Thermal behaviour of a heat exchanger coil in a stratified store

THERMAL BEHAVIOR OF A HEAT EXCHANGER COIL IN A STRATIFIED STORAGE

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ABSTRACT

A top-bottom distributed helically wound coil is an attractive option for a stratified solar energy storage. Literature survey provided relevant data covering natural and forced convection heat transfer in and outside the coil. Numerical models, based on the experimental heat transfer data, yield the storage and the solar system performance. The heat exchanger geometry is improved with respect to the thermal performance.

KEYWORDS

Stratified storage, helical heat exchanger coil, natural convection, DHW-system, heat transfer, "single pass", "low flow".

INTRODUCTION

In most solar Domestic Hot Water (DHW) systems, a heat exchanger separates the collector circuit hydraulically from the mains water circuit to facilitate overheating- and freeze protection by collector drain back or use of glycol, also protecting the collector circuit from corrosion and scaling by mains water. The figure below shows two configurations using a helical heat exchanger coil.

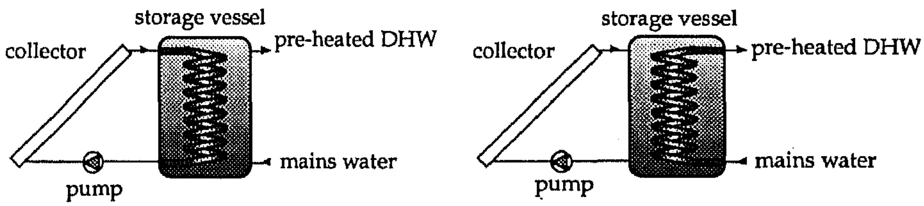

Fig. 1. Solar DHW system configurations

In the configuration shown left, the power to be transferred by the coil equals the relatively low collector power output whereas in the configuration shown right the power to be transferred equals the higher DHW-load. If, however, a DHW-coil is applied the storage vessel does not need to be mains water resistant.

Main objective of the research is to gain insight in the thermal behavior and to provide design guidelines for both options. As natural convective heat transfer phenomena are temperature dependent, detailed modelling is required. Ongoing research concentrates on thermally stratified storages. Conservation of high temperature differences in a stratified storage showed to be beneficial for reducing the heat exchanger size. This advantage adds up to the more generally known advantage of a stratified storage (reduction of entropy production and therefore conservation of exergy).

Inside Heat Transfer

Like in a straight tube the flow in a helically coiled tube shows a laminar, a transition and a turbulent flow region. Apparent discrepancy with the straight tube flow is the higher transition point from laminar to turbulent flow caused by the stabilizing effect of centrifugal forces, Gnielinski (1986) :

$$Re_{crit} = 2300\left(1 + 8.6\left(\frac{d}{D}\right)^{0.45}\right) \quad (1)$$

in which :

Re_{crit}	critical Reynolds number	[-]
d	tube diameter	[m]
D	coil diameter	[m]

The transition to a fully turbulent flow occurs at Re = 2.2 10^4.
In the laminar region the flow may be influenced by buoyancy– and centrifugal effects. Figure 2 shows the influence of both effects on the flow field.

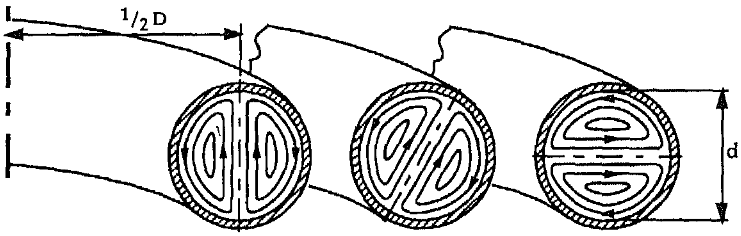

Fig. 2. Buoyancy and centrifugal effects on the flow in a curved tube

Laminar Heat transfer inside a helically coiled tube is studied by Futagami and Aoyama (1988). The Nusselt number Nu_b for flows in which buoyancy effects dominate :

$$\left(\frac{Nu_b}{Nu_0}\right)^{4.5} = 1 + \left[0.19\,(Re\ Ra\ Pr)^{0.2}\right]^{4.5} \quad (2)$$

in which :

Nu_b	Nusselt number for buoyancy effects	[-]
Nu_o	Nusselt number for Poisseulle flow in a straight tube	[-]
Re	Reynolds number	[-]
Ra	Rayleigh number	[-]
Pr	Prandtl number	[-]

