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A NUMERICAL STUDY OF ACTIVE AND PASSIVE HEAT TRANSFER ENHANCEMENT IN LATENT HEAT THERMAL ENERGY STORAGE DEVICES

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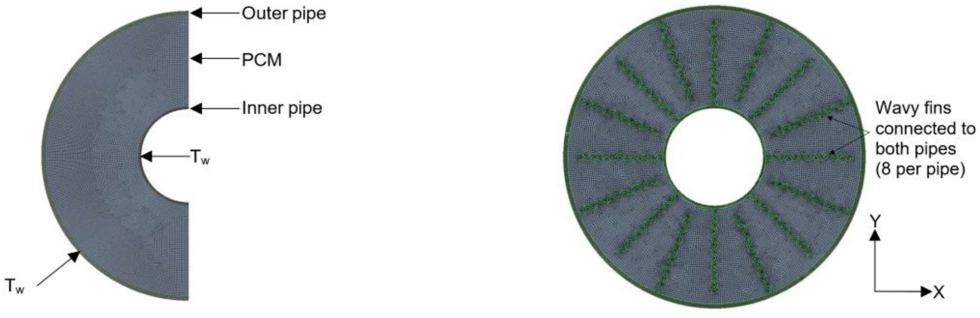
1. ABSTRACT

Latent Heat Thermal Energy Storage (LHTES) devices implement phase-change materials (PCM) to store and release the thermal energy from the latent heat of fusion of a material. PCMs have high energy densities and can be effective at storing a large amount of thermal energy. This is especially useful when used in conjunction with renewable energy sources to balance energy production with demand. One of the disadvantages of PCMs, however, is their poor thermal conductivity, inhibiting the LHTES device's melting and solidification performance. This numerical study aims to research both active and passive heat transfer methods to improve the melting performance of a PCM, making them more effective when used in thermal energy storage.

2. COMPUTATIONAL METHODOLOGY

The computational domain (Figure 1) derived from Zhao et al. [1], was a 2-dimensional (2D) cross-section of a triplex-tube heat exchanger (TTHX). An annulus filled with RT-82 Rubitherm GmbH PCM was surrounded by two concentric, isothermally heated, copper pipes which would contain a heat transfer fluid. The inner pipe was 1.2mm thick with a maximum radius of 25.4mm and the outer pipe was 2mm thick with a maximum radius of 75mm. The temperature of the walls was 363.15K, whereas the initial temperature of the PCM was 300.15K [1]. For active heat transfer methods, the domain was filled completely with PCM, and the wall temperatures fluctuated sinusoidally between 353.15K and 373.15K at 0.05Hz, 0.1Hz and 0.2Hz; however, still having the same average heat transfer rate throughout.



a. Base model   b. Wavy fin model

Figure 1: 2D cross-section of a TTHX with isothermally heated pipes. **a.** The base model without heat transfer enhancement. **b.** a model enhanced with a wavy fin.

For passive methods, the PCM domain contained fins which acted as a heat transfer medium. Initially, a wavy fin was designed to imitate a 2D cross-section of a porous metal foam. The wave had a 3.33mm wavelength and a 1.11mm amplitude. The total fins length was 42mm and had a thickness of 1mm. This design was based on Attarzadeh et al.'s study [2]. The wavy fins were then compared against straight fins of the same length and thickness. The straight fin's surface area was substantially less than the wavy fin of same length, hence, a model containing shorter wavy fins of equal surface area to the straight fins was simulated.

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The transient flow and thermal fields in the PCM were modelled in ANSYS FLUENT by solving mass, momentum, and energy equations based on the finite volume method. The fluid flow is laminar, incompressible, and Newtonian. The Boussinesq approximation was applied to simulate buoyancy-driven natural convection in the PCM domain. To simulate the phase change process, the enthalpy-porosity approach was adopted. This approach assumes phase change to occur over a finite temperature range; thus, generating an artificial mushy region in which the melt fraction of a fluid element varies from zero (solid phase) to 1 (liquid phase). Naturally, the velocity of the fluid element within the mushy region should also vary from zero (solid phase) to the natural convection velocity (liquid phase). The enthalpy-porosity formulation deals with this velocity transition by modelling flow within the mushy region as flow through a porous medium. A sink term, in the form of the Carman-Koseny equation, is added to the Navier-Stokes equations to mimic the effect of damping within the mushy region.

