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Using an Ersatz Thermosiphon Loop to Model Natural Convection Flows Inside a Shallow Enclosure

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ABSTRACT

Natural convection loops (NCL) can occur when extracting energy from thermal storage with immersed heat exchangers. To assist in heat exchanger design and annual performance simulations of such systems, this paper proposes modeling an NCL with a comparatively simple "ersatz" thermosiphon loop (ETL). In an actual thermosiphon loop, fluid in channels or pipes flows in a closed loop, driven by a net buoyancy head which is equal to the total pressure drop. In the proposed approach, ersatz flow channels corresponding to the actual NCL flow are first defined, based upon experiment, numerical solution, or other information. The heat transfer and friction coefficients in the simplified ETL model must then be adjusted to fit these known data. The test case analyzed here is a horizontal shallow enclosure with temperature boundary conditions at both ends. A numerical solution is used to calibrate the ETL, and an analytical solution is used to extrapolate to other conditions for testing the ETL model predictions. It is shown that over two orders of magnitude variation in heat transfer, the calibrated ETL model predicts the heat transfer to 8% RMSD.

NOMENCLATURE

Symbols

A	area [m^2]
DT	temperature difference [^oC]
f	fitting factor
g	gravity [m/s^2]
H	height [m]
L	length [m]
m	mass flow rate [kg/s]
Nu	Nusselt number
Ra	Rayleigh number
Re	Reynolds number
P	pressure [Pa]
Q	heat transfer rate [W]
T	temperature [^oC]
U	heat transfer coefficient [W/m^2-^oC]
□	density [kg/m^3]

Subscripts

hot	hot leg of ETL
cold	cold leg of ETL
fric	friction
H	denotes box height as the length scale
htr	heat transfer
hx	heat exchanger

INTRODUCTION

Installed first cost of a solar domestic water heating system (SDHW) is typically ~$3000, compared to ~$300 for a conventional electric water heating system. This relatively high first cost is a key barrier to deployment of SDHW. A multi-year program in the US Department of Energy/Solar Buildings Program has the goal of reducing SDHW cost at least 50%, predicated upon use of low-cost polymer materials and manufacturing methods. The system type currently under development is an unpressurized "integral-collector-storage" system (UPICS), as shown in Fig. 1. This unit is in series with a conventional hot water heater. UPICS must have a load-side heat exchanger to transfer heat to a pressurized water supply system. When hot water is drawn, water flows from the mains supply through the heat exchanger (where it is pre-heated) into a conventional water heater. An advantage of this system type is that tank walls can be made thin to reduce cost and weight, since the tank is subject only to hydrostatic pressure of several hundred Pa (several psi). The disadvantage is that the load-side heat exchanger must be added. Issues in the geometry of the heat exchanger are the motivation for the work reported on here.

The heat exchanger presents a set of complex, interrelated issues. First, the heat exchanger must have sufficient capacity to transfer heat at a relatively high rate. For a typical temperature rise of 40°C (72°F) and a moderate flow of 0.1 lps (1.6 gpm), the thermal power is ~17 kW. Using literature heat transfer correlations for single tubes immersed in "large" enclosures, simple analysis as in (Arora 2001) indicates that, for the case of 3 mm OD tubes, a reasonable area is ~3 m^2 (UA_{hx} ~ 1500 W/°C). Tube spacing will influence U values, and optimum spacing under natural convection currents is unclear.

A second issue is the placement of this heat exchanger area within the solar storage. The tubes could be "dispersed" throughout the storage, or "concentrated" in a compact tube bundle. A tube bundle near the top of the tilted storage (heat exchanger tubes running horizontally across the top of the storage and parallel to the roof ridgeline above the unit) has practical advantages and is the configuration of interest here. A natural convection loop (NCL) is defined as a velocity field driven by buoyancy differences such that streamlines within the dominant flow field near the enclosure's physical boundaries form isomorphic closed loops. As indicated roughly in Fig. 1, the heat transfer with top-mounted tube bundle likely promotes a single-roll natural convection loop (NCL) inside the enclosure. However, the enclosure NCL is coupled to the heat exchanger flow, and the NCL resistance may reduce heat transfer. ETL may help to understand the geometries and storage fluids where the NCL limits heat transfer.

