# POTENTIAL IMPROVEMENT IN THE DESIGN OF IMMERSED COIL HEAT EXCHANGERS

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## Abstract

Although there is a notable move towards fresh water stations in the European markets, using the cheaper internal solutions of smooth or finned coil heat exchangers for smaller solar thermal system storage devices still dominates. Varying significantly in type and geometry, the functions used to account for heat transfer are possibly outdated and in light of the still relatively high investment costs of solar thermal systems, not cost optimizing.

An overview of the current thermal storage market implementing typical immersed heat exchanger types and variations has been collated such that geometrical dependencies and specific advantages or disadvantages can be elucidated. By analysing the correlation between parameters, the relationship between trends in design and current *rules of thumb* is discussed.

From the market research a reference solar DHW was defined and simulated in the TRNSYS simulation environment. A method for interpreting the results in dependency of the energy transferred through an immersed coil heat exchanger at any given annual condition is presented and a selection of representative immersed coil heat exchangers is discussed for future experimentation based on the knowledge gained from the simulation results. Based on commensuration of the experiments with computational modeling (computational fluid dynamics as well as TRNSYS models), future work will entail the development of an immersed coil heat exchangers optimization methodology.

## 1. INTRODUCTION

While Immersed coil Heat Exchangers (IHX) employed in the *indirect* charging or discharging of Thermal Energy Storages (TES) influence overall system performance, the nature of this influence depends heavily on *natural convection heat transfer* and its transient fluctuation in bound enclosures. The factors influencing this fluctuation range from the physical characteristics of the IHX through to the conditions under which it is to operate within the TES, but these are not thoroughly accounted for in the design criteria widely used today. Often simple *rules of thumb* based on measurements from a limited number of IHX in the late 80s (Kübler, 1988 or Farrington and Bingham, 1986) and early 90s are used in which little more than IHX surface area is addressed. Given the variety of IHX designs in the current Domestic Hot Water (DHW) market, the applicable range of these rules of thumb may have to be revised in light of the experimentation from whence they came and developed further.

This paper presents parts of our continuing study of heat transfer with immersed heat exchangers. In order to confirm the statement that many of the IHX currently on the market do not fully fit existing knowledge, a research of the German TES market (taken as a big and representative market) is presented and discussed and a reference DHW system is defined (representing a typical commercial system). The reference system is used in TRNSYS simulations for the elucidation of common IHX operating conditions. This information is needed in defining the conditions for experiments in order to optimise the heat transfer area and the geometrical parameters of an IHX regarding both the heat transfer effectiveness and the affect to storage stratification. Thereby particular attention is paid to the interaction between IHX and TES temperatures and the influence mass flow rate has thereon (comparison between high-flow and low-flow). Furthermore, a representative selection of commercially available products for an extensive testing regime is discussed.

Heat transfer from an IHX to a TES fluid via natural convection is a problem with many facets. In the case of heat being collected from the sun, the weather dependent variance of power over time sets a basis for our analysis. Whereas the rate of energy stored from a boiler or electrical source is mostly constant, the power function of a solar absorber when viewed over the surface area of an IHX makes any simplification of the physical phenomenon of natural convection inside the TES flawed from the beginning. Compounded to this are the operating conditions set out in the selection of a suitable mass flow rate which defines the temperature gradient across the IHX surface area and with it, the intensity of natural convection. The positioning of the IHX within the TES influences global temperature gradients between the solar and standby partitions of the TES and thus the effectiveness of the solar collector - through the mean collector temperature and associated losses. The entrainment of cold water into the bottom of the TES is user specific and largely random. Primarily, an IHX is dimensioned as large as possible but such as to fit within the geometrical constraints of the TES. On the other hand, it should be as small as possible and easy to manufacture to avoid unnecessary cost to the system. How it is dimensioned though seems to be a combination of contemporary trends, speculation and experience.