The Nusselt number Nu_c for flows in which centrifugal effects dominate :

$$\left(\frac{Nu_c}{Nu_0}\right)^{6} = 1 + \left[0.195\,(Dn\ Pr^{0.54})^{0.5}\right]^{6} \quad (3)$$

in which :

Nu_c	Nusselt number for centrifugal convection	[-]

The dimensionless number Dn depicts the Dean number :

$$Dn = Re\sqrt{\frac{d}{D}} \quad (4)$$

For flows with buoyancy and centrifugal effects, the Nusselt number is an appropriate combination of (2) and (3). Due to the chaotic flow patern, turbulent flow is appreciably less influenced by buoyancy- and centrifugal effects. Gnielinski (1986) provides an equation similar to the Pethukhof-Popov equation, in which the Nusselt number depends mainly on the Reynolds and Prandtl number.

Outside Heat Transfer

Experimental research concerning **natural convective heat transfer in a stratified environment** is conducted by Eichhorn (1974). The temperature distribution is described in terms of a stratification index S :

$$S = \frac{a\,d}{\Delta T} \qquad (5)$$

in which :

S	stratification index	[-]
a	axial temperature gradient	[Km^{-1}]
Δt	temperature difference	[K]

The stratification index indicates the thermal level to which the plume rises. Figure 2 gives an impression of natural convection flow from a heated globe for several stratification indexes.

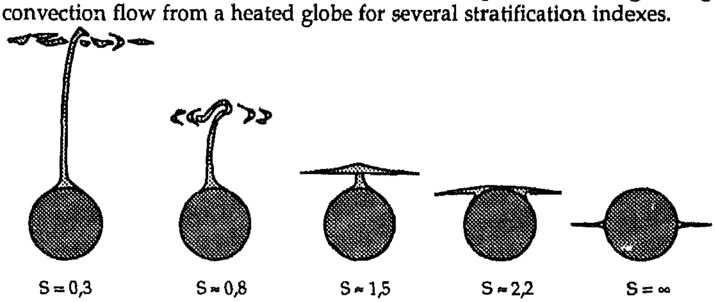

Fig. 3. Natural convection heat transfer flows from a heated globe in thermally stratified environment, Eichhorn (1974)

Rough experimental data showed a Nusselt number depending on the Rayleigh number and the stratification index. The experimental research, however, did not yield comprehensive heat transfer data.
Natural convection heat transfer from a vertical array smooth tubes is studied by Marsters (1972). Experiments showed that, for moderate Grashof numbers ($1.2\ 10^5$) and a tube pitch larger than twice the tube diameter the heat transfer from an upper tube is hardly influenced by a lower tube.
Henderson (1982) conducted thorough research on **heat transfer data outside smooth or finned tubes** in a solar energy storage. For outside heat transfer of smooth tubes the data correlated :

$$Nu = 0.53\ Ra^{0,25} \qquad (6)$$

For finned tubes :

$$Nu_s = 0.3365 \left(Ra_s \frac{s}{d}\right)^{0.285} \qquad (7)$$

in which

Nu_s	fin distance based Nusselt number	[-]
s	fin spacing	[m]
d	fin diameter	[m]

MODELLING OF HEAT STORAGE AND INTEGRATED HEAT EXCHANGER COIL

Based on the experimental heat transport data, a numerical segment model is developed. In the model, the storage is thought to be divided into several temperature layers, each of them fully mixed.

The helically coiled heat exchanger is thought to be distributed over the entire height of the storage. The heat exchanger inlet temperature in a layer equals the heat exchanger outlet temperature of the upstream layer. For simplicity, a perfect horizontal heat transfer is assumed, vertical heat transfer and heat loss to the environment are not taken into account. As temperature and heat transfer coefficient depend on each other, the computations have an iterative nature.

Step Charge Test Simulation

To gain insight, storage step charge- and discharge tests are simulated. As an example, the simulated step charge test results are presented for the configuration with the heat exchanger in the collector circuit.

