Thermal integration of SOFC and plate heat exchanger desorber

Venkataraman, Vikrant; Pacek, Andrzej; Steinberger-Wilckens, Robert

DOI:
10.1149/06801.0221ecst

Thermal Integration of SOFC and Plate Heat Exchanger Desorber

Vikrant Venkataraman[ab], Andrzej W Pacek[a], Robert Steinberger-Wilckens[ab],

[a] Centre for Fuel cell & Hydrogen Research, University of Birmingham
[b] School of Chemical Engineering, University of Birmingham, Edgbaston, B15 2TT UK

A Plate Heat Exchanger (PHE) desorber is thermally integrated with an SOFC stack via a specially designed tube in tube heat exchanger with internal fins in which thermal oil is heated to the required desorber temperature and then serves as the coupling fluid in the PHE desorber. A modelling approach has been adopted where the PHE desorber is solved for heat and mass transfer using MATLAB & EES and the tube in tube heat exchanger with internal fins has been modelled and optimized using COMSOL multiphysics. The results show that the PHE desorber is able to produce the required quantity of refrigerant needed for a 1 kW cooling load. The use of PHEs as desorbers not only gives a high heat transfer surface area but also leads to considerable reduction in desorber volume when compared to conventional falling film desorbers.

Introduction

SOFCs are a potential candidate for use as Auxiliary Power Units (APU's) on board heavy duty trucks. Heat from the SOFC exhaust is generally used either for cabin heating or for recuperative heat exchanger and even after that there is still a considerable amount of high quality heat available. Not using the heat from an SOFC stack is tantamount to using only half the available useful energy from the fuel.

The ongoing research work focusses on the design and development of a compact Vapour Absorption Refrigeration System (VARS) unit driven by heat from an SOFC stack for a refrigerated truck application. A thermally driven compact refrigeration unit offers the following advantages over an electrically driven refrigeration unit:

i) The refrigeration unit is decoupled from the main diesel engine which in turn reduces the load on the main diesel engine. Considerable diesel savings can be achieved when compared to the conventional case.
ii) Ensures quieter operation of the refrigeration unit thereby removing any restrictions on truck delivery times which in turn means trucks could make deliveries at any time of the day.
iii) Since the heat from the SOFC is used, the electricity generated is available for other purposes. This enables design of hybrid configurations for refrigerated trucks.
iv) The refrigeration unit is being driven by free thermal energy which would otherwise be dissipated to the environment.

In a VARS, the desorber and the absorber are the most crucial components as they act as replacement for the electrically driven compressor of the VC (Vapour Compression) system. Since the intended application is for trucks, it is absolutely essential to have a very compact system in order for the VARS to compete with the conventional VC system. PHEs are very compact heat exchangers with high heat transfer surface area and hence can be packaged in a small volume and have been proposed for use as desorbers in VARS.

Within the broader framework, this report presents results obtained from thermal integration of a PHE desorber with an SOFC stack. The whole system has been designed to cater to a 1 kW cooling load and the performance of the desorber is then evaluated. Appropriate design maps have been drawn based on the results obtained, to suit larger and smaller cooling loads.

System design & architecture

The architecture for thermal integration of an SOFC stack and PHE desorber is shown in Figure 1.

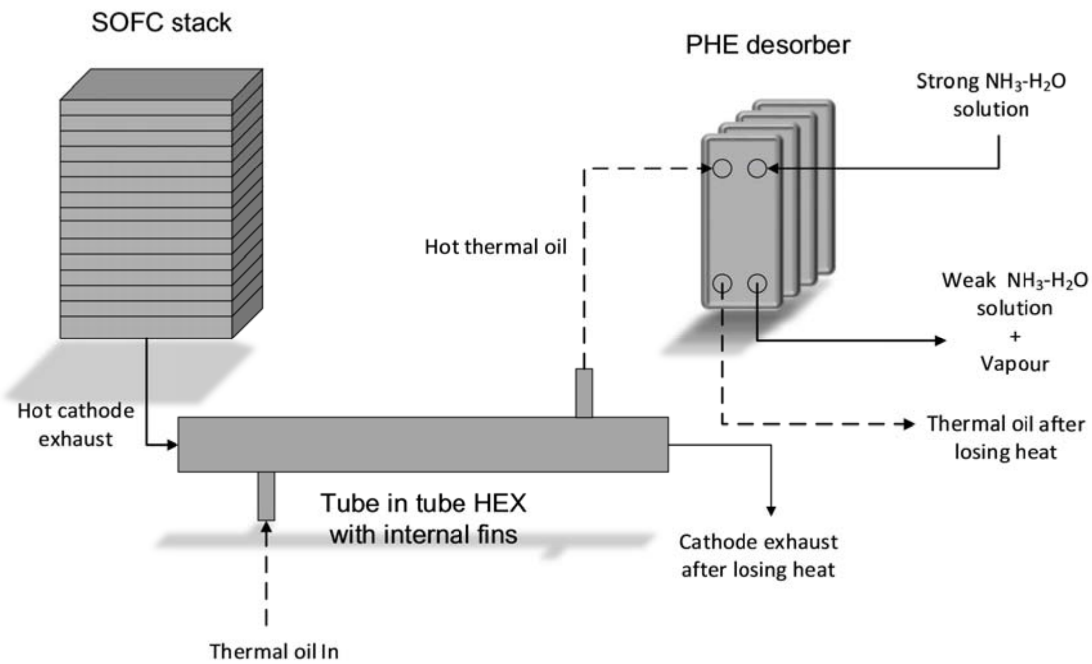

Figure 1: System architecture for thermal integration of SOFC & PHE desorber

The hot cathode exhaust from the SOFC stack is passed through a tube in tube heat exchanger with internal fins where the thermal oil is heated to the required desorber temperature. The hot thermal oil is then passed through the PHE desorber where it transfers its heat to the NH_3-H_2O solution. This results in flow boiling of NH_3-H_2O solution and generation of vapour and weak solution at the outlet of the PHE desorber which are then separated in a vapour separator. The generated vapour at high pressure is then passed on to the condenser (not shown in Figure 1).

SOFC cathode exhaust flow rate

The cathode exhaust flow rate from an SOFC stack is dependent on the fuel utilization factor. Higher fuel utilization factors will lead to lower cathode exhaust flow rates and vice versa (1). The variation of cathode exhaust flow rate with fuel utilization factor from a 1 kW & 5 kW SOFC stack is shown in Figure 2. This data is important for modelling the tube in tube heat exchanger with fins.

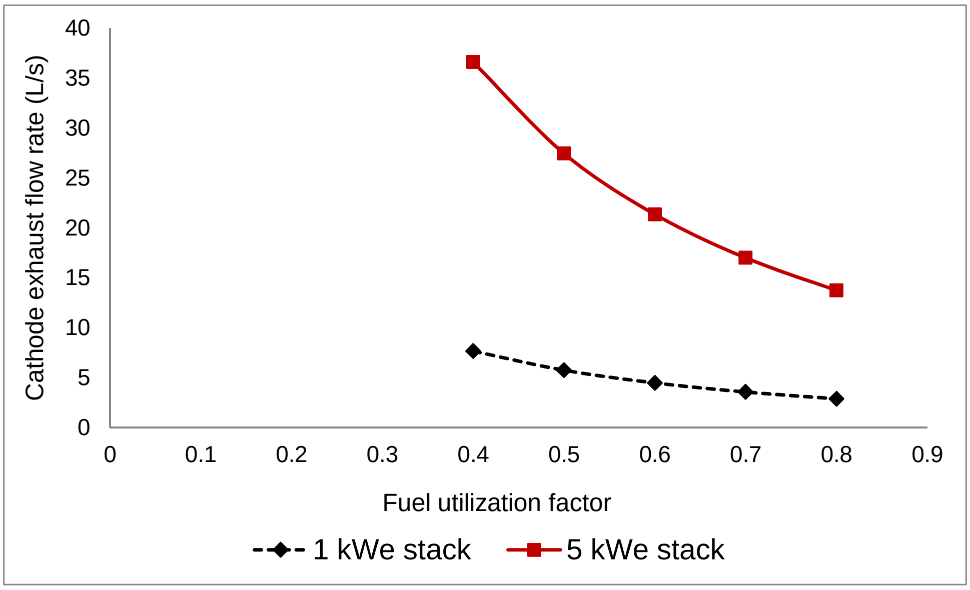

Figure 2: Variation of cathode exhaust flow rate with fuel utilization factor

.