To account for all the above mentioned interaction, spatial (2D or 3D) simulations with suitable computational fluid dynamics software of typically occurring operating conditions would allow an optimisation at considerable computational cost. Dynamic similitude approaches such as those implemented in TRNSYS types lighten the computational load yet have until now relied heavily on empirical functions not conforming to physical geometry or other such units of measure. A comprehensive study must also account for stratification in tank (quantification of mixing phenomena). In addition, the validation of CFD against empirics (calorimetric and observations with PIV and LIF) is required.

In general, a physical dimensioning would lead to a heat exchange area made to suit a particular range of operating conditions. It has yet to be clearly defined how this area is to be arranged such that a good heat transfer and an optimum stratification behaviour can be achieved.

Through identification of the cardinal geometrical parameters as per Figure 1, attention may be drawn to the relationship of the pitch of the coil helix to the diameter of the tube - the so called *pitch-to-diameter* ratio – and its affect on natural convection. Exploration by Messerschmid (2002) and others have shown for example that too small a $p/d$ leads to a decrease in heat transfer due convective currents not passing between tube helixes, but rather gaining momentum along the equivalent-vertical-wall presented by the closeness of the coils inside and outside surface areas.

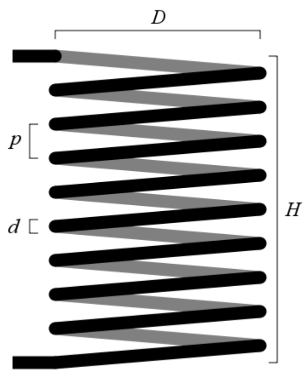


Figure 1. Cardinal geometrical parameters of an IHX. Through mathematical simplification we can sketch such a helixed tube in cartesian coordinates by rotating a circle around the helixes centre with a pitch $p$ climbing the z-axis. When we speak of the $p/d$ ratio though, we subtract $d$ from $p$ such that only the distance between coils is implied; which then coincides with common nomenclature.

The most rudimentary rules of thumb used to estimate the heat exchange area vary and are rather arbitrary. For example DGS (2004) simply recommends 0.25m² IHX area (in case of smooth tubes) for every m² of collector area in the dimensioning of IHXs for small DHW systems. Estimated heat exchanger performance might then be graphed as a function of mass flow given an assumed constant $\Delta T$ across the IHX (constant power).

## 2. MARKET RESEARCH

The market research was performed on two different levels. The first encompassed a representative cross-section of the TES market between 350 and 450 litres - collecting all geometrical values of the TES and IHX together into a dataset - and the second level involved an appreciation of the solar packages available on the market integrating these TESs in various configurations. Observing these two dataset correlatively showed not only deviation from mean parameters, and insofar imply significant variation in the methods used for the designing of IHXs, but also identified certain constraints within the design process that are perhaps at first glance not visible.

It became evident during the collation of TESs that installation of smooth tube helical coils falls into three main geometrical IHX configurations in the TESs market, as illustrated in Figure 2. This made the recording of geometrical parameters (specifically the observation of $p/d$) a matter of splitting those relevant for the total IHX (average parameters) and those relevant for the majority of the IHX (main).

