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Pipe Freeze Prevention for Passive Solar Water Heaters Using a Room-Air Natural Convection Loop

Preprint

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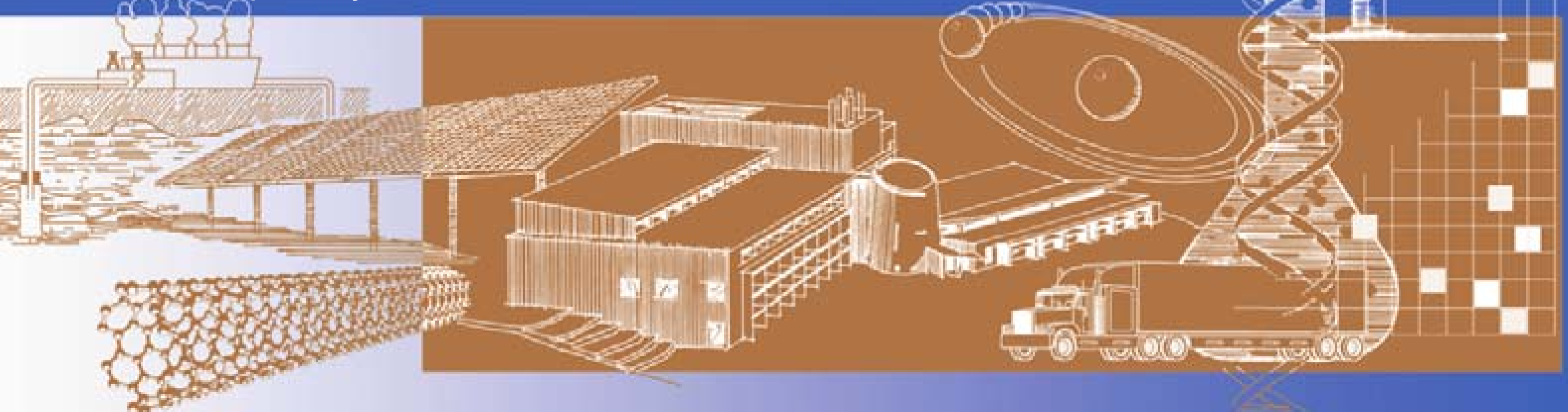

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PIPE FREEZE PREVENTION FOR PASSIVE SOLAR WATER HEATERS USING A ROOM-AIR NATURAL CONVECTION LOOP

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ABSTRACT

With the goal of extending the geographical market for passive solar water heaters northward, a room-air natural convection loop (NCL) was investigated for providing freeze protection for supply/return pipes carrying pressurized domestic water through unconditioned space. An NCL model was developed based upon balancing buoyancy and friction in the loop. Calibration factors (C_{fric},$C_{ht-trsf}$) were introduced to scale calculations and account for un-modeled details and correlation bias. An NCL prototype was constructed with 2”, 4”, and 8” diameters. Least-square regression on data from the 4” diameter duct yielded (C_{fric},$C_{ht-trsf}$) = (2.05, 1.8). The calibrated model was used to provide site-specific design guidance, and to estimate energy losses with and without a thermostatic control to stop flow whenever $T_{amb} > 0\,^{\circ}C$.

1. INTRODUCTION

Reducing costs and increasing reliability of solar water heaters is believed necessary for a substantial market to exist in the U.S. (1). One strategy to achieve these goals is increased use of passive solar water heaters (PSWH). PSWH include integral-collector-storage and thermosiphon systems, as in Fig. 1. PSWH eliminate the pumps, controller, sensors, and power needs of active systems, saving ~$400-$800 in life-cycle cost and eliminating related failures. PSWH also eliminate the need for separate storage inside the home, although roof weight is an issue. New polymer-based PSWH with potential for >50% cost reduction are nearing market entry (1). However, the market for PSWH has been limited by the risk of freeze-induced bursting of the supply and return lines (2). The supply/return pipes carry pressurized potable water through the typically-unconditioned attic space to the thermal storage, as shown in Fig. 1. The pipes are almost always copper because most SWH can reach temperatures that will burst polymer piping. On the other hand, copper pipe can freeze and burst in the attic during freeze spells. A burst attic pipe is catastrophic, as one incident may readily cost much more than the SWH’s lifetime savings. Therefore, the market for PSWH has rightfully been restricted as shown on the left side of Fig. 2. Note that if the building top-side insulation (normally at the ceiling) is raised to the roof plane, pipe freeze is eliminated in the attic. Such exceptions aside, lower-cost, more-reliable PSWH are at present generally considered unsuited for almost all the continental United States. If the pipe freeze problem could be solved, the PSWH market becomes limited by non-zero risk of collector-storage freeze, as indicated in Fig. 3. In the case of an indirect thermosiphon with well-insulated storage, the region of zero collector freeze risk would cover the entire lower 48 states.