The second order upwind scheme was used for discretisation of the convective terms and central difference schemes for diffusive and conductive terms. The SIMPLE method was implemented for pressure-velocity coupling and the PRESTO scheme for pressure interpolation. The base model was run symmetrically along the y-axis, containing ~20,000 unstructured mesh shown in Fig.1. It was necessary that the simulation captured melting over a time-period and, therefore, was under transient conditions. The transient time-step was chosen as 0.5s having been influenced by Zhao et al.'s [1] sensitivity study, which showed negligible impact in the liquid fraction of the domain between 0.1s, 0.3s and 0.5s.

3. RESULTS AND DISCUSSION

The melting times of the PCM with and without the oscillating temperature profiles are shown in Figure 2. The difference is only small and the PCM does not reach 100% liquid fraction in any scenario, but it is apparent that, with a greater oscillating frequency, there is some improvement in the melting performance of the PCM. After 3000s of flow time, the 0.2Hz scenario had an 8.629% higher liquid fraction than the constant wall temperature scenario.

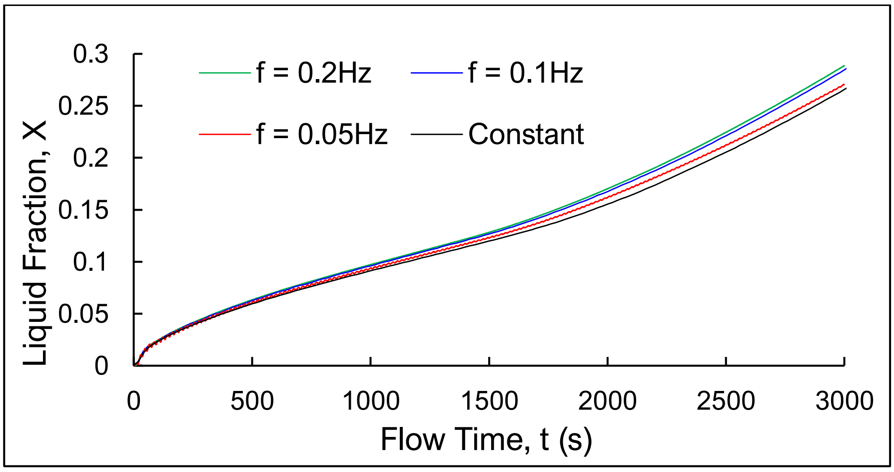


Figure 2: A plot comparing the melting performance of the constant wall temperature scenario with three scenarios with different oscillating wall temperature frequencies.

The contours in figure 3 highlight the subtle differences between the constant and oscillating scenarios. The liquid fraction contour for the oscillating temperature shows a wider mushroom pattern where the PCM has melted and a circulating convection pattern has formed. The velocity contours support this showing higher velocities for the oscillating temperature, meaning an increase in convection rate.