A third issue is development of stratification in the storage during discharge of the heat exchanger. Previous work has indicated stratification can affect performance of integral-collector-storage systems by as much as 25% (Christensen 2000). As cold water moves from the heat exchanger toward the bottom of the unit and hot water flows up to the heat exchanger, stratification develops in the enclosure, depending on the temperature difference across the heat exchanger, relative flow rates, and to what extent the cold down-flowing stream mixes with the hot water above. The ETL model may eventually help to understand these flows generally and, being a simple model, efficiently predict the stratification dynamics in the general case for annual performance simulations.

Studies are underway to characterize the heat transfer and flow dynamics of the heat exchanger in a tilted enclosure, using a complimentary experimental and numerical (CFD) approach. As indicated in Fig. 2, the experimental results are fundamental, serving to derive generalized heat transfer correlations and to validate the assumptions in the CFD modeling. After validation, CFD will provide detailed information on flow and temperature, support general heat transfer correlations, and may be used directly to optimize the heat exchanger design. However, CFD models are complex, and are computationally intensive. They do not per se provide understanding of flows, nor are they appropriate for use in simulations of annual system performance. One solution to address these issues is to develop a sufficiently simple "semi-empirical" model "explaining" the experimental and CFD results. A semi-empirical model is a simple but approximate physical model which is fit to "real" data under specific conditions and which incorporates the dominant trends under other conditions. This paper proposes a semi-empirical model for NCL flows, based upon replacing the real, complex problem with a relatively simple ETL within the enclosure. First, the basic ETL approach and theory are presented. Next, the CFD and analytical results used to derive ETL parameters and validate the approach are presented. The ETL model is then fit to the CFD result for one flow condition, and the adjusted ETL is compared to the analytical solution over a range of Rayleigh numbers. Lastly, conclusions are given and future work is outlined.

BASIC THEORY

This work postulates that when a single-roll natural convection loop (NCL) is known to exist in an enclosure, the NCL flow can be treated as a closed "ersatz" thermosiphon loop (ETL) consisting of "channels" forming a loop around the enclosure boundaries topologically identical to the real NCL. The concept is illustrated in Figs. 3-4. Figure 3 defines the case of a horizontal shallow enclosure adiabatic on top and bottom surfaces and with temperature conditions at the ends. Even in this simple case, complex flow patterns will exist, especially near the ends. Figure 4 shows the corresponding simple ETL model, consisting of 2 horizontal and 2 vertical pipes forming a loop roughly corresponding to the actual flow loop in the actual enclosure.

The equations governing the ETL behavior are derived from the Navier-Stokes equations assuming: a) temperature and velocity are uniform laterally across the channel; b) density variations are limited to the buoyancy term (Boussinesq approximation) (Dahl 1998). The resulting model equates the loop buoyancy head with pressure losses from friction and fittings (ΔP_f):

$$\oint \rho \vec{g} \cdot d\vec{s} = (\bar{\rho}_{cold} - \bar{\rho}_{hot}) g H_{vert} = \Delta P_f \qquad (1)$$

where the closed line integral is taken around the ETL loop and reduces to the middle term. ETL models allow temperature and/or flux boundary conditions anywhere in the loop. Heat transfer coefficients can be constant or depend on any problem variable. Friction losses for channel flow can be taken from standard literature correlations. Lastly, we expect that flow elements (such as a heat exchanger) can be inserted into the channel flow in the form of additional friction and source of heat. Although the heat-exchanger – ETL combination is the ultimate goal for this work, it is not treated further in this paper.

For the applications of interest here, it is expected the NCL flow will be laminar, with Rayleigh numbers on the order of 10^8 compared to turbulent transition around 10^{10} (Incropera 1981). ETL friction is calculated here using a laminar flow friction factor of 24/Re, appropriate for infinitely wide rectangular channels (Bejan 1984). Solutions of the coupled equations for mass flow rate and loop temperatures in the ETL are provided by a commercial equation-solver with thermodynamic functions (Klein 1999).

