation of combustion. The burner pipe temperature, the condenser pipe temperature, the working fluid temperature and the natural gas temperature with extra pipes and wall temperatures were unknown variables to be solved by the network model above.

The burner is represented in the model by a pipe, and the temperature, T_1, is at the midpoint of the burner wall, at radius, R_1, which can be represented by the transient conduction equation through the pipe mass, m_1, specific heat capacity, C_{p1}, conductivity, k_1, length, L_1, and outer radius, R_{1o}, as:

$$m_1 C_{p1} \frac{\partial T_1}{\partial t} = Q_{in} - \frac{2\pi L_1 k_1}{\ln(R_{1o}/R_1)} \left(T_1 - \frac{k_1 T_1 + h_e R_{1o} T_e \ln(R_{1o}/R_1)}{k_1 + h_e R_{1o} \ln(R_{1o}/R_1)} \right) \quad \textbf{(11)}$$

where t denotes the time, and h_e is the evaporator heat transfer coefficient to the working fluid that has a temperature T_e.

Similarly, for the condensation pipe, at the middle radius, R_4, the temperature is given by

$$m_4 C_{p4} \frac{\partial T_4}{\partial t} = -\frac{2\pi L_4 k_4}{\ln(R_{4o}/R_4)}\left(T_4 - \frac{k_4 T_4 + h_e R_{4o} T_e \ln(R_{4o}/R_4)}{k_4 + h_e R_{4o} \ln(R_{4o}/R_4)}\right) - \frac{2\pi L_4 k_4}{\ln(R_4/R_{4i})}\left(T_4 - \frac{k_4 T_4 + h_{NG} R_{4i} T_{NG} \ln(R_4/R_{4i})}{k_4 + h_{NG} R_{4i} \ln(R_4/R_{4i})}\right) \tag{12}$$

The condensation pipe properties are defined in a similar fashion as those of the burner pipe as above, in addition to the inner radius, R_{4i}, h_c and h_{NG} represent the heat transfer coefficients at the condenser outer and inner surfaces, respectively. The natural gas temperature is represented by T_{NG}.

The pool temperature is assumed to be uniform throughout the evaporator boiling domain and is given by

$$m_e C_{pe} \frac{\partial T_e}{\partial t} = \frac{2\pi L_1 k_1}{\ln(R_{1o}/R_1)}\left(T_1 - \frac{k_1 T_1 + h_e R_{1o} T_e \ln(R_{1o}/R_1)}{k_1 + h_e R_{1o} \ln(R_{1o}/R_1)}\right) + \frac{2\pi L_4 k_4}{\ln(R_{4o}/R_4)}\left(T_4 - \frac{k_4 T_4 + h_c R_{4o} T_e \ln(R_{4o}/R_4)}{k_4 + h_c R_{4o} \ln(R_{4o}/R_4)}\right) - Q_{loss} \tag{13}$$

Q_{loss} accounts for pipes and vertical wall losses to the ambient environment.

To complete the set of equations, it is necessary to evaluate both the evaporator and condenser heat transfer coefficients h_e and h_c respectively. For a cylinder of inner and outer radii R_{in} and R_{out}, h_e is given by [6]

$$h_e = 0.32\left(\frac{g^{0.2} {\rho_l}^{0.65} {k_l}^{0.3} C_{p,l}^{0.7} (\rho_l - \rho_g)}{\rho_g^{0.25} h_{fg}^{0.4} \mu_l^{0.1}}\right)\left(\frac{P_{sat}}{P_{atm}}\right)^{0.3} \dot{q}^{0.4} \tag{14}$$

The condensation heat transfer coefficient, h_c, was shown earlier in equation 4.

The resulting set of equations were first order linear differential and were solved by a fourth order Runge-Kutta method.

4. Load profile

In order to assess the capabilities of the thermosyphon, a 24 hour load cycle for a specific site was used. As the flow rate through the preheater site changes continuously throughout the day due to demand, the heat required to heat the natural gas changes, as displayed in Figure 4, which shows peak gas requirements in the morning and evening, before and after the average working day, when the demand is the highest.