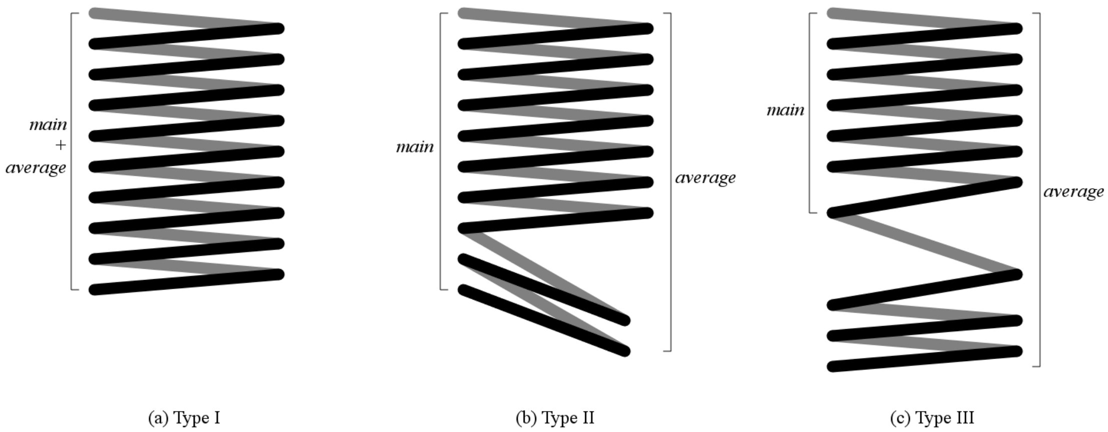


Figure 2. Predominate variations on IHX designs – a fourth class is defined in the combination of type II and III.

In addition to those parameters shown in Figure 1, TES geometry was recorded and a dataset consisting of fifty-two TESs was collated. After comparing a few of the parameters against one another, for example in looking at the variation in $p/d$ and $A_{IHX}$ over the sample set (see Figure 3), it was decided to look at the correlation of all parameter variables with one another over the dataset, as shown in Table 1.

Observing how two variables move with one another can elicit information concerning unthought-of constraints to variability in the system design. Taking the example illustrated in Figure 3 and finding the correlation between $p/d$ and $A_{IHX}$ in Table 1 reveals a mild negative correlation coefficient of -0.11 and -0.39 for the main and average $p/d$ respectively, indicating that an increase in the surface area of the IHX could lead to smaller $p/d$, which when thought about is intuitive in that the greater the surface area, the more IHX there is to fit into a specific volume, which induces smaller $p/d$ ratios (given that $d$ hardly varies).

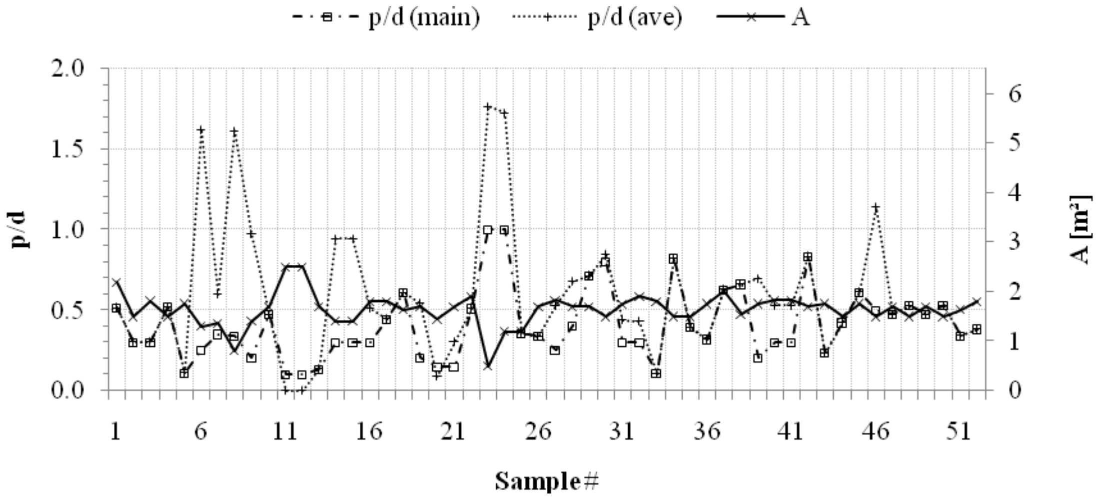


Figure 3. The variance of $p/d$ and $A_{IHX}$ over the sampled dataset. What we see is a large variance in $p/d$ and a not so large variance in $A_{IHX}$. This would tend to imply that $A_{IHX}$ adheres to some rule and $p/d$ less so.