One weakness in ETL is that the channel widths and relative placement are somewhat arbitrary, and external results are essential to guide the choices. CFD or experimental results would be the basis for choosing the channel geometry. It is reasonable that the channel widths be defined by distances between zeroes in the NCL velocity field. An example is given below.

When the conditions change, the flow fields will change. For example, natural convection vertical boundary layer thickness should scale with Rayleigh number as $(Ra_H)^{-1/4}$ (e.g., Bejan 1984). If the ETL is applied "too far" from the calibrating case, we can expected the approach to break down. One implication is that ETL accuracy should be checked over the range of conditions of interest, for example calibrating the ETL at a "midrange" point and checking the extremes.

Another fundamental weakness of the ETL approach is that the conduction interaction between fluid in different ETL channels must be accounted for explicitly. For an actual thermosiphon loop in actual pipes, there would be no such thermal interaction. However, in an actual NCL the flows generally thermally interact via conduction. For two counterflowing streams interacting across a stagnant core, the distance between midpoints of the ETL channels is a reasonable choice for conduction distance. The conduction term affects the temperature profile and net buoyancy head. An example is given below.

CALIBRATION AND VALIDATION OF ETL

To determine the feasibility of ETL approach, we consider a case with a known solution where we can calibrate the ETL at one point and compare at other points. The case of Fig. 3 was chosen, since converged CFD results are available and a general analytical solution exists (Bejan and Tien, 1978). A specific CFD solution will be used to calibrate the ETL, and the analytical solution will be used for comparing the resulting model under other conditions. If the ETL approach works well in this case, it gives hope of working well when the enclosure is tilted and a heat exchanger is added into the loop.

The CFD solution was derived from a commercial fluid flow package (FLUENT, 2000). Fluid density was described as a linear function of temperature. Viscosity and specific heat were held constant. The solution was assumed laminar, although use of certain turbulence models seemed to aid in convergence by speeding dissipation of spurious vortices. End-to-end temperature difference was 10 ^{o}C, enclosure H/L = 0.073, and L = 1m. Figure 5 give results for the horizontal velocity field in the core at the middle of the box. Figure 6 shows the vertical velocity at the middle of the end region. The overall heat transfer was 144 Watts, and the NCL flow rate was 0.011 kg/s.

The CFD velocity field is used to define the ETL channel widths and locations. As in Fig. 4, there will be 2 horizontal and 2 vertical pipes forming the closed ETL. Figure 5 shows that the counterflowing horizontal streams each peaking at about 0.11H in from the wall, with return to zero reached at about 0.30H in from the wall. On this basis, it is reasonable to define the two horizontal pipes as each 0.30H in width and L in length, situated at top/bottom of the enclosure. Figure 6 shows the vertical velocity profile at the vertical hot wall. In this case, there is a thin boundary layer peaking at about 1 mm, returning to zero at ~ 5 mm. We define the two vertical pipes in the ETL model as each 5 mm in width and H in length.

It is interesting to note that the horizontal flow fields in the core region do not change significantly across the enclosure (.1L to .9L), as previously observed (Bejan and Tien, 1978). Although the vertical flow field (defined between ~ zeroes) at the ends does not change much in width, the flow grows about 40% between .33H to .66H due to entrainment of slower-moving nearby fluid. We systematically neglect all such complicating details.

It is important to note that the details of the "joining" of the 4 pipes are not considered. Acceleration terms (expansion, contraction and turning) are neglected. We also note that although the vertical height of the NCL streamlines is generally less than the full height of the box, the latter was taken as the height to use in the head computation. Other assumptions for H_{vert} (with $H_{vert} \le H_{box}$) will be accommodated by changes in f values (defined below). With $H_{vert} = 0.6*H_{box}$, f values (defined below) changed about 30%, with negligible change in the bottomline ETL validation.

Lastly, we need to incorporate terms for thermal conduction where it is expected to be significant. For the case analyzed in this paper, we neglect conduction in the short vertical pipes since the pipes are relatively short and the nearby stagnant core is near the boundary layer temperature. In the long horizontal pipes, we know that the temperature difference between flows across the stagnant stratified core cannot be ignored (Bejan 1989). The temperature difference is constant. The distance between the velocity peaks (.78H) is chosen as the conduction distance for interaction between the flows in the two horizontal pipes. The affect of this term is a temperature variation in the horizontal pipes which is linear in the horizontal distance. The average temperature (at the midpoint) is used to compute average density in Eqn. 2.