Modelling & Optimization of tube in tube heat exchanger

The tube in tube heat exchanger plays a crucial part in the whole system. The performance of this heat exchanger during steady state operation is critical in ensuring the thermal oil gets heated to the required temperature needed at the desorber. After carrying out optimization studies, the final geometrical dimensions for the heat exchanger were fixed and steady state simulations performed to check the heating of the thermal oil in the heat exchanger.

Physics employed & governing equations

COMSOL multi-physics was employed to study the performance of the heat exchanger. The different physics employed in various domains of the geometry and the corresponding equations are as follows:

Heat transfer in solids – Stainless steel tube
Heat transfer in fluids – Thermal oil & Air
Laminar flow – Thermal oil & Air

Air is modelled as compressible flow and oil as incompressible flow.

Energy Equation

$$\rho C_p u.\nabla T = \nabla.(k\nabla T) + Q \quad [1]$$

Heat transfer in fluids

$$\rho C_p u.\nabla T = \nabla.(k\nabla T) + Q + Q_{vh} + W_P \quad [2]$$

Navier Stokes Equation for compressible flow

$$\rho(u.\nabla)u = \nabla.\left[-pl + \mu(\nabla u + (\nabla u)^T) - \frac{2}{3}\mu(\nabla.u)l\right] + F \quad [3]$$

Navier stokes equation for incompressible flow

$$\rho(u.\nabla)u = \nabla.[-pl + \mu(\nabla u + (\nabla u)^T)] + F \quad [4]$$

Continuity Equation

$$\nabla.(\rho u) = 0 \quad [5]$$

Where,
ρ = density (kg/m^3),
C_p = specific heat capacity (J/kg K),
T = temperature (1), u = velocity of the respective fluid (m/s),
p = pressure (1),
μ = dynamic viscosity of the fluid (Pa.s)

It is desirable to use a thermal oil that has got a higher degradation temperature because constant heating and cooling of the oil will cause it to degrade. The thermal oil used for modelling is *Paratherm HR* which has got a higher degradation tolerance of around 644 K. The properties of Paratherm HR can be found from (2) and were fed as temperature dependent equations in COMSOL multi-physics.

Geometry of the heat exchanger

The optimized geometry of the heat exchanger is shown in Figure 3 & 4 respectively.

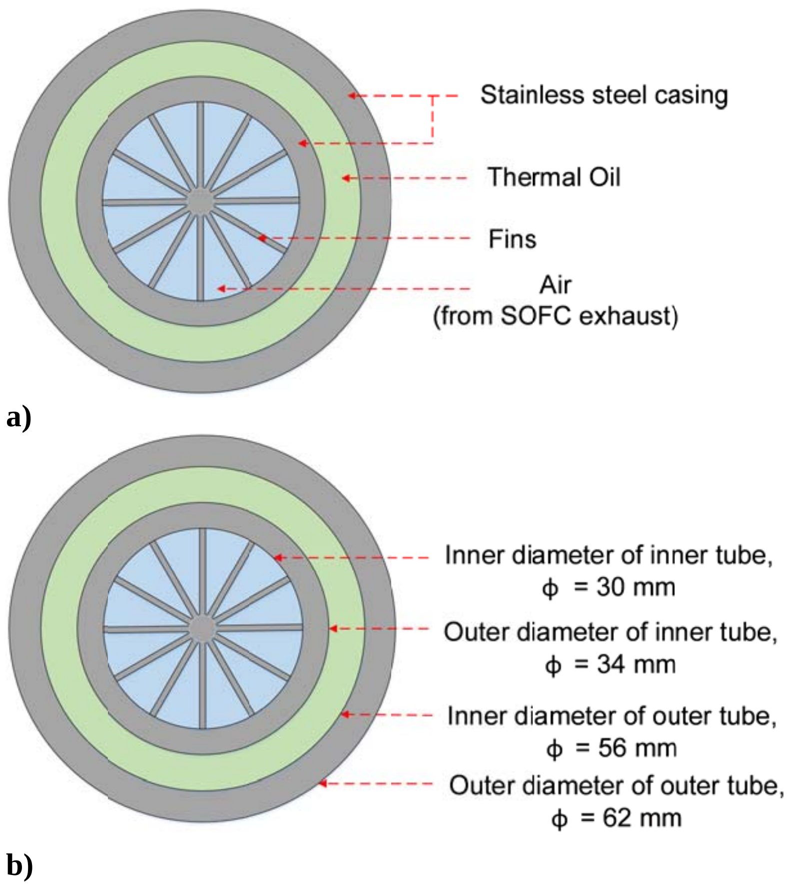

Figure 3: Cross sectional view of heat exchanger
a) Different domains present, b) Geometric dimensions

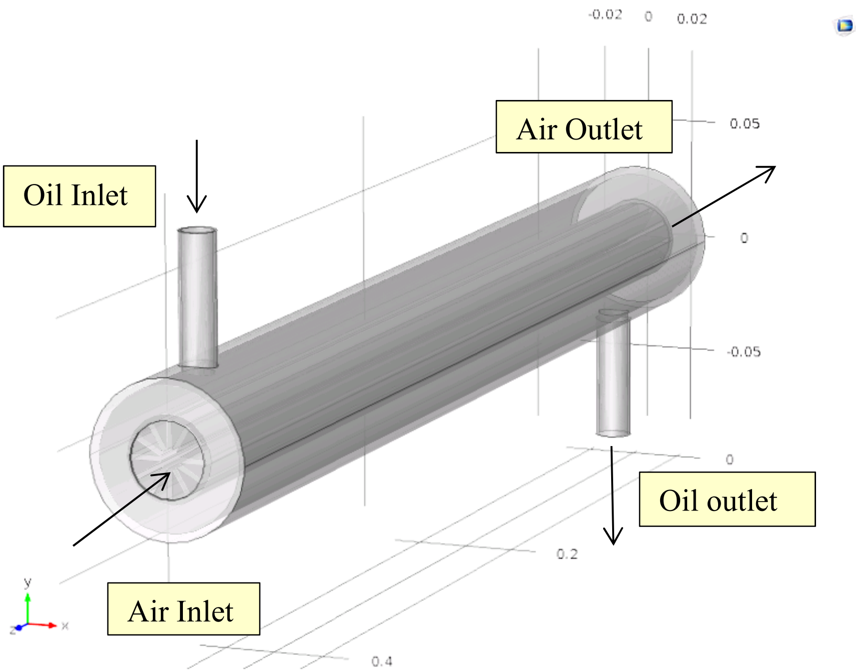

Figure 4: Complete 3D view of the heat exchanger

The operating parameters for the heat exchanger are given in Table I.

Table I: Operating parameters of the fluid for the heat exchanger

Parameter	Value
Volume flow rate of air	0.00574 m^3/s
Mass flow rate of oil	0.02 kg/s
Inlet temperature of air	973.15 K
Inlet temperature of oil	300.15 K

Modelling of plate heat exchanger desorber

Two numerical modelling platforms – MATLAB & EES (Engineering Equation Solver) were used to model the PHE desorber. The two platforms are not linked automatically; hence the user has to manually feed in the outputs from one platform into the next. The interlinking of the two platforms is shown in Figure 5.