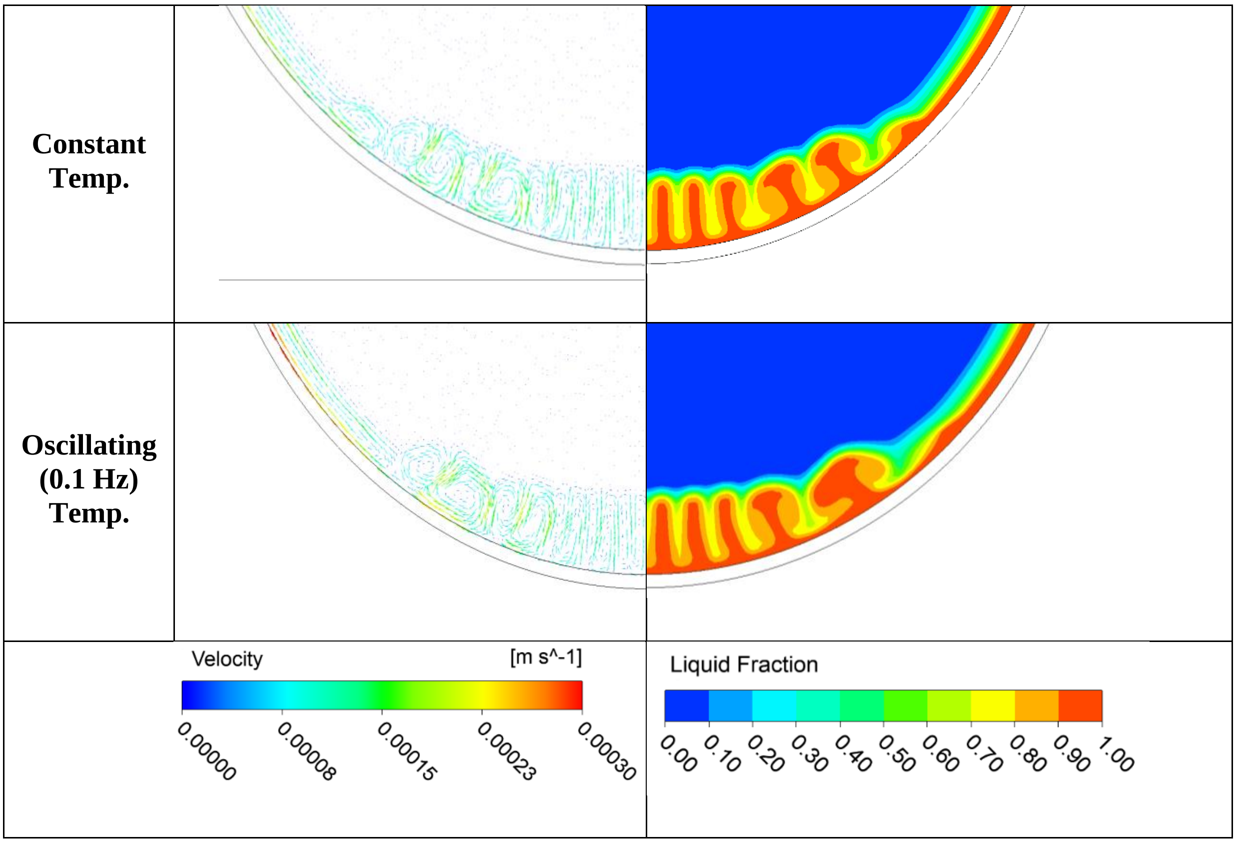
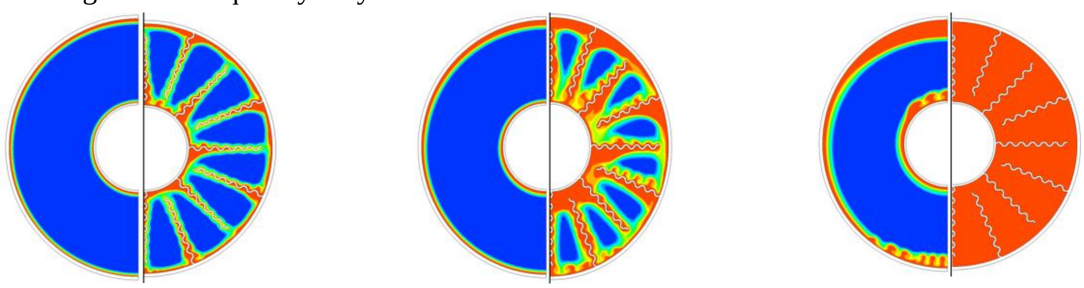


Figure 3: Comparison of the Velocity (left) and Liquid Fraction (right) contours for constant and oscillating wall temperatures.