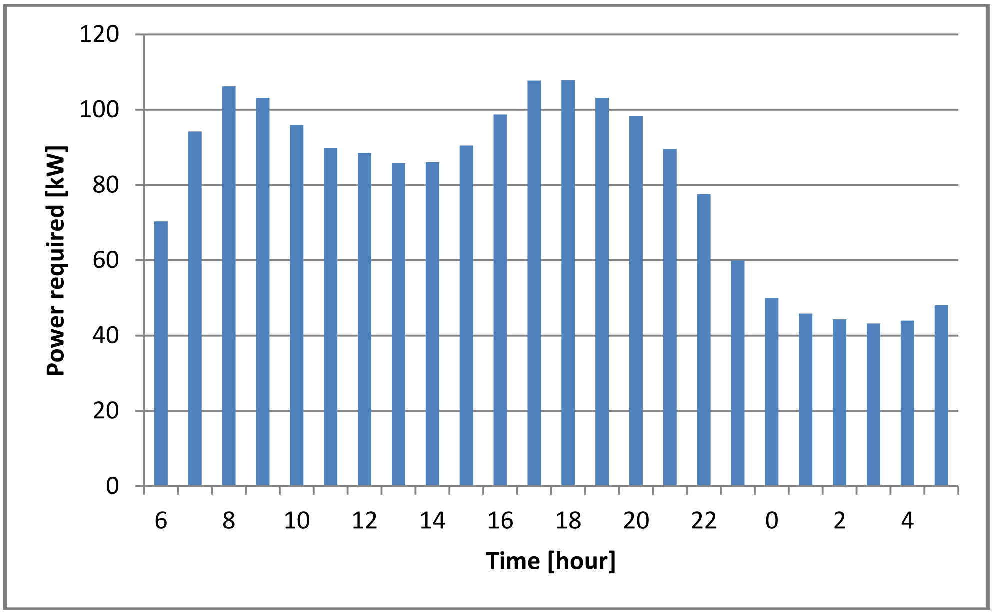

Figure 4. Twenty four hour load profile.

To simulate the thermosyphon for the 24 hour cycle, the control of the system was set so that the burner was kept on until the outlet temperature of the natural gas reached the required temperature. Once the temperature was reached, the burner automatically turned off, and the system continues to run in this manner. For the results shown, the ambient losses were switched off.

5. Results

A temperature plot shown in Figure 5 shows the temperatures at the locations associated with the thermosyphon. The natural gas outlet temperature initially takes 200 seconds to reach the required output temperature, but remains fairly constant thereafter. The burner wall, vapour, working fluid and condenser wall all follow the same trends, which are directly linked to the flow rate of natural gas, and therefore required power shown in Figure 5.

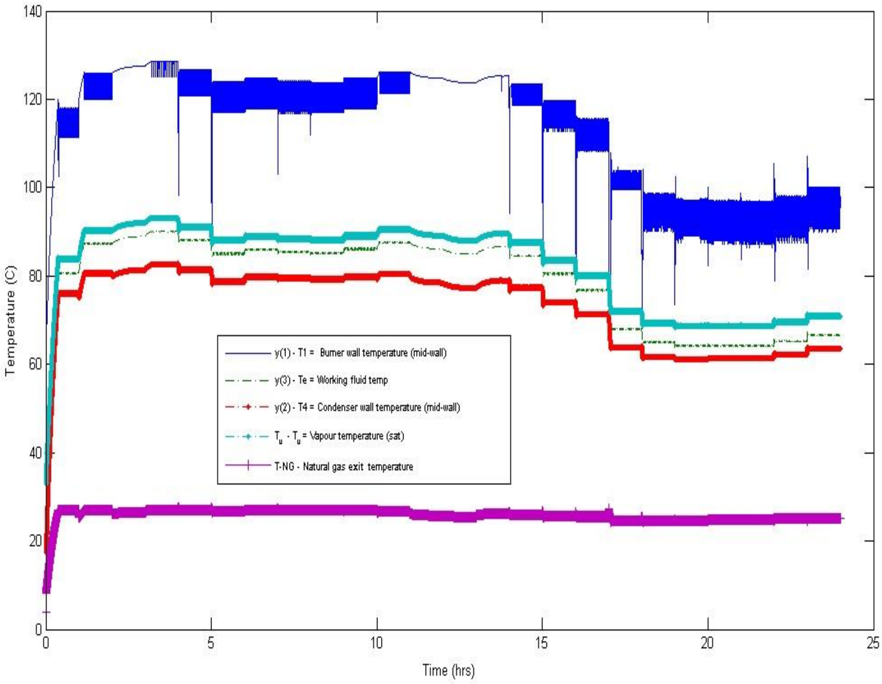

Figure 5. Temperature at locations for 24 hour load.