Table 1. Correlation matrix of geometrical parameter dataset of fifty-two TESs ($V$ refers to Volume and $A$ to surface area) - the correlation coefficient can range from -1.0 to +1.0; the former indicating a "moving against one another", the later a "moving with one another" and zero, a non-correlation in that the variance (or non-variance) of one variable cancels the other out.

| | $V_{TES}$ | $H_{TES}$ | $D_{TES}$ | $H_{IHX}$ | $D_{IHX}$ | $A_{IHX}$ | $p/d_m$ | $p/d_a$ |
|---|---|---|---|---|---|---|---|---|
| $H_{TES}$ | 0.36 | | | | | | | |
| $D_{TES}$ | 0.32 | -0.49 | | | | | | |
| $H_{IHX}$ | 0.05 | 0.17 | -0.06 | | | | | |
| $D_{IHX}$ | 0.33 | -0.24 | 0.54 | -0.39 | | | | |
| $A_{IHX}$ | 0.62 | 0.26 | 0.47 | 0.22 | 0.19 | | | |
| $p/d_m$ | -0.06 | 0.09 | -0.08 | 0.42 | -0.22 | -0.11 | | |
| $p/d_a$ | -0.27 | -0.01 | -0.26 | 0.03 | 0.24 | -0.39 | -0.04 | |
| $Type_{IHX}$ | -0.30 | -0.11 | -0.31 | 0.25 | 0.07 | -0.70 | 0.26 | 0.57 |

Generalisation based on the correlation matrix include:

- The dimensions of the TES (diameter $D_{TES}$, height $H_{TES}$ and volume $V_{TES}$ of the storage) do not influence the height of the IHX, which seems to be something quite left up to the designer, but do seem to mildly influence the diameter and area of the IHX in a positive way.
- For both the TES and the IHX there is a medium negative correlation between diameter and height, supporting the idea that there is no agreed optimal height to diameter ratio in either case, and that these vary from manufacturer to manufacturer.
- An increase in $p/d$ requires either an increase in IHX height or decrease in surface area.

- As the dimensions of the TES increase (relatively speaking within the range of our selection) $p/d$ decreases. This raises the question of whether the relationship between TES volume and IHX surface area is non-linear or not, in that a rise in system size and with this volume requires proportionally more IHX surface area than there is room in the TES to be filled by it.

The question of which - if any - rule of thumb is being used for the dimensioning of the IHX surface area became the second task in a study of the market. In this task the relationship between TES size, IHX area and collector area (all flat-plate collectors) of thirty-eight packaged systems (between 300ℓ and 550ℓ) was collated - shown in Figure 4 against the DGS rule of thumb mentioned earlier in Section 1. It is obvious that for small DHW systems (for collector areas between 4 and 8 m²) the DGS rule holds more or less true. Within this sample are four manufacturers offering a range between 300ℓ and 500ℓ which display on average a 0.15 m² increase in IHX area with every m² collector area, which may be a rule of thumb from another source. Whether the deviation of collector areas to IHX surface area can be accounted for in the respective performances of each systems collector was not explored in the scope of this task.