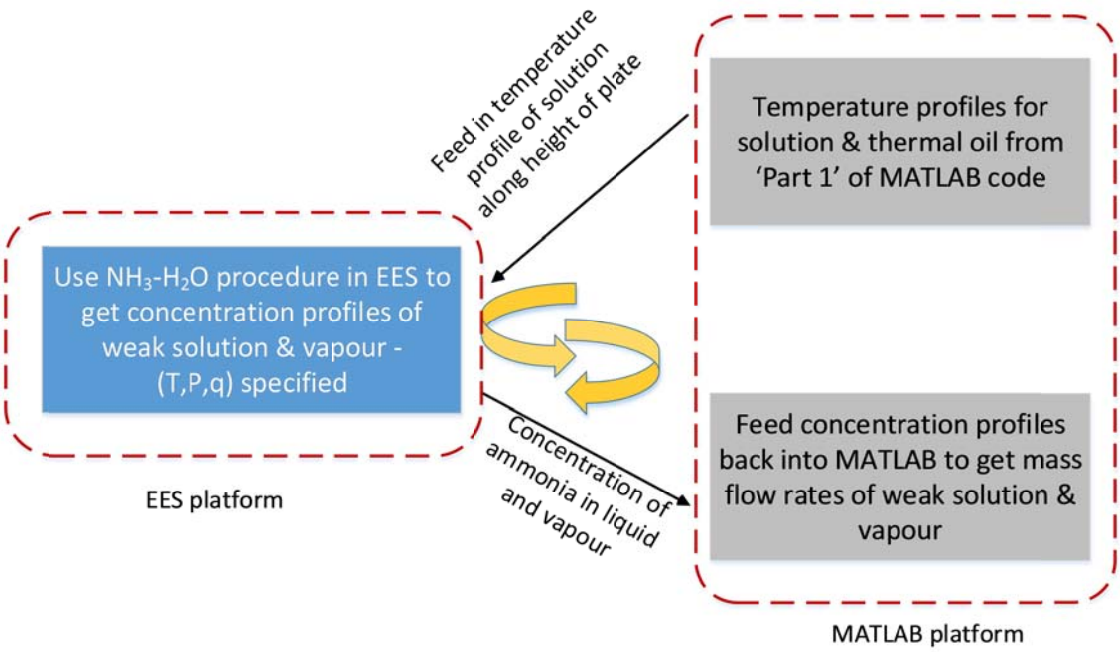

Figure 5: Block diagram showing linking of MATLAB and EES platforms

<u>Modelling methodology</u>

A segmental approach is adopted for modelling the PHE desorber where the outputs from one segment serve as inputs for the second segment and so on. The schematic of the segmental approach is shown in Figure 6. The desired outputs from modelling the PHE desorber are as follows:

- Mass flow rate of refrigerant produced at the outlet of one plate.
- Mass fraction of refrigerant produced at the outlet of one plate.
- Outlet temperature of solution.

- Determination of number of plates required in plate heat exchanger.
- Identification of boiling and heating zones within the plate heat exchanger.
- Heat flux distribution along the plate of the PHE.
- Local heat transfer coefficients along the plate.

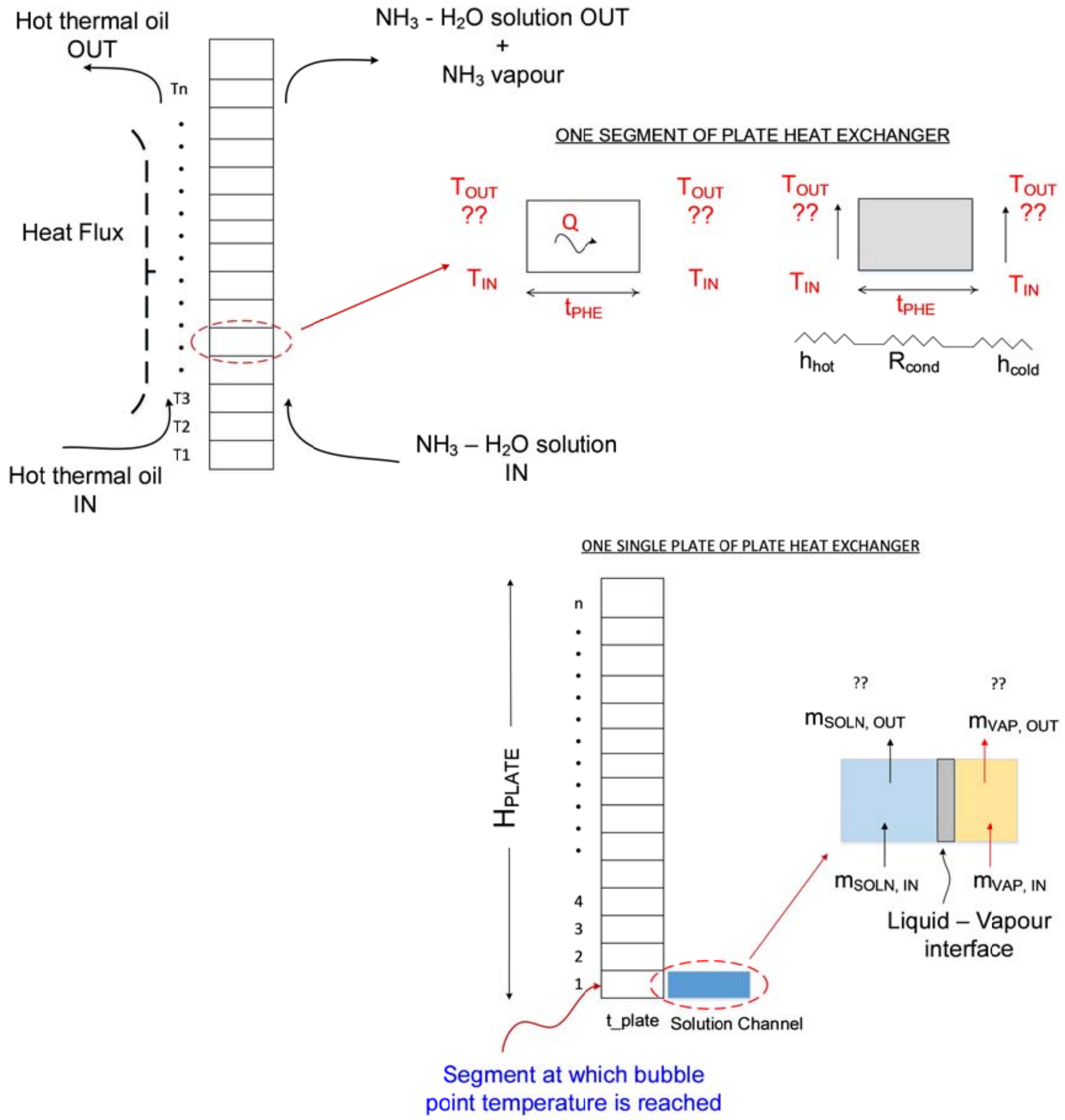

Figure 6: Segmental approach for modelling the plate heat exchanger

Equations employed

For temperature distribution:

The equations employed for each segment, to determine the temperature distribution along the plate are given in [6] to [8]

$$\mathrm{Q_{seg}} = \mathrm{m_{oil,ch}} * \ \mathrm{C_{p,Oil}} * (\mathrm{T1} - \mathrm{T2}) \tag{6}$$

$$\mathrm{Q_{seg}} = \mathrm{m_{soln,ch}} * \ \mathrm{C_{p,soln}} * (\mathrm{T4} - \mathrm{T3}) \tag{7}$$

$$\mathrm{Q_{seg}} = \mathrm{U_{seg}} * \mathrm{A_{seg}} * \mathrm{LMTD_{seg}} \tag{8}$$

The overall heat transfer coefficient for each segment is calculated from Equation [9]

$$\frac{1}{\mathrm{U}} = \frac{1}{h_{oil}} + \frac{t}{k} + \frac{1}{h_{soln}} \tag{9}$$

Where,

$\mathrm{Q_{seg}}$ = Heat transferred per segment (1),
$\mathrm{m_{oil,ch}}$= mass flow rate of oil in the channel,
$\mathrm{m_{soln,ch}}$ = mass flow rate of solution in the channel,
$\mathrm{C_{p,Oil}}$ = specific heat capacity of oil (J/kg K),
$\mathrm{C_{p,soln}}$ = specific heat capacity of solution (J/kg K),
$\mathrm{A_{seg}}$ = Heat transfer area of each segment (m^2),

For the thermal oil, only the single phase heat transfer coefficient was used whereas for the NH_3-H_2O solution both single phase and two phase heat transfer coefficient was used, depending on when the flow changed to two phase.

$$\mathrm{h_{SP,i}} = \ (\mathrm{Nu_i} * \ \mathrm{k_i})/\mathrm{D_h} \tag{10}$$

Where,

i = thermal oil or NH_3-H_2O solution

$\mathrm{h_{SP,i}}$ = single phase heat transfer coefficient (W/m^2K),
$\mathrm{Nu_i}$ = Local Nusselt number,
$\mathrm{k_i}$ = thermal conductivity of fluid (W/m^2K),
$\mathrm{D_h}$ = hydraulic diameter of channel (1)