Figure 6 shows the accumulated efficiencies associated with the thermosyphon preheater. The accumulated efficiencies were calculated from the accumulated power out, in terms of the natural gas rise in temperature, divided by the accumulated power in from the burner. Each bar represents the rolling accumulation of efficiency, therefore, the last bar represents the mean efficiency over the 24 hour period.

The thermosyphon preheater manages an efficiency of 65% in the first hour and increases in efficiency thereafter. It must be noted that atmospheric based thermal losses have not been considered, therefore, an overall reduction in the predicted efficiencies is expected when these are accounted for in the model.

The rapid response time of the thermosyphon design is apparent during higher rates of load requirement, managing an efficiency of 65% over the first hour; this is the result of a lower saturation temperature, and utilising the high heat transfer of condensation.

The average efficiency of the thermosyphon over the 24 hour period is 84%. In the study, the power from combustion by the burner used in the system was 98%. The thermosyphon burner efficiency used is that measured by the burner manufacturer [12].

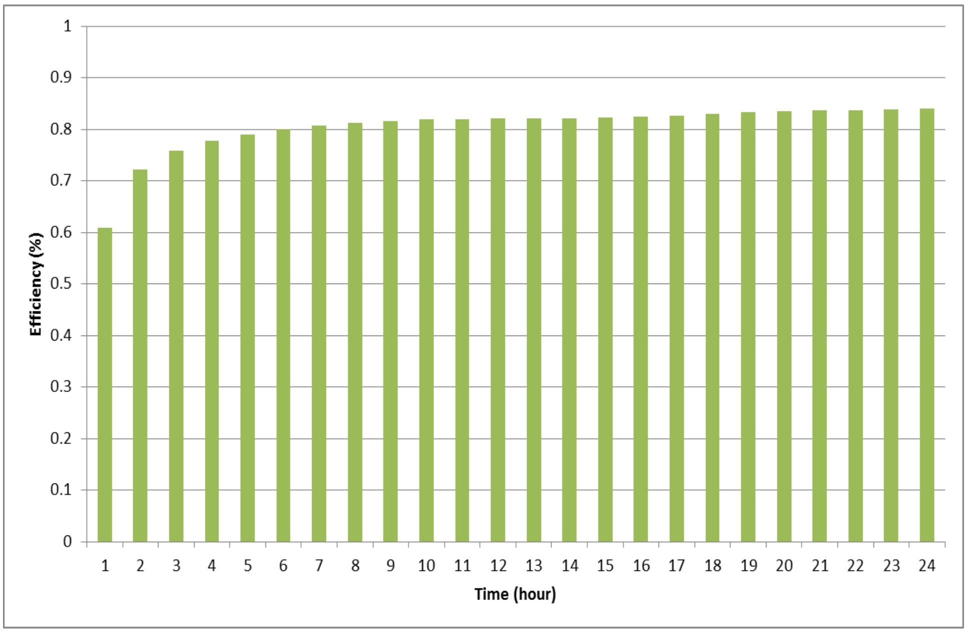

Figure 6. Accumulated efficiency for thermosyphon preheaters over 24 hour period.

5. Conclusions

A new preheater has been designed utilising sub-atmospheric two phase thermosyphon theory. A numerical model of the thermosyphon has been developed using a lumped capacitance method. A 24 hour period is simulated and efficiencies are calculated.

In the simulations, the thermosyphon shows a fast response time and high level of efficiency. The use of a sub-atmospheric pressure allows vapour to be generated at low temperatures, and due to the high energy levels transferred through the condensation process, a large amount of heat transfer can occur, thus, the efficiency is high.

The model can be used to assess various geometrical design ideas, and compare against other sites for which load data is available.

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