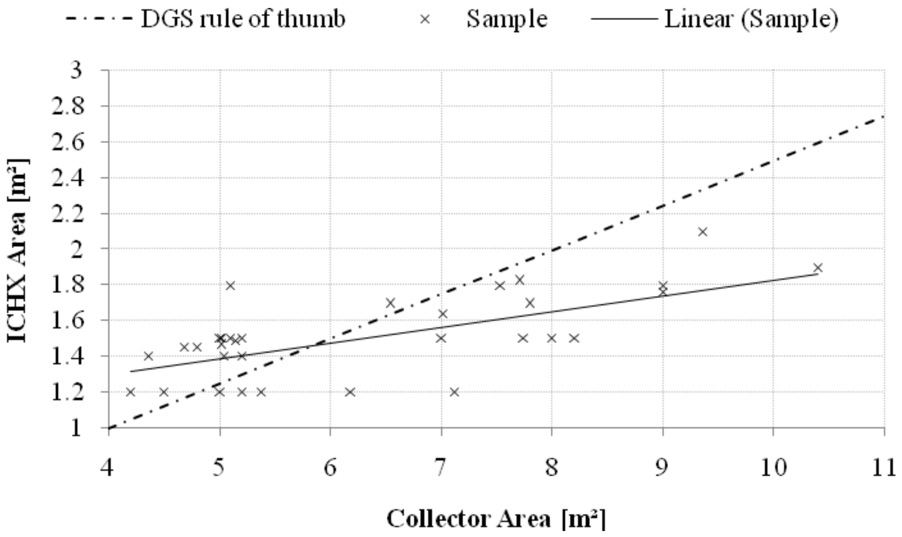


Figure 4. A sample of 38 solar thermal packaged systems showing the relationship of collector area to IHX surface area.

## 3. SYSTEM SIMULATIONS

To assist in the defining of *typical operating conditions* which are necessary for the experimental and numerical design optimisation, a TRNSYS deck was created and annual simulations were performed to generate results presenting statistically *typical* conditions. An ancillary aim of this work was to understand the manner in which IHXs are currently handled in system simulations and open a discussion on the various merits of flow rate and system control when observed from IHX performance. Formed predominately from the mean values obtained from Section 2, a TRNSYS simulation deck representing a 400ℓ bivalent domestic hot water TES (Table 2) with approximately 7m² collector area inclined at 45° was created and simulated in Wurzburg/Germany for both low-flow (10 kg/h.m²) and high-flow (50 kg/h.m²) specific collector mass flow rates and temperature threshold switches. A 200 ℓ/day *random* draw profile as defined within SHC Task 32 methodology was implemented (cf. Heimrath and Haller, 2007).

Table 2. TRNSYS Type 340 TES and IHX parameters - see Drück et al. (2006) for details concerning the modelling of IHXs in Type 340

| | | |
|---|---|---|
| TES volume | 400 | ℓ |
| TES height | 1500 | mm |
| Thickness of insulation | 100 | mm |
| Solar IHX area | 1.7 | m² |
| Solar IHX U-value | 70 | W/m².K |
| Auxiliary IHX area | 1.2 | m² |
| Auxiliary IXHC U-value | 100 | W/m².K |

**Energy Sums and Mean Heat Fluxes**

A summary of the TRNSYS simulations for both low- and high-flow is listed in Table 3. The roughly 3.5% higher collector yield of the low-flow system may be a result of better stratification in the storage or might be an artefact of the modelling limits to the TRNSYS Multiport Type, in that stratification and natural convection are handled numerically. Presumably because collector losses to the environment are higher in the case of a low-flow system, the results for mean collector power measured at the IHX (see Figure 5) are better for high-flow. It would be pre-emptive to conclude anything decisive about the difference in system performance between low- and high-flow systems purely from TRNSYS simulation results, however it is safe to assume that the low-flow system does achieve this greater yield from the collector due to a *better* stratification in the solar IHX partition of the TES, which improves IHX effectiveness.

Table 3. Energy sums of TRNSYS simulations for low and high specific collector mass flows.

| Mass flow | $Q_{collector}$ MJ | $Q_{auxiliary}$ MJ | $Q_{DHW}$ MJ |
|---|---|---|---|
| 10 kg/h.m² | 9095 | 5198 | 10831 |
| 50 kg/h.m² | 8779 | 5486 | 10831 |