There are numerous correlations available in literature for calculating the heat transfer coefficient in a plate heat exchanger however the most widely suggested correlation is that of Muley & Manglik (3) and this is used in the present modelling studies.

$$\mathrm{Nu} = 0.44 * \left(\frac{\beta}{30}\right)^{0.38} * \ \mathrm{Re}^{0.5} * \ \mathrm{Pr}^{\left(\frac{1}{3}\right)} * \ \left(\frac{\mu}{\mu_w}\right)^{0.14} \tag{11}$$

The above equation is valid for $30^o \leq \beta \leq 60^o$ and $30 \leq \mathrm{Re} \leq 400$

$$Nu = \left[(0.2668 - 0.006967 * \beta + 7.244 * 10^{-5} * \beta^2) * (20.7803 - 50.9372 * \phi + 41.1585 * \phi^2 - 10.1507 * \phi^3) * Re^p * Pr^{\frac{1}{3}} * \left(\frac{\mu}{\mu_w}\right)^{0.14}\right] \quad [12]$$

The exponent of Re, p is given by

$$p = 0.728 + 0.0543\ \sin\left(\frac{2 * \pi * \beta}{90} + 3.7\right) \quad [13]$$

The above equation is valid for $30^o \leq \beta \leq 60^o$ and $Re \geq 1000$

In equations [11] and [12], the different variables mentioned are as follows:

β = corrugation angle of the plate wrt vertical,
μ = temperature dependent dynamic viscosity of the fluid (Pa.s),
μ_w = temperature dependent dynamic viscosity of the fluid at the wall (Pa.s),
ϕ = Area enlargement factor, which takes into account the amplitude of the corrugation and the pitch of corrugation.

For the two phase heat transfer coefficient, the correlation suggested by Taboas et al (4) is used. This correlation is used because it was arrived at after relating it with experiments carried out on NH_3-H_2O desorption in PHE by the same research group.

$$h_{TP} = 5 * Bo^{0.15} * h_{SP} \quad [14]$$

Bo is the boiling number which is incorporated to take care of two phase flow and is given by

$$Bo = q / (\dot{m}_{soln} * H_{fg}) \quad [15]$$

Where,

q = heat flux (W/m^2), $\dot{m}_{soln}$ = solution mass flux (kg/m^{2}s), H_{fg} = enthalpy of vapourization (J/kg)

The above relations for Nusselt number ensure that the fluid flow is coupled to the geometric dimensions of the plate heat exchanger.

For mass flow & vapour distribution:

Once the temperature profile of both the fluids along the plate is calculated the mass flow rate of the weak solution along with the vapour can be calculated. The equations employed are as follows:

$$m_{SOLN,IN} + m_{VAP,IN} = m_{SOLN,OUT} + m_{VAP,OUT} \quad [16]$$

$$m_{SOLN,IN} * \chi_{SOLN,IN} + m_{VAP,IN} * \chi_{VAP,IN} = m_{SOLN,OUT} * \chi_{SOLN,OUT} + m_{VAP,OUT} * \chi_{VAP,OUT} \quad [17]$$

Where,

$m_{SOLN,IN}$ = mass flow rate of the solution into the segment (kg/s)
$m_{SOLN,OUT}$ = mass flow rate of the solution from the segment (kg/s)
$m_{VAP,IN}$ = mass flow rate of vapour into the segment (kg/s)
$m_{VAP,OUT}$ = mass flow rate of vapour out of the segment (kg/s)
$\chi_{SOLN,IN}$ = NH_3 mass fraction of solution into the segment
$\chi_{SOLN,OUT}$ = NH_3 mass fraction of solution from the segment
$\chi_{VAP,IN}$ = NH_3 mass fraction of vapour into the segment
$\chi_{VAP,OUT}$ = NH_3 mass fraction of vapour from the segment

Initial PHE geometry

Since only four research groups have worked on the experimental aspects of PHE desorber (5-8), the geometry from one of the groups (7) is selected for initial modelling. The dimensions of the selected geometry are given in Table II.

Table II: Dimensions of PHE desorber used in modelling

Parameters	**Dimension**
Height of plate	519 mm
Width of plate	175 mm
Thickness of plate	0.4 mm
Hydraulic diameter	3.9 mm
Corrugation type	Chevron
Corrugation angle	58.5^{o} from vertical
Amplitude of corrugation	2.4 mm
Pitch of corrugation	9.85 mm
Surface area of plate (H x W)	0.091 m^2
Number of plates	20

Results & Discussion

Heating of thermal oil in heat exchanger

Steady state simulations in the heat exchanger revealed that the thermal oil gets heated by 32 K in one pass through the heat exchanger. This temperature rise is sufficient enough because this is precisely the amount of temperature drop that the oil goes through in the PHE desorber as will be discussed in the next section.

Figure 7 shows the meshed geometry of the heat exchanger. As can be seen the meshing is quite dense around the critical areas (air flow path & oil flow path) and hence gives accurate results. Figure 8 shows a graphical representation of the heating of the thermal oil in the heat exchanger and Figure 9 shows the variation of average outlet temperature of the thermal oil with mass flow rate of thermal oil.