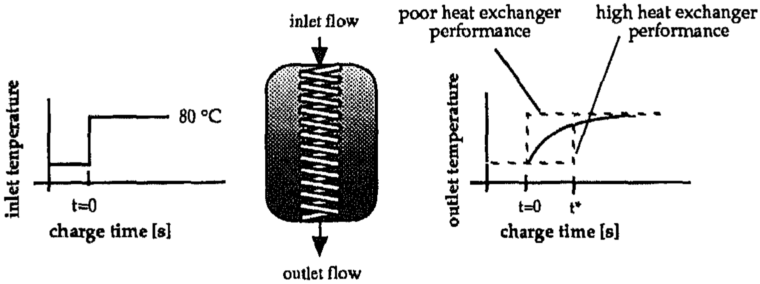

Fig. 4. Step charge test

The dimensions of the storage and the heat exchanger selected are based on a Dutch solar storage design comprising a cylindrical Ø 450 x 730 mm, 100 liter storage vessel, and a 6 m Ø 25 x 23 mm smooth stainless steel tube heat exchanger coil. The coil diameter amounts to 340 mm, the pitch equals 5-times the tube diameter. An initial uniform storage temperature of 10 °C is assumed. At $t = t_0$ the storage is charged with a 80 °C supply flow. The 20 lh^{-1} "low" flow rate is selected according to the "single pass" collector flow strategy which requires a daily collector throughput matching the daily DHW consumption. The heat transfer data are presented for a time interval of twice the characteristic charge-time interval $t^* = 18000$ s, the time in which the cumulative heat exchanger throughput equals the storage volume. If the heat exchanger would be absent (or perfect), and the storage perfectly stratified, the storage would be charged completely in the time interval t^*. Computations showed that the flow is laminar up to a flow rate of 300 lh^{-1}. Figure 5 presents the heat transfer coefficient α in- and out-side the heat exchanger coil. For graphical readability, the results are presented for a 5-layer model.

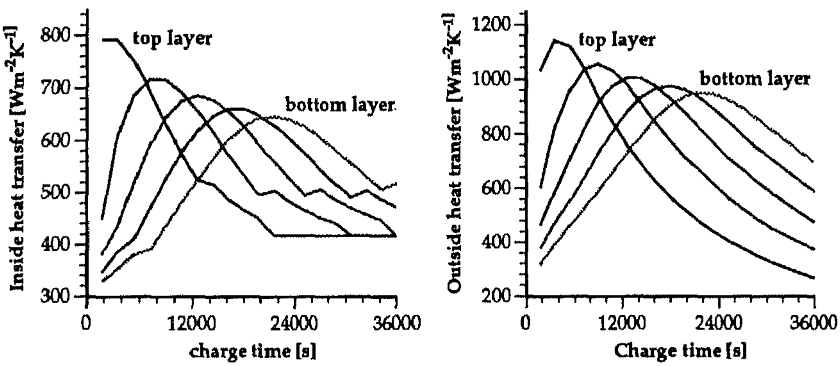

Fig. 5a,b. In- and outside heat transfer coefficients

The step-wise changes shown in the graph correspond with changes in flow regime caused by changes in the temperature dependent fluid properties. The average heat transfer coefficients in- and outside the tube are equivalent, indicating a well selected tube geometry.

As can be seen from fig. 5 both the in- and the outside heat transfer are governed by natural convection. The heat transfer coefficients depending on the local temperature differences move through the storage like a wave. Figure 6 shows the storage in- and outlet temperature.

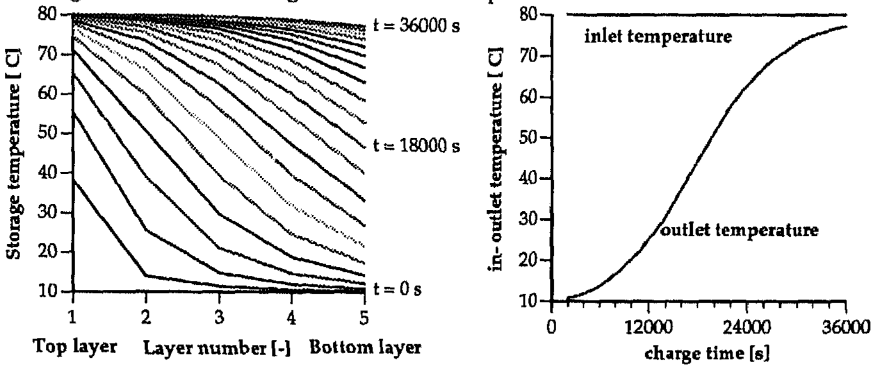

Fig. 6a,b. Storage-, in- and outlet temperatures

The gradually varying storage temperature over the height of the storage and the S-shaped variation in time of the outlet temperature indicate that the storage is not charged perfectly stratified. With an absent or perfect heat exchanger, the storage temperature would slide step-wise over the storage height, whereas the outlet temperature would show a step-wise variation from $T = T_{low}$ to $T = T_{high}$ at $t = t^*$. The thermal performance of the storage with integrated heat exchanger coil is indicated by the charge fraction defined as the ratio of energy charged after a time interval t^* compared to the maximum energy content of the storage. Graphically the charge fraction corresponds to the area beneath the storage temperature line at $t = t^* = 18000$ s (Fig. 6a) in comparison with the total area beneath the 80 °C temperature line. The charge fraction for the 20 lh^{-1} flow rate amounts to about 80 %. Perfectly stratified and fully mixed storages without a heat exchanger yield charge fractions of 100- and 63 %, respectively. A charge fraction of 100 % is hardly achievable with a heat exchanger. The more the storage is charged, the smaller the temperature differences driving natural convection. This mechanism limits the charge fraction, especially at high energy flow rates (the heat transfer coefficient does not vary proportionally to the mass and energy flow rate through the heat exchanger tube). As expected, the simulated step test at the "standard" flow rate of 200 lh^{-1} ($t^* = 1800$ s) shows a worse thermal performance, the charge fraction being about 50 %.