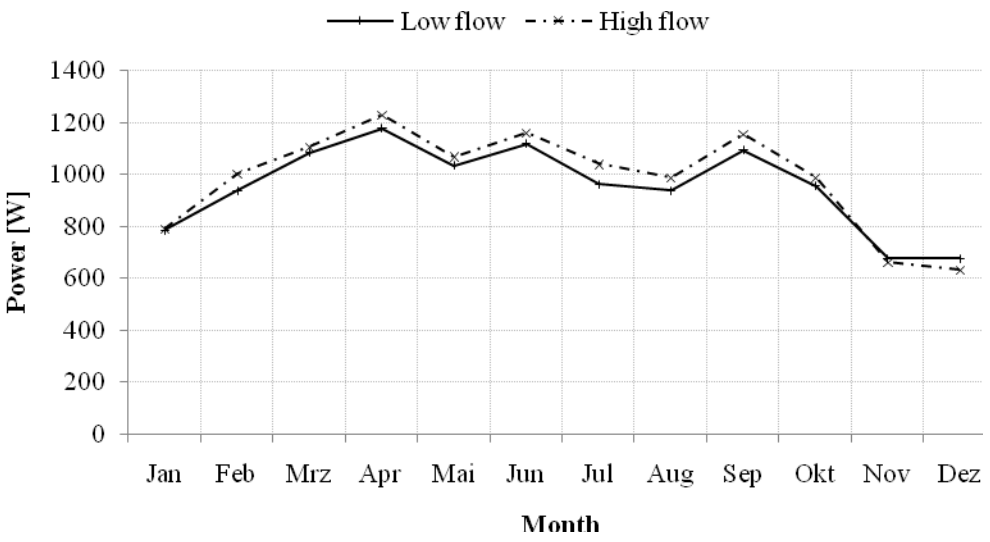


Figure 5. Monthly mean collector power observed from the collector IHX

The affect of the 45° collector inclination can be seen where April and September are the most powerful operating months. Seeing as this mean IHX power is a function of the time-step of the simulation in that each instance of power being generated in a time-step (3mins) is weighted as equally as the next, one cannot yet be certain whether the mean IHX power viewed over a year is in fact the *typical condition* one should concentrate on or not.

The question of what the most important operating conditions of an IHX that need to be focused on when conducting experimental investigations are is not necessarily easy to answer. Therefore, to help accentuate the importance of any particular operating condition throughout an annual simulation, it was decided to *weight* any figure displaying the annual results with the energy exchanged for that time-step and operating condition. This was done by creating discrete intervals over which the observed temperatures and heat fluxes could be binned (speaking in the sense of making histograms) and subsequently displayed on meshes where contours indicate the *height* of energy generation upon the chosen axes. The information we extract from such visualisation is of statistical value in that mean temperatures and fluxes can be identified and weighted according to the energy transferred under these conditions.

In Figure 6 we display what an *energy-transfer-weighted* power in dependency of the mean temperature difference between IHX and TES looks like for a low-flow system with a simulation time step of 3 minutes. Although much energy is still transferred around the mean collector power of approximately 1 kW across a varying $\Delta\vartheta_{\mathrm{IHX,TES}}$ the operating conditions under which most energy is transferred in a year is closer to 2 kW.

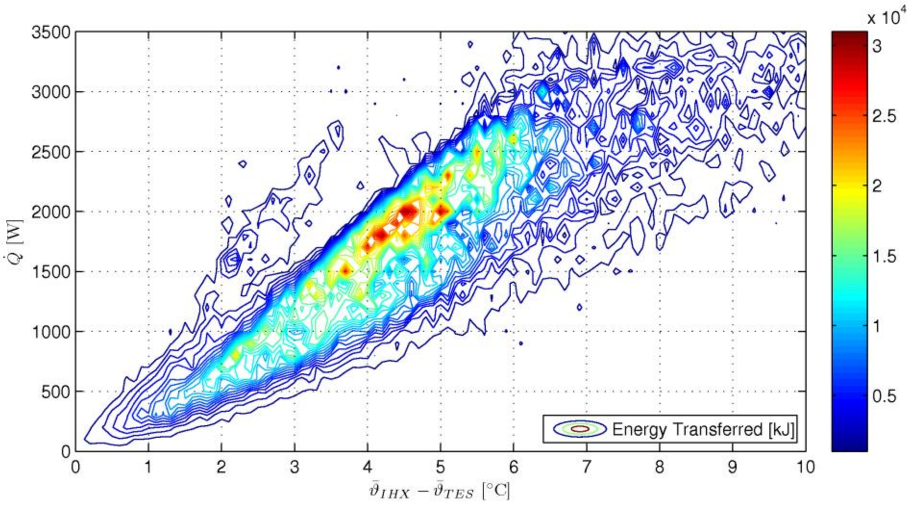


Figure 6. Collector power as a function of *mean temperature difference* between IHX and TES for $\dot{m} = 10 kg/h.m^2$ .