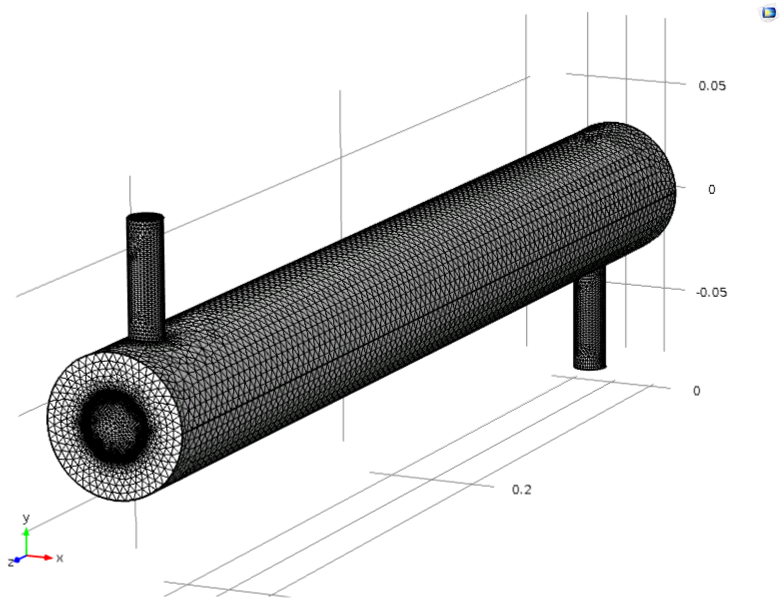

Figure 7: Meshed geometry of the tube in tube heat exchanger

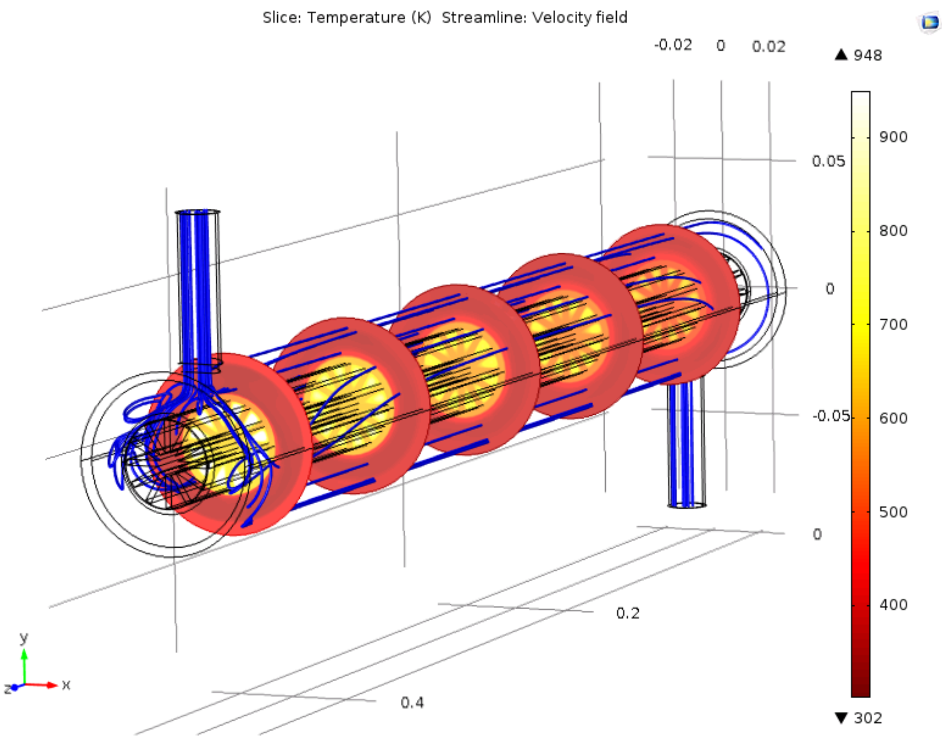

Figure 8: Graphical representation of heating of thermal oil & fluid flow

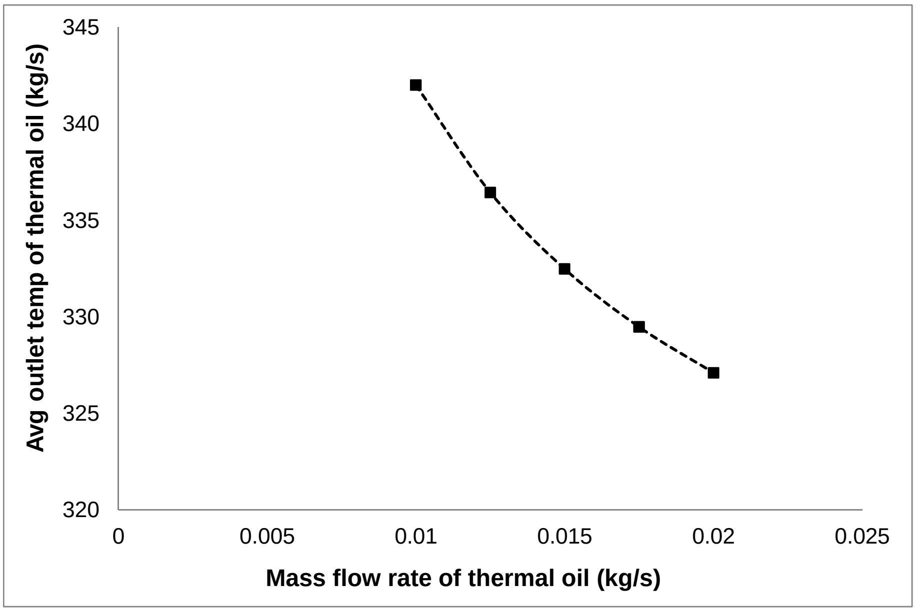

Figure 9: Variation of average outlet temperature of thermal oil with mass flow rate

The main idea is to use a simple heat exchanger so as not to make the system complicated but at the same time have the functionality and performance required of it. Adding fins to on the inside of the heat exchanger improves the heat transfer.

<u>Mass flow rate & quality of vapour from PHE desorber</u>

The temperature variation along the height of the plate is shown in Figure 10. The NH_3-H_2O solution enters the PHE desorber at a temperature of 110^oC and reaches the boiling point within a short distance (as indicated in the Figure 10). This is the point where the solution reaches bubble point temperature and the first vapour bubble starts to form. Hence till this point, the heat transfer from the thermal oil to the solution is categorized as sensible heat transfer and beyond this point it is categorized as latent heat transfer. As seen from Figure 10, the thermal oil loses about 40 K when transferring heat to the NH_3-H_2O solution.

Figures 11 & 12 show the heat flux and heat transfer coefficient variation along the height of the plate. The overall heat transfer coefficient is dominated by the heat transfer coefficient of the thermal oil.