For the passive technique, three fin designs were simulated: a long-length waved fin, a long-length straight fin, and a short-length waved fin. The long length wavy fin was simulated first and it was evident that using fins as a heat transfer medium significantly reduced the PCM melting time. Figure 4 shows liquid fraction contours of the base versus long-length wavy fin simulation. The finned simulation melted fully in 2350s, whereas, at this point, the base simulation had a liquid fraction of roughly 19%.

The long-length straight fin was then simulated to compare whether the fin's shape had an impact on the melting performance. The melting time was also drastically shorter than the base scenario, however, took 92s longer to reach full liquid fraction compared to the wavy fins. The wavy shape was expected to have a better performance over straight fins. Although the length was the same, the surface contact area against the fluid was 26% greater, so one would expect a greater melting performance. It is important to emphasize that, even though the surface area of the long wavy fin was 26% greater, the PCM melting was consequently only 3.915% faster.



t= 500 s t= 1000 s t= 2500 s

Figure 4: Liquid fraction contours at 500s, 1000s, and 2500s of flow time comparing the base design (left) against the long-length waved fin design (right).

The shorter waved fin was designed to have a surface contact area equal to the straight fin in order to draw conclusions between the effects of the fin's shape, surface and length. When simulated, the total melting time of the short fin scenario was 2832s. The difference in melting time is shown in figure 5 - a plot of the three finned scenarios. Initially, the three curves follow the same trend but eventually they diverge from each other between 350s and 650s.

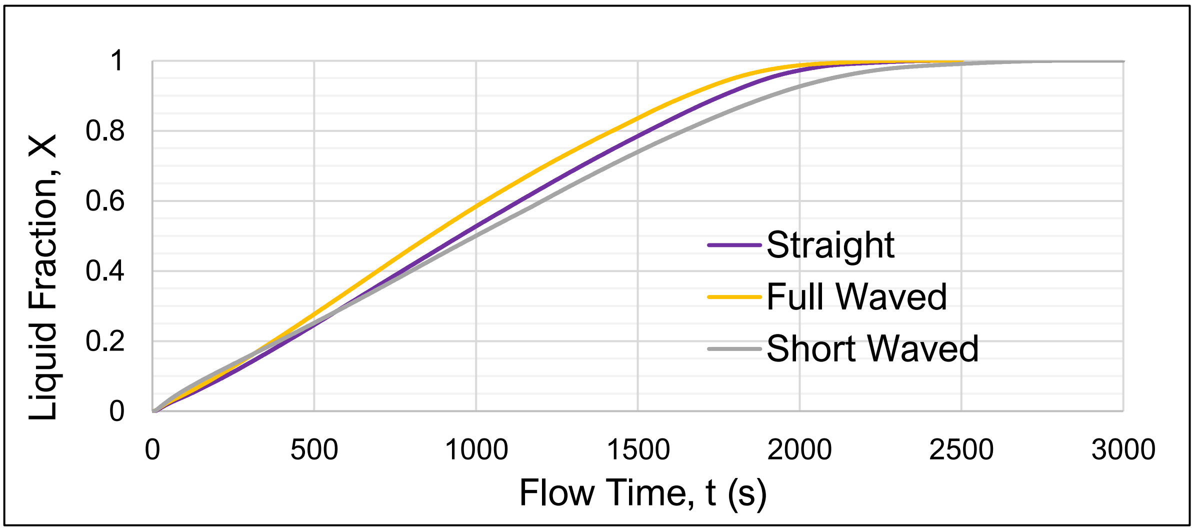


Figure 5: Comparison of the melting performance of all three fin types, where liquid fraction is plotted against flow time.

The Nusselt number plots below give insight into why the longer waved fins melted the PCM faster. From the beginning, there is a higher average Nusselt number across both the inner walls (figure 6) and outer walls (figure 7) for the longer length fin domains. Therefore, it's clear there is a greater convective heat transfer presence in these situations, leading to a faster melting time throughout the domain. As the shorter finned domain appears to have a lower average Nusselt number, the heat transfer mechanism is dominated more by conduction than the other two scenarios, which in a PCM, is not as effective due to its lower thermal conductivity.