Improvement of the "low flow" Heat Exchanger Geometry

The charge fraction of the low flow heat exchanger is computed for various coil geometries (tube- and coil diameter, heat exchange area). One of the dimensions is varied, while holding the other two constant. Data are based on the 6 m Ø 25 x 23 mm smooth tube collector circuit heat exchanger at the flow rate of 20 lh^{-1}. Computations showed an increasing charge fraction with smaller tube diameters. Apparently the decreasing natural convection for smaller tubes is lower than the increasing forced convection. Moreover, outside natural convection heat transfer roughly depends on $d^{-0.25}$. The charge fraction showed a 6 % increase over an inner diameter range from 23 to 10 mm. Obviously the tube diameter is limited by construction- (the smaller, the longer the tube) and pressure drop considerations. Provisional calculations showed a 0.04 bar pressure drop (0,022 W power loss) for a 20 m long Ø 8 x 6 mm tube at 20 lh^{-1}. The charge fraction seems hardly sensitive to the the coil diameter, indicating that at 20 lh^{-1} centrifugal effects are weak. For the Ø 25 x 23 mm tube, the charge fraction showed a 0.7 % increase over a diameter range from 340 to 50 mm. With respect to an adequate horizontal distribution of heat the coil diameter might be selected so that the storage vessel plan area is equally divided between the inside and outside of the coil. Variation of the heat exchange area (by varying the tube length) showed an apt heat exchange area of about 0.45 m^2. Below this value, the charge fraction drops sharply, above this value, the charge fraction increases weakly. Cost considerations will determine the optimum coil size.

Solar Energy System Simulation Results

For solar energy system performance computations several numerical models are added to the storage/heat exchanger model. For simplicity the numerical model of the 2.6 m^2 flat plate collector is based on elementary collector theory (Duffie & Beckman,1980). Detailed modelling, required for "low" flow operation did not fit the research framework. Additional weather input data are derived from the Dutch reference weather file. The daily 100 liter DHW usage is concentrated in the morning and evening. An energy conservative mixing routine is added to the storage model to redistribute layer temperatures in case a negative storage temperature gradient might occur.
Computations are executed for "standard" and "low flow" systems with a heat exchanger in the collector circuit, the DHW circuit and, for reference purposes, for systems without any heat exchanger. Tentative model computations indicate that the system with an improved heat exchanger coil in the 20 lh^{-1} collector circuit yields a solar fraction of 56.4 %, which is merely 2,4 % lower than the highest solar fraction computed for the "low flow" system without any heat exchanger, indicating a well designed heat exchanger geometry. Apparently a heat exchanger yields a higher thermal performance when placed in the collector circuit. The charge time is longer, and the heat transfer rate is lower. Moreover, deteriorating heat transfer is counteracted by an increased heat exchanger supply temperature, the collector power output remaining nearly constant. The DHW coil encounters a serious penalty, as the limited discharge fraction of the DHW coil limits the yearly solar fraction considerably. Moreover, the heat exchanger supply temperature remains constant during the discharge mode. It is expected that the thermal performance of the DHW coil can be improved easily by increasing the storage volume, avoiding low temperature differences at the end of a discharge mode. Further research on this point is required.

CONCLUSIONS

1) Heat transfer coefficients of a storage vessel integrated heat exchanger vary considerably and cannot be treated as constant
2) For a "low flow" solar DHW-system with a heat exchanger in the collector circuit, the design guidelines comprise :
   - ☞ distribution of the heat exchanger over the entire height of the storage.
   - ☞ a tube diameter as small as pressure loss and construction considerations permits.
   - ☞ a coil diameter according to an equal storage vessel plan area in- and outside the coil.
   - ☞ a tube length which results in a heat exchange area of about 0.45 m^2 per 100 l storage volume.
3) A solar system designed according to these guidelines yields a yearly solar fraction which is merely a few percent lower than a system without any heat exchanger.
4) Further research is required on model validation and DHW-coil optimization

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