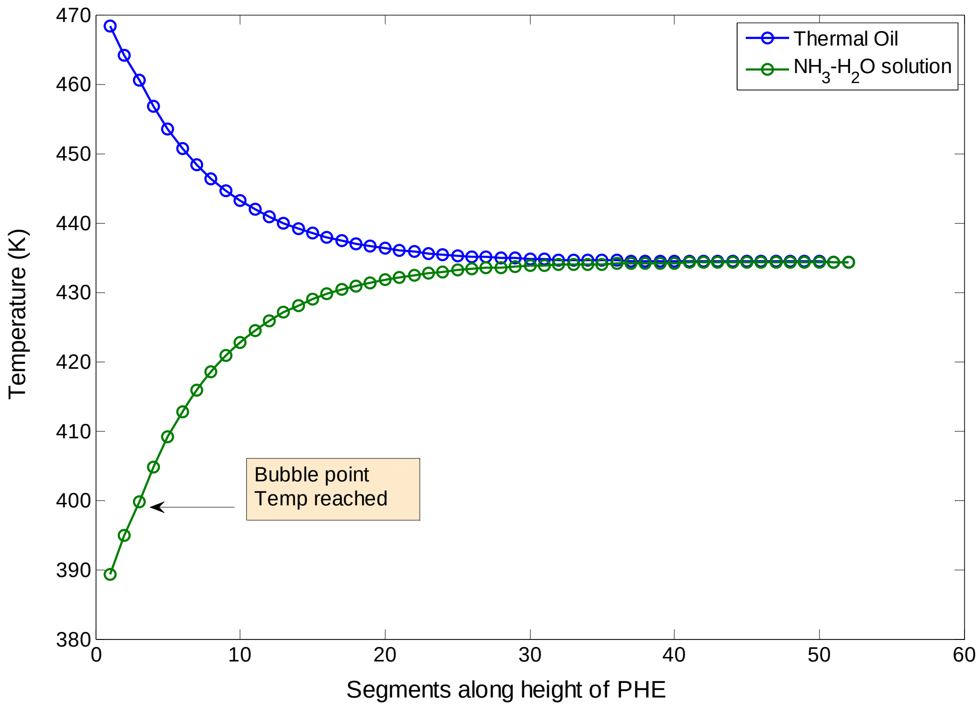

Figure 10: Temperature profile along height of plate

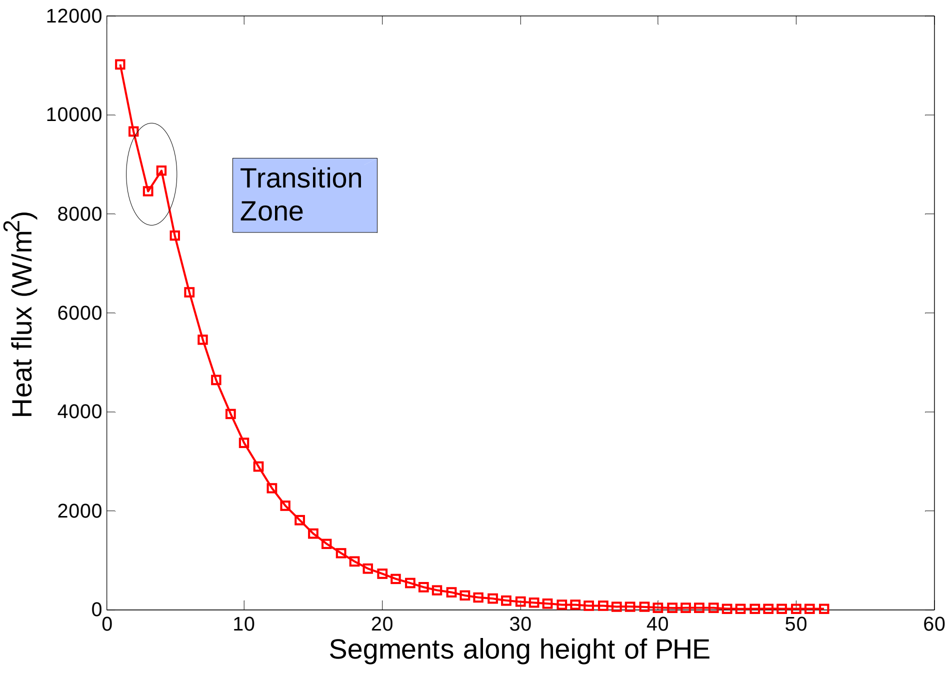

Figure 11: Heat flux profile along the height of the plate

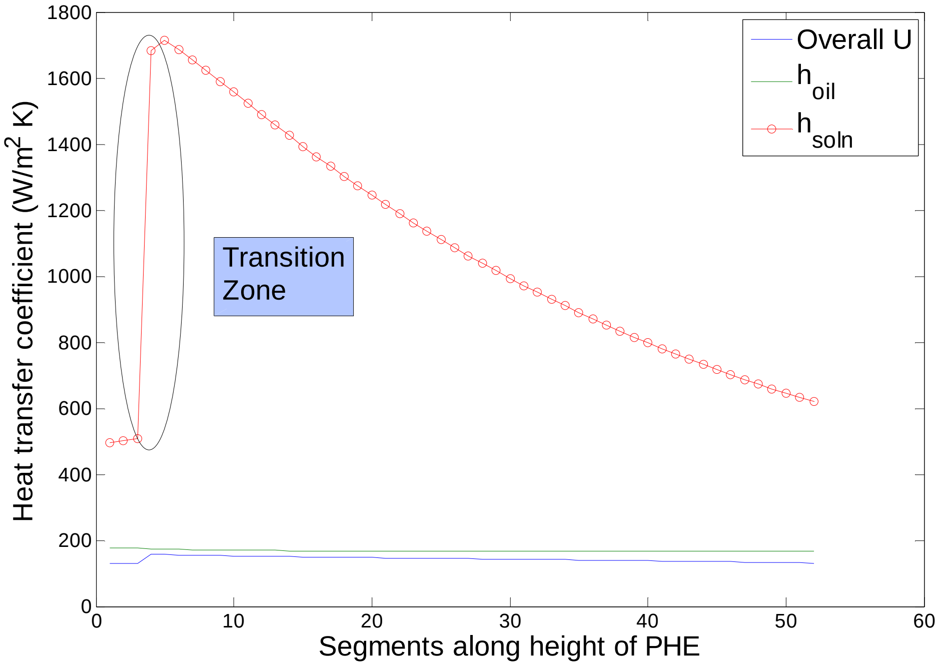

Figure 12: Heat transfer coefficient along height of plate

Figures 13 & 14 show the concentration and mass flow rate from one of the channels in the PHE desorber. The amount of vapour generated from each channel is about 0.3 g/s, leading to a total refrigerant production of 2.7 g/s from the PHE desorber of dimensions given in Table II. The amount of vapour needed for a 1 kW cooling load is about 1 g/s as found from the '0D' model performed on the VARS [Ref, own modelling studies]. Hence the PHE desorber is able to produce sufficient quantity of vapour.

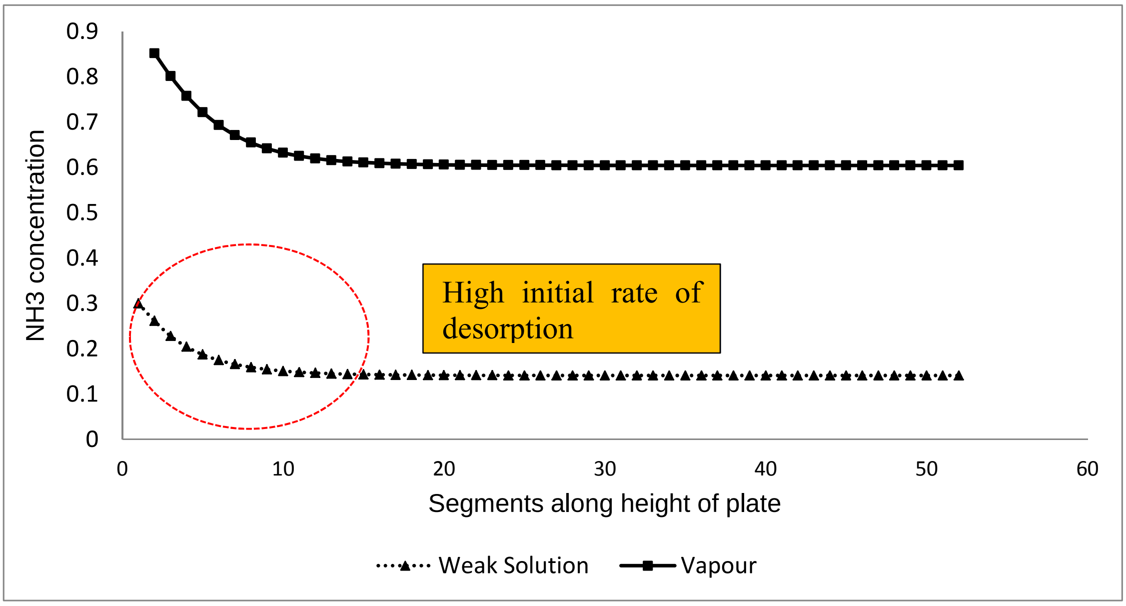

Figure 13: Concentration profile of vapour and solution along the height of the plate

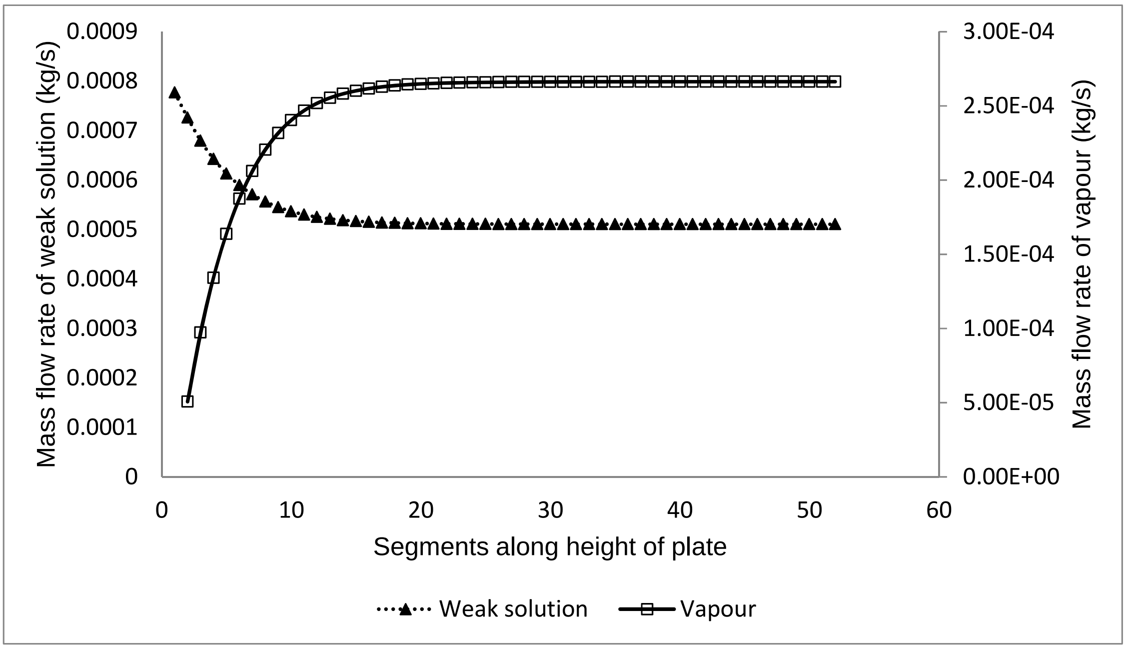

Figure 14: Mass flow rate of weak solution and vapour along height of plate

Sensitivity analysis to assess PHE desorber performance

There are a number of factors that could affect the performance of the PHE desorber and it is vital to capture and study these effects. Performing a sensitivity analysis helps in optimizing the dimensions of the PHE desorber. Table III categorizes the effect that different parameters have on the mass flow rate and NH_3 concentration generated from a PHE desorber.

Table III: Effect of different parameters on PHE desorber performance

	Strong effect	**Medium effect**	**No effect**
Number of plates	✓		
Height of plate		✓	
Width of plate			✓
Inlet temperature of oil	✓		
Corrugation angle of plate		✓	
Pitch of corrugation		✓	
Mass flow rate of strong solution	✓		
Mass flow rate of thermal oil	✓		

As seen from Table III, the number of plates used in the heat exchanger has a profound effect on the mass flow rate and NH_3 concentration besides the operating parameters of the respective fluids.. The effect is shown in Figure 15. The optimum number of plates is around 10 after which there is no drastic effect both on vapour generation and NH_3 concentration.