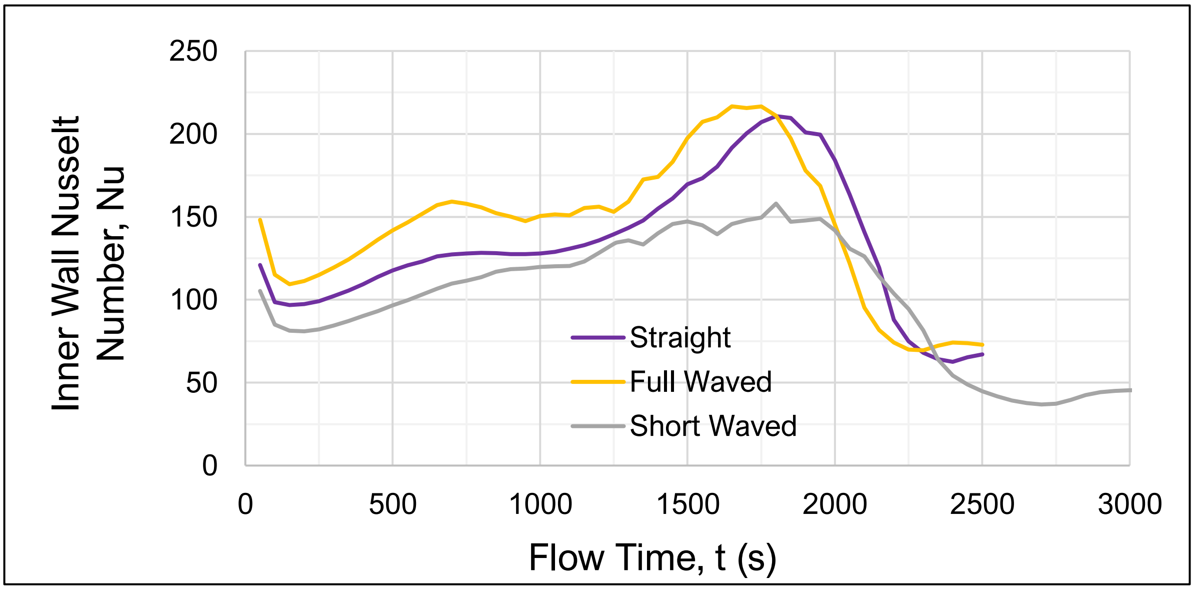


Figure 6: Inner wall average Nusselt numbers for the three different fin configurations plotted against flow time.