The ETL model is first fit to the CFD solution to adjust for the various "sins" of the simplified ETL model. Scale factors on the friction pressure drop (f_{fric}) and the end region heat transfer (f_{htr}) were introduced for this purpose. A value of one for the scale factor indicates that no adjustment is made from the "nominal" ETL model. A numerical search in f_{fric}- f_{htr} space was done to minimize the metric:

$$\chi^2 = [(m_{CFD} - m_{ETL}) / m_{CFD}]^2 + \\ [(Nu_{CFD} - Nu_{ETL}) / Nu_{CFD}]^2 \quad (2)$$

The metric is defined so that actual heat transfer and mass flow rate are both exactly matched via the two free parameters f_{fric} and f_{htr}. With the above assumptions, f_{htr} and f_{fric} values were 1.493 and 1.233. That these "adjustment factors" are near 1 indicates that the ETL may be a reasonable choice of simplifying model. It is worth noting that the two end pipes dominated ΔP_f , being about 17 times the core pressure drop. The f values are both dependent on the choice of channel widths. The f_{htr} values varied in proportion to the assumed width of the vertical channels (as expected for a laminar flow channel), and depended little on the horizontal width assumption. f_{fric} varied less than f_{htr}, changing ~25% for factor of 2 change in assumed channel widths.

An analytical solution for the case of Fig. 3 is given in (Bejan and Tien 1978). The analytical solution is derived by use of scale analysis in the core region and the end regions, matching the two solutions at their interface. Nu (relative to $Nu_{pure\ conduction} = k\Delta T/L$) is given as:

$$Nu = 1 + \left\{ \left[\frac{(Ra \times H / L)^2}{362880} \right]^n + \\ [0.623 Ra^{1/5} (H / L)]^n \right\}^{1/n} \quad (3)$$

with n = -.386.

The analytical result yields a heat flow of 148 W for the case at hand, agreeing well with the 144 W from the CFD. The constants in the general analytical expression for the mass flow rate were adjusted to match the CFD result, and this formula was then used to provide the "known" mass flow rates at "off-fit" conditions. It would have been more consistent to use CFD results for the "off-fit" values, but the analytical solution was more convenient.

Using the above values for the fitting factors, comparison is shown in Fig. 7 between the ETL model and the analytical model over a range of conditions. The predicted values of overall heat transfer, mass flow rate, and core temperature differences are shown. The Rayleigh number was varied by changing the hot wall temperature difference from 1^oC to 30^oC, keeping the cold wall temperature at 10^oC. The ETL model does appear to reasonably approximate the trends in the analytical model. Over the range shown (1 to 30^oC temperature difference), the heat flow varies a factor of 70, whilst the normalized RMS deviation between ETL and analytical solution averages ~ 8%, with a mean bias of 5%. This reasonable agreement indicates that the neglect of changing channel widths with changing conditions is not a serious error over the range of conditions investigated in this instance.

CONCLUSIONS AND FUTURE WORK

Simple modeling of single-roll natural convection loops in enclosures by a "semi-empirical" ersatz thermosiphon loop (ETL) has been described. The geometric details of the ETL are based upon experiment, CFD results, or other source. Factors are defined scaling the ETL friction and heat transfer coefficients to fit "known" results for heat transfer and total flow rate, at one operating point. The ETL model was fit to CFD results for the case of a horizontal shallow enclosure with heated and cooled ends, minimizing a metric that included both overall heat transfer and NCL mass flow rate. The friction factor f_{fric} was about 1.23, and the heat transfer coefficient adjustment f_{htr} was 1.49.

The calibrated ETL was also compared to an analytical solution for a wide range of Rayleigh numbers. After adjusting the details of the analytical model's mass flow rates to the CFD results, the ETL model acceptably reproduced the trends in heat transfer, flow rate, and core temperature difference over roughly two orders of magnitude variation in the heat transfer, differing by 8% RMSD with a mean bias of ~5%. On this basis, the calibrated ETL model appears a reasonable surrogate for actual results. Over the range of interest, it appears neglect of change in channel geometry with conditions is not a serious error.