**Temperatures of IHX and TES**

The collector mass flow dependency of temperature difference across the IHX influences the natural convection within the TES and as such, choice of system flow rate defines not only the performance of the collector (in the form of losses to environment) and IHX, but also how much the heat brought in by the IHX is able to stratify in the volume of the TES occupied by it. Observation of the temperature gradient in the TES between the bottom and top of the IHX revealed that a low flow system has on average a 15 K gradient while the high flow system only has 5 K.

In Figure 7 and Figure 8 one can see the energy weighted relationship between the collector IHX entrance inlet temperature and the average TES temperature, whereby temperature switching thresholds relating to collector pump control for both have been included. From these two Figures we can visualise the temperature gradient across which the boundary layers of natural convection most often find themselves. To the right of the solid black diagonal line indicating where IHX inlet temperature and TES temperature are equal, the simulation artefact of *negative* energy on the TES is visible. In such cases the collector pump is running but the net effect of this operation is negative to the TES energy balance. This is mainly a result of collector pipe losses and inevitable control inaccuracies.

In the case of high-flow it can be seen that most energy transferral occurs within a difference between the IHX inlet temperature and the storage temperature of 10 K, with all other energy transferred above this limit resulting from occasions where DHW is drawn at the same time as the TES is charged from the collector. The linear *energy-transfer-weighted* interpolation (red line with equation) is a least-squares fit of the data whereby every time-step instance is weighted with the energy that was transferred for that time-step and shows how the temperature gradient between IHX and TES varies depending on the mean temperature between IHX and TES. The weighted linear interpolation is not so heavily influenced by instances where both charging and discharging are taking place and sets out for any given mass flow a mean $\Delta\vartheta$ dependent on a mean TES temperature.

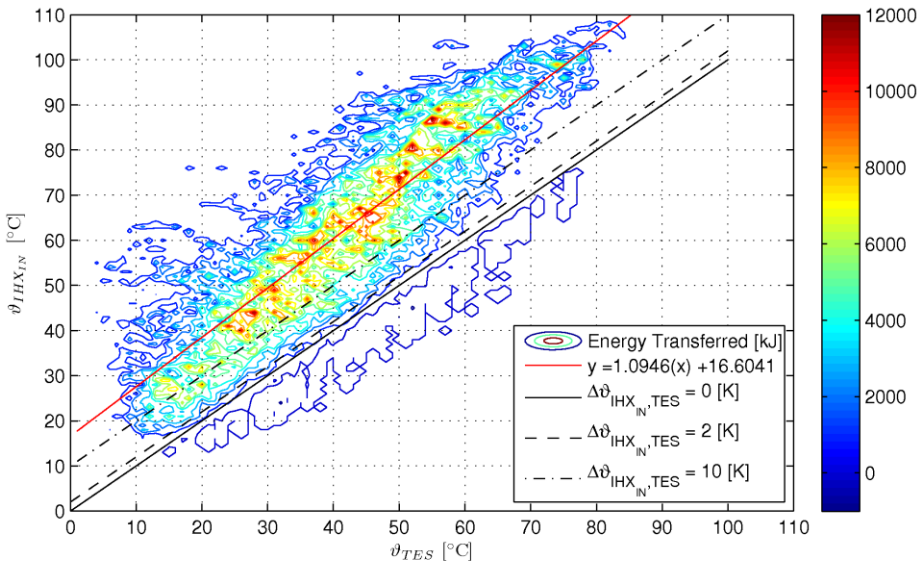


Figure 7. Energy weighted temperature histogram for $\dot{m} = 10 kg / h.m^2$ collector area; $\Delta\vartheta_{IHX_{IN},TES}$ is 10 K for switching on and 2 K for switching off.