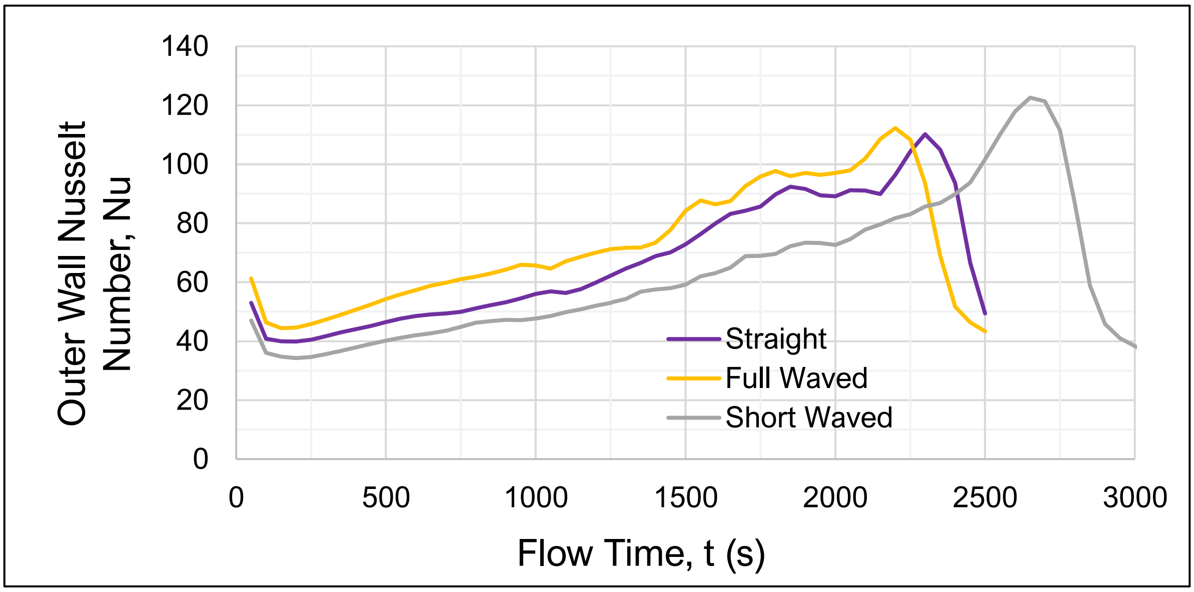


Figure 7: Outer wall average Nusselt numbers for the three different fin configurations plotted against flow time.

The Nusselt number plots peak at different stages in the melting process. The inner wall Nusselt numbers peak much earlier than the outer wall. The inner wall tends to peak as the upper region contains almost all liquid PCM and convective forces will be at their highest before dropping off. The outer wall is in contact with the bottom of the domain where the PCM will remain partially melted until the end of the process - mainly due to the hotter PCM rising to the upper region of the domain caused by the buoyancy forces as the PCM changes phase into a liquid. The outer wall therefore peaks much later and suggest before the liquid fraction reaches 100%.

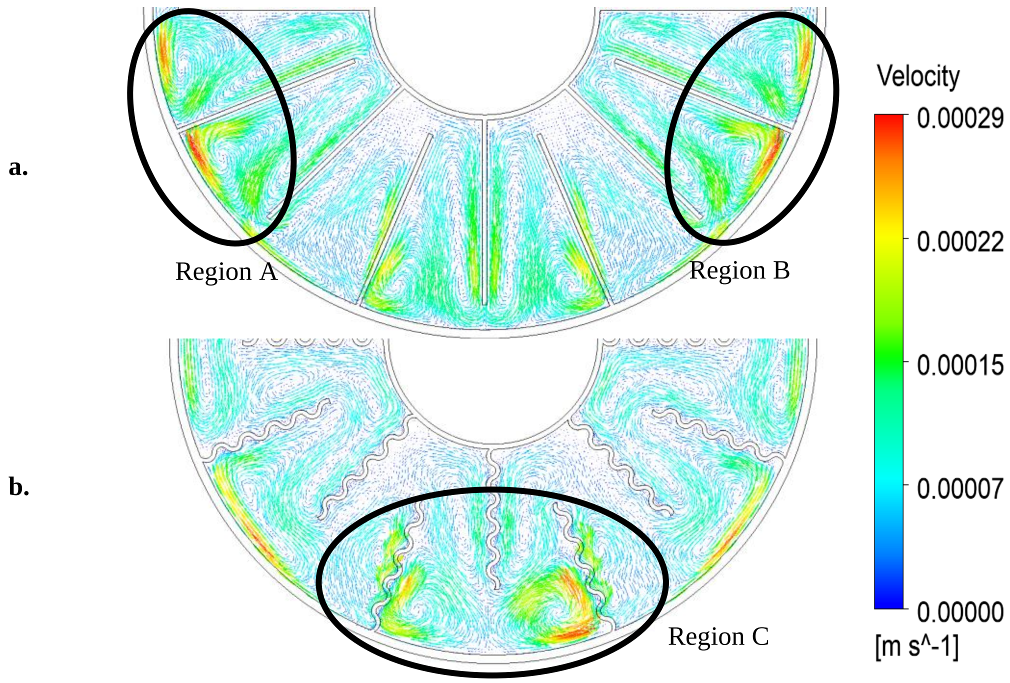


Figure 8: Velocity contours for the (a) short straight fins and the (b) short waved fins at their peak outer wall Nusselt numbers.