Future work will investigate whether the ETL model is an acceptable approximation for a tilted enclosure with a heat exchanger concentrated at the top end. It is expected that a similar procedure for calibrating the ETL at one set of conditions will be done, subsequently validating the calibrated ETL with CFD at other geometries, tilt angles and temperatures.

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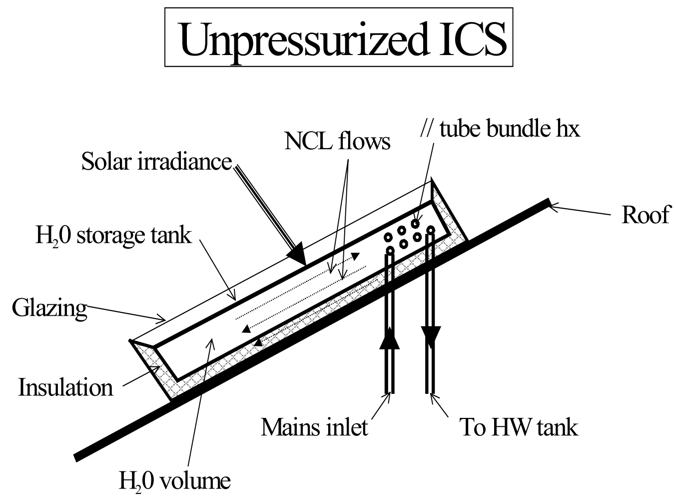

Figure 1. A schematic unpressurized integral-collector-storage system. The immersed load-side heat exchanger tube bundle (hx) is shown here as concentrated near the top of the storage. Natural convection loop (NCL) flows occur when there is a draw through the hx. Supply pipe (from mains inlet to hx inlet) and return pipe (from hx outlet into conventional hot water (HW) tank) are also shown schematically. As in many solar thermal collectors, the glazing on top of the absorber/storage tank and the insulation limit thermal losses.

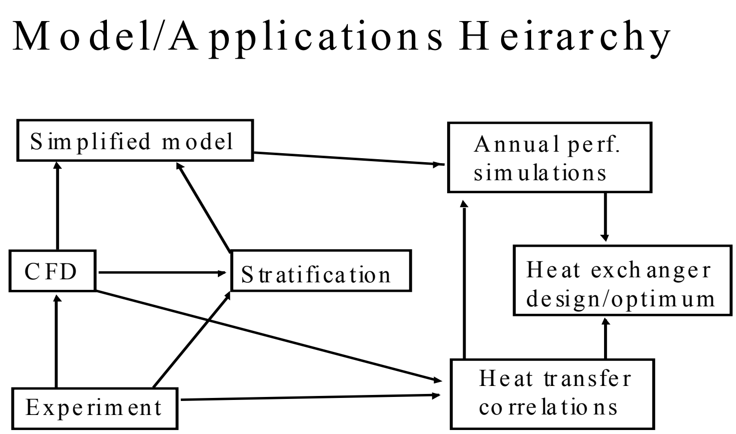

Figure 2. Heirarchical relationships between models and experiment. Experiment validates CFD assumptions, which in turn provides inputs to the simplified model (e.g., channel geometry). Experiment and CFD provide basic heat transfer correlations and characterize stratification development, which are needed to optimize the heat exchanger design. The simplified model allows variations in performance with conditions and flow stratification to be accounted for in annual simulations of system performance.