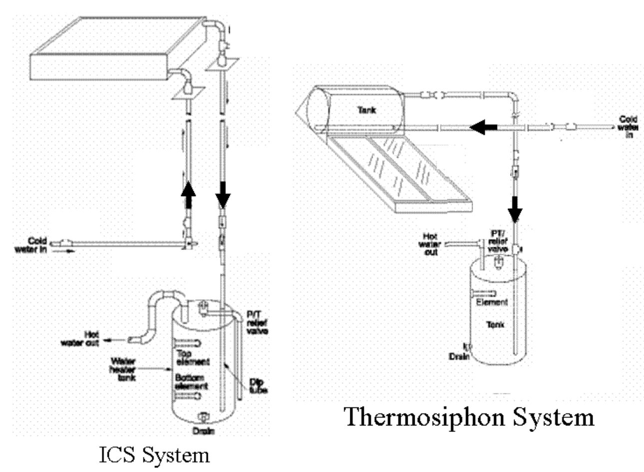

Fig. 1. Schematic integral-collector-storage and thermosiphon SWH. The balance-of-system is very simple, lowering costs and improving reliability compared to more common active systems.

It is suggested in (3) that a primary piping freeze protection (PFP) method combined with freeze-tolerant piping as the fail-safe backup to the PFP could be an acceptable solution to the pipe freeze problem. If the PFP fails and piping freezes, the homeowner must bypass the PSWH to regain hot water flow, which is a significant inconvenience. Thus, PFP failure must be very rare to be acceptable to the market. PFP must be passive to work during power outages and satisfy code regulations that any PFP cannot depend on electrical power. PFP must be low in cost, as SWH generally are marginally cost-effective against current conventional electricity.

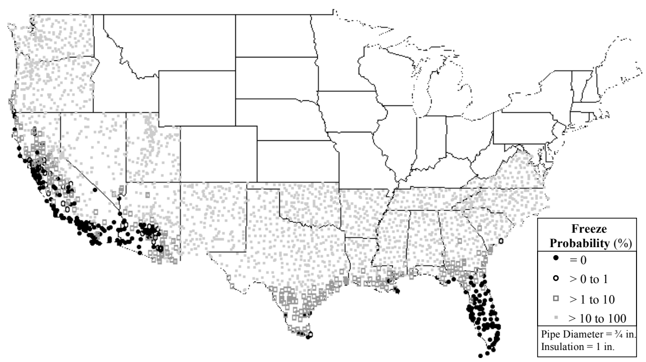

Fig. 2. Probability of at least one freeze in 20 years, for 3/4" copper pipe with 1" insulation. Zero-freeze-probability sites are shown as large dots. Map taken from (2).

PFP recently investigated are water-based methods using the supply/return pipes: 1) an NCL driven either by tank heat or room air heat (4,5); and 2) a freeze protection valve placed just upstream of where the return line crosses back into conditioned space (6). For NCL in the pipes, an additional pipe (with flow-limiting means) must be added to close the loop (4,5), adding plumbing costs. Pipe-based NCL can fail because: a) the flow-limiting restriction can become plugged; or b) the source of heat can fail (gas/electric storage tank, or room air). Similarly, freeze protection valves can fail because: a) scale or grit buildup can prevent the valve from opening; b) the actuator (e.g., wax compound) can degrade; c) the valve can improperly seat and leak; or d) water pressure is lost. This work considers an air-based, bi-directional NCL circulating room air through an insulated duct surrounding the piping, as shown in Fig. 4. To provide freeze protection, the exit temperature must be above zero when T_{amb} is set to $T_{amb,min}$. A room-air NCL may have improved reliability, because it eliminates issues with scale, orifice clogging, or plumbing leaks. However, there are still failure modes: a) room air too low; b) duct obstruction (e.g., insect infestations); or c) holes in the duct (e.g., poor installation or mechanical abuse). The air duct also presents more area for heat losses compared to pipe insulation. The objectives here are to construct and validate a model for a bi-directional NCL driven with room air.