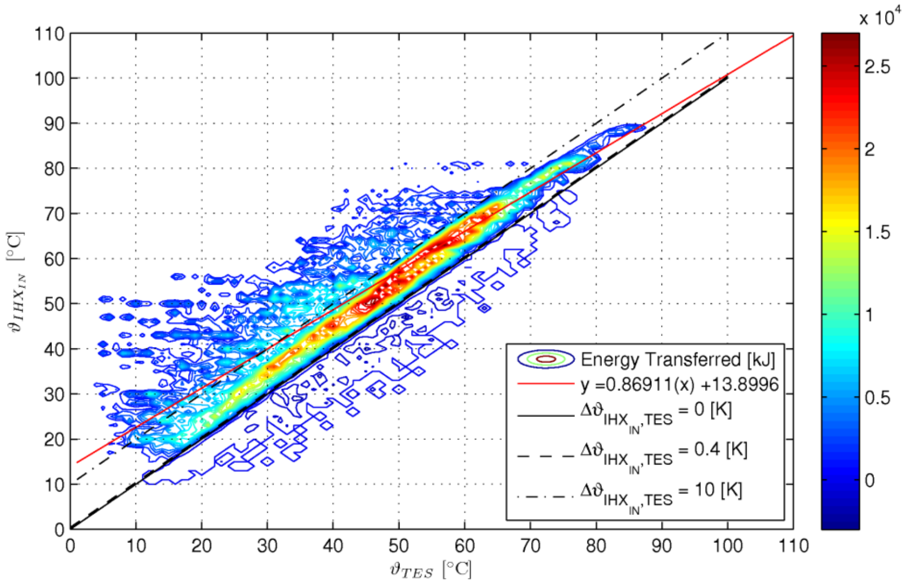


Figure 8. Energy weighted temperature histogram for $\dot{m} = 50kg / h.m^2$ collector area; $\Delta\vartheta_{IHX_{IN},TES}$ is 10 K for switching on and 0.4 K for switching off.

## 4. CONCLUSIONS AND OUTLOOK

In the process of the market research it became evident that indeed much variation of IHX geometry can be found. Taking for example the $p/d$ ratio, it was discovered that despite almost every IHX utilising a one-inch tube (33.7 mm), it ranged from 0.1 to 1.7 with most falling somewhere between 0.2 and 0.4. The most common IHX type seemed to be Type II while the variation in IHX heights ranged from 300 up to 750 millimetres.

While it is important for testing sequences based on simulated insights to be performed on real market IHXs, the likelihood of finding any two, between which only one of the many geometrical parameters is or can be varied, is near impossible. For this reason testing shall involve an investigation of market IHXs embodying the variations over single parameters $A_{IHX}$, $p/d$ and $H_{IHX}$ on the one hand and parametrical variations, for which specifically built IHXs are required, on the other.

To summarise, the results of the market research and the analysis of the simulation results both indicate that there is potential for cost reduction where an optimisation of the geometry of IHX leads to higher IHX effectiveness and better storage stratification. Refining of the commonly used rules of thumb would include optimised geometry parameters that maximise IHX performance for minimum IHX surface area.

Once the selection of representative IHXs based on the market research and the operating conditions that need to be focused on for the experimental investigations are finalised following the results presented in this paper, detailed flow and heat transfer measurements will be conducted in the SPF laboratory. For this, the methods of Particle Image Velocimetry and Laser Induced Fluorescense will be integrated. In addition, a suitable numerical IHX and storage model will be explored in CFD and TRNSYS and validated with measurements. A comprehensive optimization methodology of IHX will subsequently be aimed at.

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