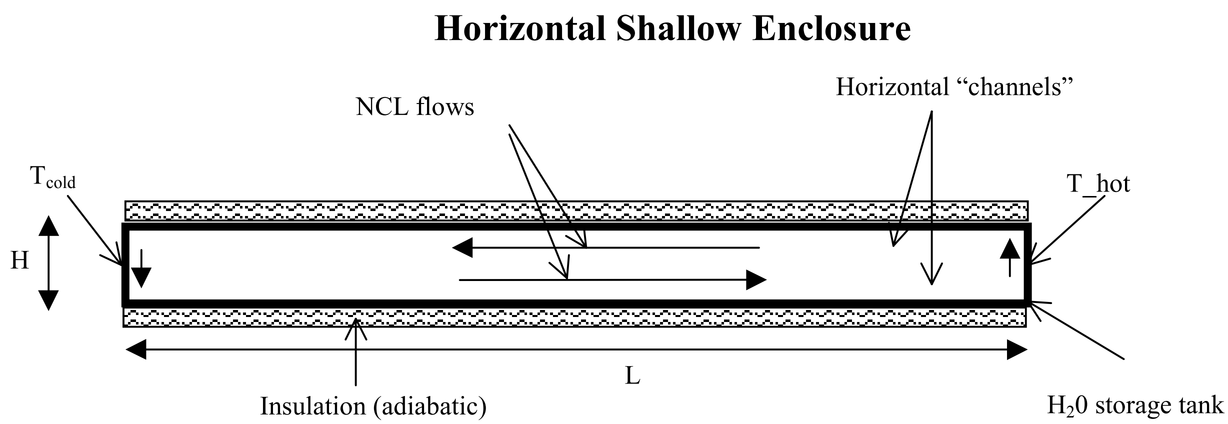

Figure 3. Schematic natural convection loop (NCL) flow channels, for the case of natural convection heat transfer between hot/cold end walls of an adiabatic shallow horizontal enclosure. The analytical solution indicates that isolated counterflowing hot/cold streams exist, which become more confined near the physical boundary as $H/L*Ra_H$ increases. H/L values of interest are between ~ .03-.1.

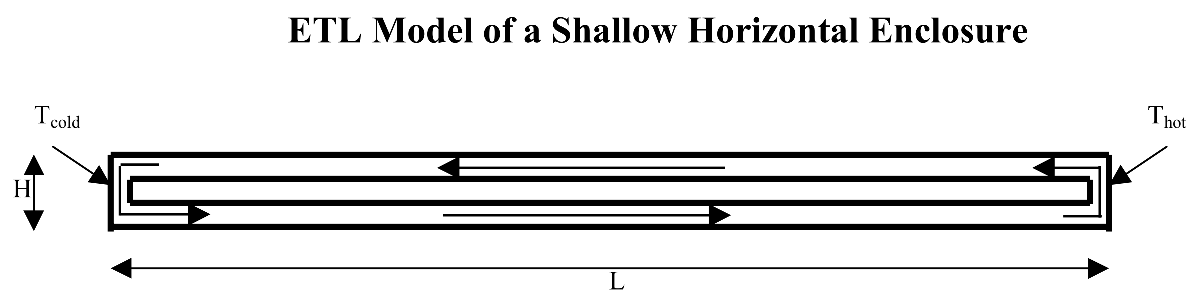

Figure 4. The problem of Fig. 3 transformed into an ersatz thermosiphon loop (ETL) consisting of 4 pipe segments, two horizontal and two vertical. If a heat exchanger were admitted, it would be added into the left vertical pipe.

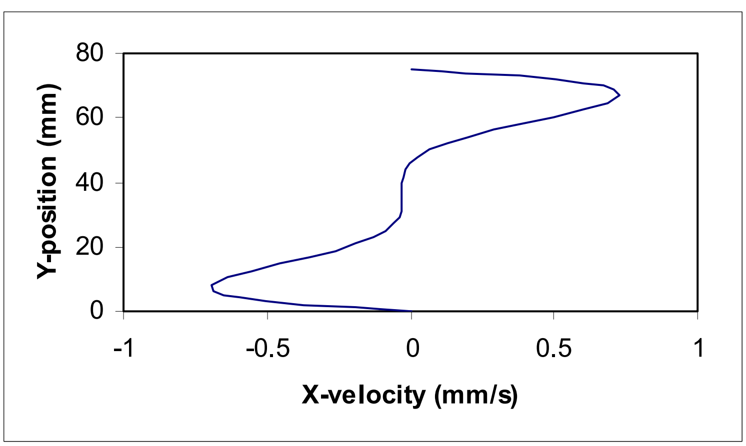

Figure 5. Velocity profile from CFD, in the core region of the shallow horizontal box of Fig. 3, at x = 1/2 L. There are two counterflowing boundary layers around a stagnant region in the middle of the profile.