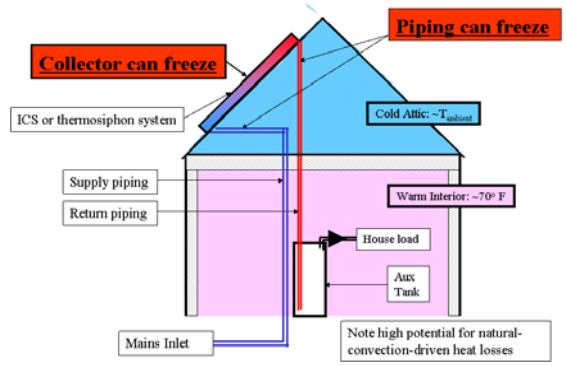

Fig 3. There are *two* freezing problems for PSWH: pipe freeze, and collector-storage freeze. If the pipe freeze problem were to be resolved, risk of collector-storage freeze damage becomes the limiting factor in freeze risk.

2. NCL MODEL

A room-air NCL is shown schematically in Fig. 4. Warm room air enters the duct on the right, travels up the tube on the inlet side of the divided duct, crosses to the outlet side at the top of the duct near the connection to the passive system, and then descends back into the room. The air becomes colder as it traverses the loop, making the exit column of air colder and heavier than the entering column and providing the net buoyancy force to drive the loop. The outlet is shown with an extension into conditioned space intended to induce a counterclockwise flow, as shown.

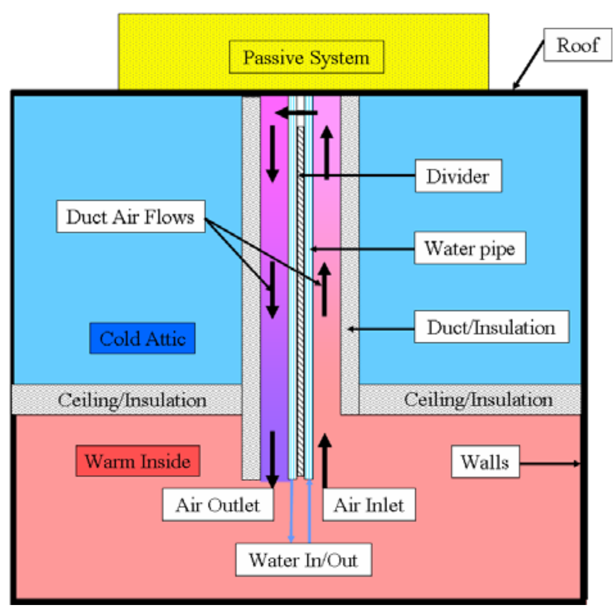

Fig. 4. Schematic of a room-air natural convection loop. Warm room air ascends on the right, and descends on the left side of the divided duct that surrounds the pipes. The NCL could also have air exiting at the top rather than returning to the room (7).

The room-air NCL is modeled as in (8). The driving force for the flow is the difference in density of the two columns of air in the loop (loop buoyancy):

$$P_{buoy} = \oint \rho \vec{g} \cdot d\vec{s} = (\bar{\rho}_{cold} - \bar{\rho}_{hot}) g H \qquad (1)$$

The buoyant pressure P_{buoy} is exactly dissipated by the friction losses around the closed loop:

$$P_{buoy} = P_{fric,shear} + P_{fric,hyd} = ½\rho v^2 * [fL_p/D_h + \Sigma K_i] \qquad (2)$$

Hydraulic losses included here include entry/exit losses for rectangular openings, and one 180 bend ($\Sigma K_i = 4$). The half-circle duct with occlusion of the water tube was modeled as a rectangular duct, using the duct pressure drop correlation in (9). The turbulence-tripping effects of wires and other protrusions were neglected. A friction calibration factor (C_{fric}) is introduced, which multiplies the sum of computed pressure drops. C_{fric} scales computed pressure drops to compensate for the many friction modeling sins. C_{fric} is expected to be greater than 1, due to unmodeled friction. If friction were modeled exactly, C_{fric} would be exactly 1.0.

For calculation of buoyancy, the heat transfer at the pipe wall is modeled similarly to (8). The divider was considered adiabatic. At the outside surface of the pipe, a combined coefficient was used coupled to T_{amb}. The entering, exiting and mean temperatures are related through the log-mean temperature distribution, which applies to the pipe-to-air heat exchanger here. The heat transfer was modeled explicitly in both the entrance and developed regions, as in (8,10). As with friction, it is likely that un-modeled features cause higher heat transfer than is predicted from correlations for simpler geometries. To compensate, a second calibration factor $C_{ht\text{-}trsf}$ is introduced that multiplies the inside wall convection coefficients. $C_{ht\text{-}trsf}$ is expected to be greater than 1. $C_{ht\text{-}trsf}$ would be exactly 1.0 if all heat transfers were modeled exactly. No accounting was made for turbulent mixing or radiation transfer at the outlet, which tend to warm the outlet air for a short distance up the outlet tube. Any solar radiation incident on the piping is not modeled.