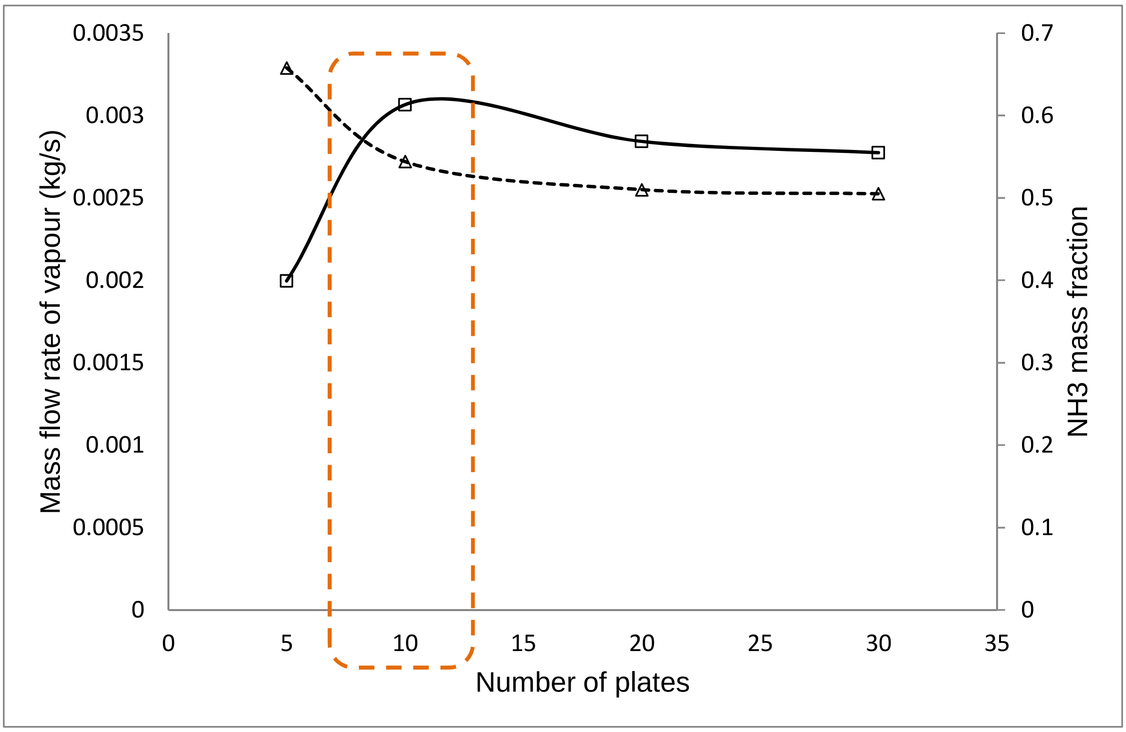

Figure 15: Variation of total mass flow rate and NH_3 mass fraction of vapour with number of plates

Sizing of plate heat exchanger

Based on the above results the plate heat exchanger can be sized to meet the required cooling load of 1 kW. The dimensions of the PHE desorber to suit a 1 kW cooling load are outlined in Table IV.

Table IV: Dimension of PHE desorber to suit 1 kW cooling load

Number of plates	10
Height of plate	300 mm
Width of plate	50mm
Corrugation angle	60^o
Pitch of corrugation	5 mm

The other dimensions of the PHE can be the same as the one mentioned in Table II.

Design maps for different cooling loads

Figures 16, 17 & 18 show the design maps for using SOFC stacks on board refrigerated trucks and the corresponding mass flow rate of thermal oil needed at the PHE desorber.