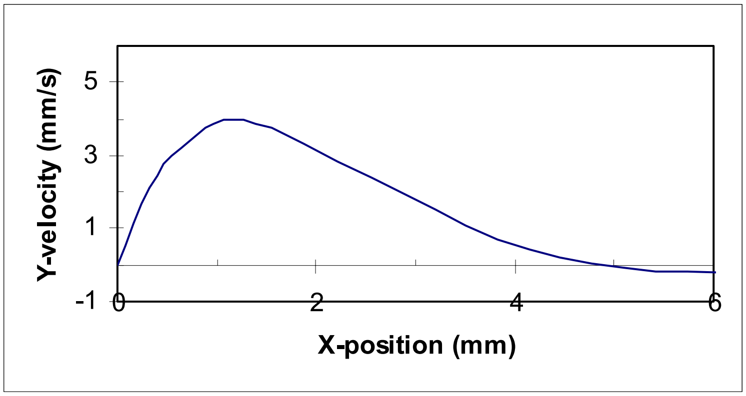

Figure 6. Velocity profile from CFD, at the midpoint of the heated end region of the shallow horizontal box of Fig. 3. There is a narrow velocity profile up the heated wall, peaking about 1 mm from the wall and extending out to about 5 mm. A similar (negative velocity) profile exists on the cooled end.

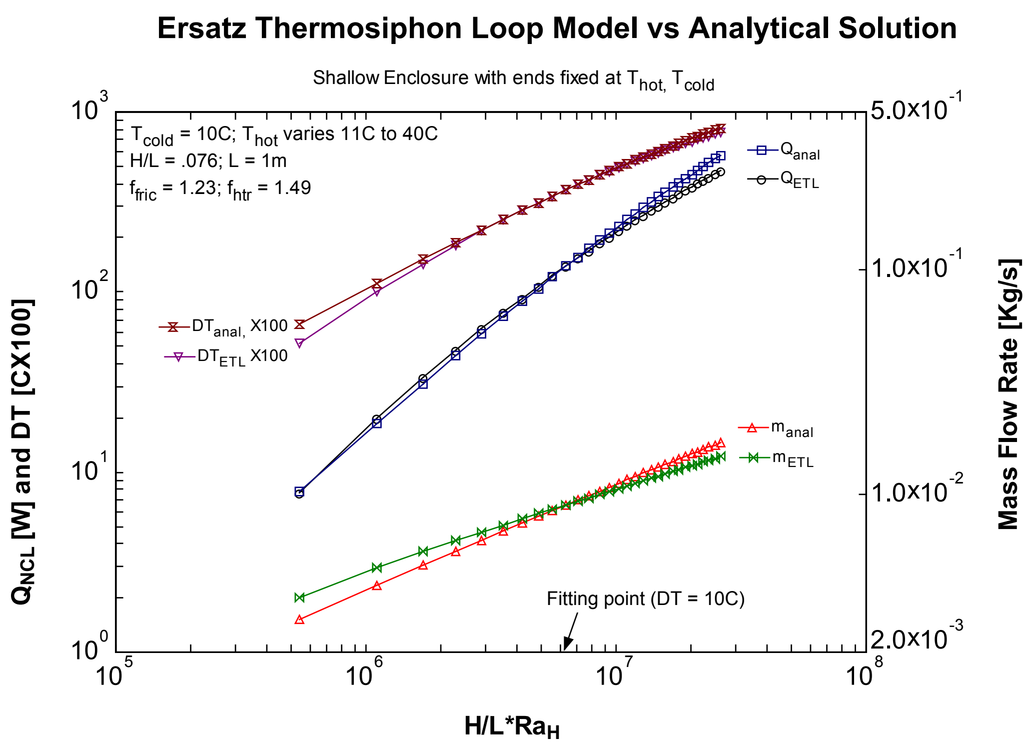

Figure 7. Comparison for the case shown in Fig. 3 of mass flow rate (m), total heat transfer (Q), and core temperature difference (DT), for the ersatz thermosiphon loop model (subscripted ETL) and for an analytical solution (subscripted anal). It can be seen that the ETL flow rate does not increase as rapidly as the analytical solution, but an opposite trend for temperature difference leads to good agreement on overall heat transfer.

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