We can deduce that having a longer fin is beneficial when using passive heat transfer media, especially in the earlier stages of the melting process. The outer wall plot does highlight a significant peak in the Nusselt number in the short-waved scenario. Figure 8 show velocity contours of the straight fins and the short wavy fins at their peak Nusselt numbers. For straight fins, this is at 2300s, and for the waved fins, at 2700s. The fluid velocity is greater in the straight fins at regions A and B where the fluids have been contained into 'cells' due to the longer fin reaching further across the domain. As for the short waves, the fluid velocity is at its highest at the bottom of the domain. Region A is an open area which allows for greater movement of fluid and effective convective heat transfer - hence the greater peak Nusselt number here. Although there is a spike in Nusselt number, the performance of the short wavy fin was lower than the others.

Figure 9 compiles the liquid fraction against flow time results from all simulations. This includes an optimised design which is a combination of the long-length waved fins with a 0.2Hz frequency oscillating temperature profile. The optimised design outperforms all scenarios, achieving full liquid fraction in 2217.5s.

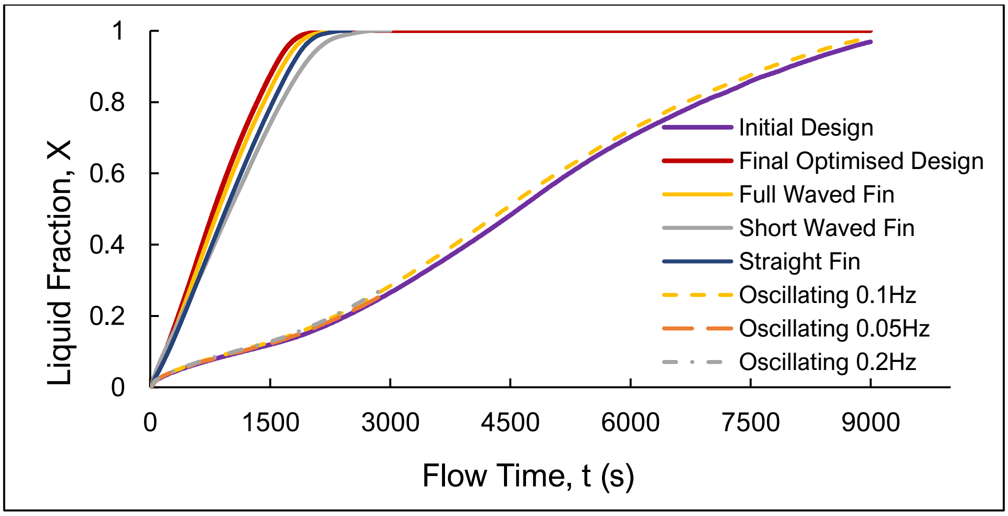


Figure 9: A plot of liquid fraction against flow time, comparing all active and passive scenarios with the base model, including the optimised design.

4. CONCLUSIONS

A numerical study was conducted into the effects of passive and active heat transfer techniques on the enhancement of a LHTES device. The active method showed only minor improvements in the melting rate of the PCM, but a positive relationship was deduced between the greater oscillatory frequency and an increase in the melting rate. For the passive methods, there was not a large difference in melting rate when comparing the different fin shapes. However, all fin shapes showed a significant improvement in the melting rate of the PCM – resulting in full melting of the fluid domain in 2350s. It was clear that the longer fins melted the PCM faster than the shorter fin. The active and passive methods were then combined into an optimised simulation, using the long, waved fins with a 0.2Hz oscillating temperature profile. This resulted in an 80% reduction in the PCM melting time, when compared with the base model.

REFERENCES

[1] Zhao, C., et al. (2020). 'Numerical study of melting performance enhancement for PCM in an annular enclosure with internal-external fins and metal foams', International Journal of Heat and Mass Transfer, 150.

[2] Attarzadeh, R., Rovira, N. and Duwig, C. (2021). 'Design analysis of the "Schwartz D" based heat exchanger: A numerical study', International Journal of Heat and Mass Transfer, 177.