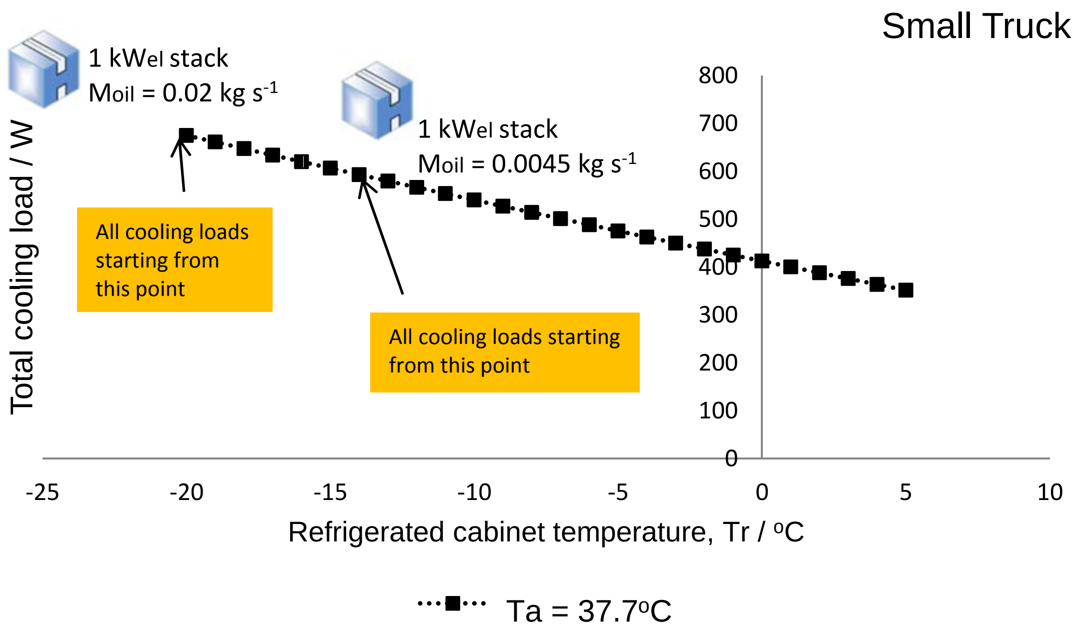

Figure 16: Design maps for a small refrigerated truck

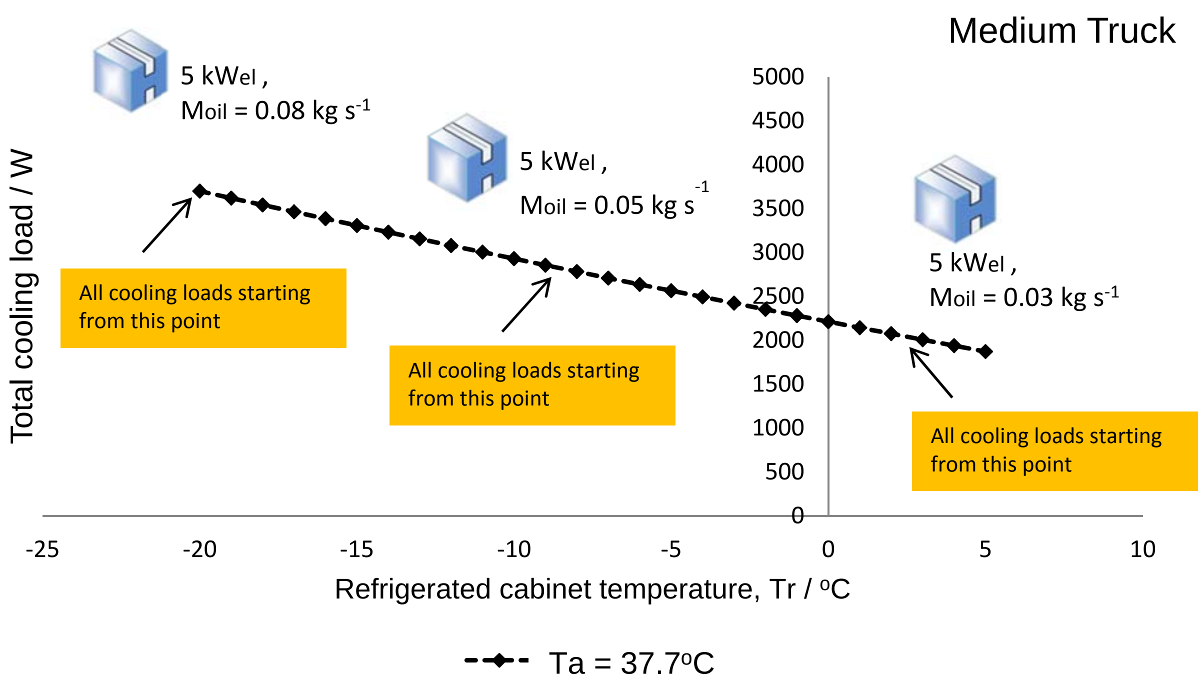

Figure 17: Design maps for a medium refrigerated truck

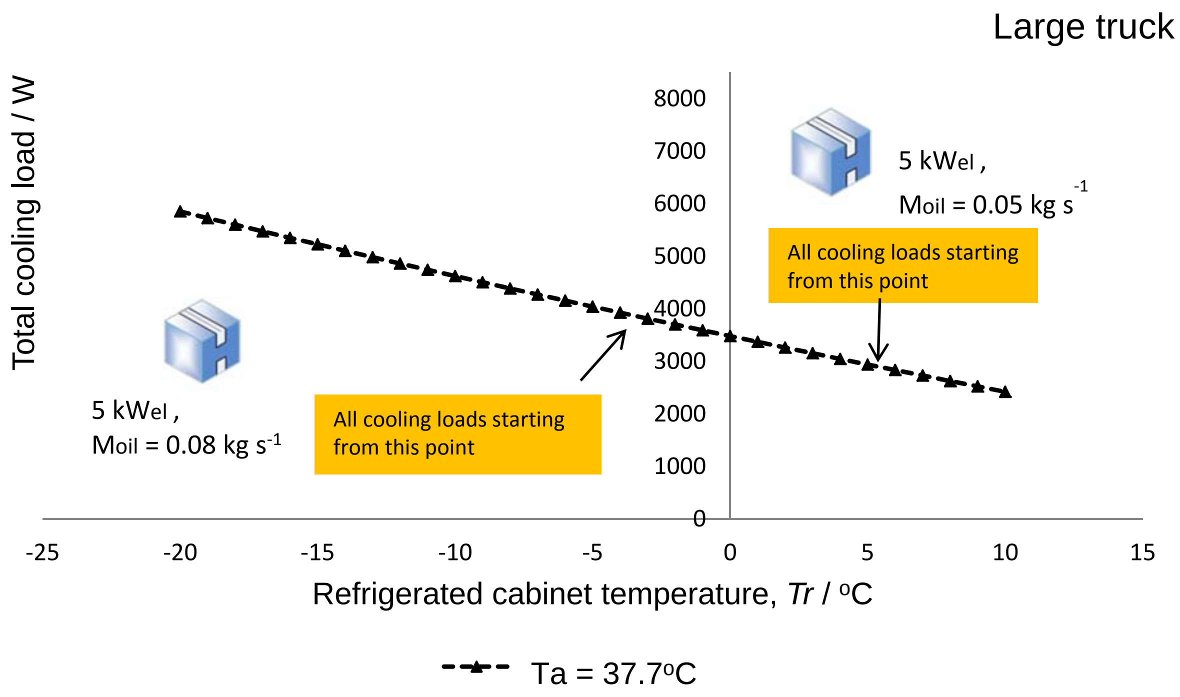

Figure 18: Design maps for large refrigerated truck

Potential Application areas

Besides the refrigerated transportation sector such a system can also find application in the residential air conditioning sector. With increased penetration of SOFC in the residential market, such a small scale VARS unit can potentially replace the conventional air conditioners, especially in hot countries where air conditioners powered by electricity run almost all the while. This will result is tremendous energy savings as this system runs on free thermal energy available from the SOFC. Such a system also provides opportunities to use solar thermal technology for integration where an SOFC cannot be used.

Conclusions & Future outlook

Complete system modelling of the SOFC stack, tube in tube heat exhcanger with internal fins and the PHE desorber were carried out. The sizing of the components is very critical since the end application is for refrigerated transport. The heat exchanger was designed and optimized using COMSOL multi-physics and steady state simulations performed to evaluate the heating of the thermal oil in it. The thermal oil gets heated to a higher temperature when the mass flow rate of thermal oil is reduce due to increased residence time of the thermal oil in the heat exchanger. For a 1 kW cooling load the mass flow rate of thermal oil needed at the desorber is around 0.02 kg/s and with this flow rate the thermal oil gets heated by 32 K for one pass through the heat exchanger. This is precisely the temperature drop that the oil goes through when passing through the PHE desorber. Hence during steady state operation the whole system will work perfectly because the amount of heat gained at one place is equal to the amount of heat lost at another.

The PHE desorber was also sized to meet a 1 kW cooling load requirement and was found to be very compact, being just 300 mm in length and 50 mm in width.. Using plate heat exchangers as desorbers allows one to develop compact VARS units which can in turn compete with VC system in places where free thermal energy is available.

The next stage of the work focuses on constructing a test rig, which is currently in progress at the University of Birmingham, to demonstrate the proof of concept.

Initial starting of the whole system can be quite challenging but a strategy is outlined to aid in faster starting of the whole system. The tube in tube heat exchanger can initially be coupled to the hot air that is used to warm up the SOFC stack. The SOFC stack takes about 90 minutes to warm up. This will lead to simultaneous heating of thermal oil when the SOFC is getting warmed. Once the SOFC stack is ready for operation the tube in tube heat exchanger can be connected to the cathode exhaust line of the SOFC stack. The schematic for the same is shown in Figure 19.

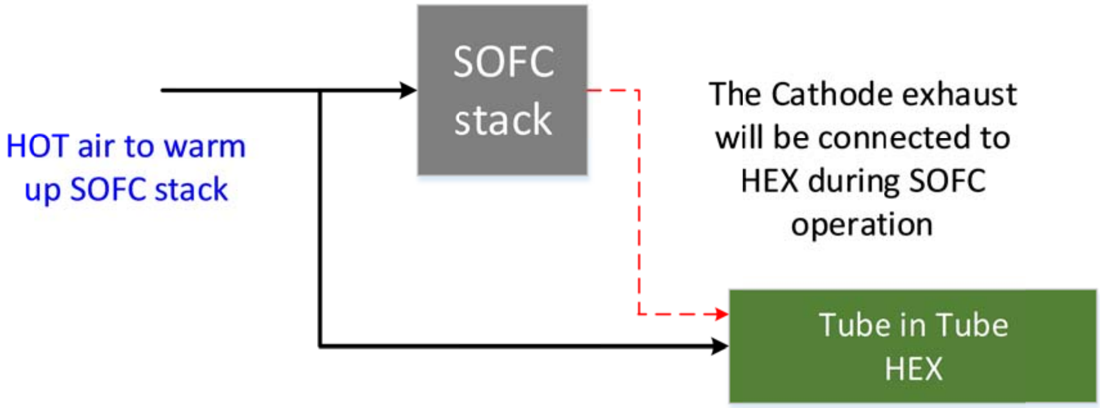

Figure 19: Schematic for faster operation of whole system

Acknowledgements

The authors would like to thank Unilever for their financial support in this project and also the EPSRC UK.

References

1. Venkataraman, V., A.W. Pacek, and R. Steinberger-Wilckens, *Coupling of SOFC and Vapour Absorption Refrigeration System (VARS) for truck applications*, in *11th European SOFC and SOE Forum 2014*. 2014: Lucerne, Switzerland.
2. http://www.paratherm.com/heat-transfer-fluids/.
3. Ayub, Z.H., *Plate heat exchanger literature survey and new heat transfer and pressure drop correlations for refrigerant evaporators.* Heat Transfer Engineering, 2003. **24**(5): p. 3-16.

4. Táboas, F., et al., *Assessment of boiling heat transfer and pressure drop correlations of ammonia/water mixture in a plate heat exchanger*. International Journal of Refrigeration, 2012. **35**(3): p. 633-644.
5. Táboas, F., et al., *Flow boiling heat transfer of ammonia/water mixture in a plate heat exchanger*. International Journal of Refrigeration, 2010. **33**(4): p. 695-705.
6. Balamurugan, P. and A. Mani, *Heat and mass transfer studies on compact generator of R134a/DMF vapour absorption refrigeration system.* International Journal of Refrigeration, 2012. **35**(3): p. 506-517.
7. Zacarías, A., et al., *Boiling heat transfer and pressure drop of ammonia-lithium nitrate solution in a plate generator.* International Journal of Heat and Mass Transfer, 2010. **53**(21–22): p. 4768-4779.
8. Marcos, J.D., et al., *Experimental boiling heat transfer coefficients in the high temperature generator of a double effect absorption machine for the lithium bromide/water mixture.* International Journal of Refrigeration, 2009. **32**(4): p. 627-637.