The resulting set of coupled equations was solved using the Engineering Equation Solver, which also provides for easy creation of a stand-alone version (11). The stand-alone EES model is available for download on the internet at: ftp://ftp.nrel.gov/pub/solar_waterheat-out/NCL/AIR.EES. The user inputs the pipe/insulation geometry under investigation, assumed room-air and ambient conditions, and the two calibration factors for friction and convection. The flow rate in the duct and temperature at the top and exit of the duct are outputs. The same calibration factors derived below should be used (these are the defaults), unless data are available to calibrate the model for the specific geometry of interest (as is the case here in Section 4 below).

3. PROTOTYPE ROOM-AIR NCL

To test the general concept and provide data for model calibration, a prototype room-air NCL was constructed, as shown in Fig. 5. The NCL was 10 feet high. There was no PSWH at the top of the loop (not needed to test the room-air NCL); instead, a small box was placed at the top to provide room for piping and air flow to turn and return to the interior. The outlet passageway was extended one foot below the ceiling plane to induce the outlet on that side of the air duct. The loop was tested throughout spring, summer, and fall 2005. Three different ducts were constructed from schedule 40 PVC piping of 2”, 4”, and 8” diameters. In each case, a 1” insulation layer with foil covering was applied to the outside of the duct, as was also done for the top box. The 4” and 8” ducts were slit in half, with the duct divider inserted between the halves. The pipes were 3/4” PEX and were not separately insulated.

Thermocouples were located in the air on each side of the NCL at 0’, 3’, 5’, 7’ up the duct, and in the top box at the 10’ position. Each location had a sensor in the center of the air-stream, and a TC fastened to the surface of the PEX pipe. Two rugged RTD-based hotwires were constructed and calibrated in a wind tunnel, as in (10). In addition, a commercial hot-wire (TSI IFA-100) was used for the 8” duct. Flow varied significantly across the aperture. A number of traverses were done with a hot wire on a mounting controlled along two axes to determine total mass flow, and this data was used in model calibration.

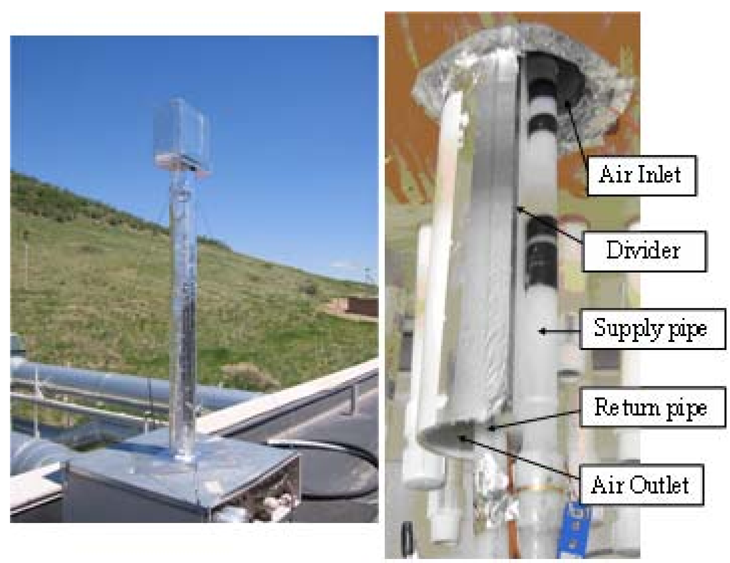

Fig. 5. Prototype room-air NCL. ***Left***: NCL outside (4” pipe with 1” foil-faced insulation). The box at top provides room to make a U-turn with the piping. ***Right***: View of the bottom of the NCL inside the building. The pipes are 3/4” PEX, attached to the divider board which separates the 4” PVC duct into two partitions.

There was no flow induced under moderate freeze conditions for the 2” duct; evidently the passageways presented too much friction to allow measurable flow, even though the simple model predicts flow with any open area. There were 10 total flow measurements on the 4” pipe, and 3 for the 8” pipe. Data were taken only when ambient conditions were not changing rapidly, implying stable flow.

Measured temperature along the flow path are shown in Fig. 6. Generally, temperatures start at room air and decrease in the flow direction, as expected. However, as the flow descends to the outlet, there is some influence from the warmer room. Perhaps turbulent mixing with room air, fan-forced room air currents, and/or infrared radiation exchanges are involved. Although of some interest, this perturbation has a small affect on the results and was not modeled. For the temperature point in the regression in Section 4, the temperature at the top of the loop was used.

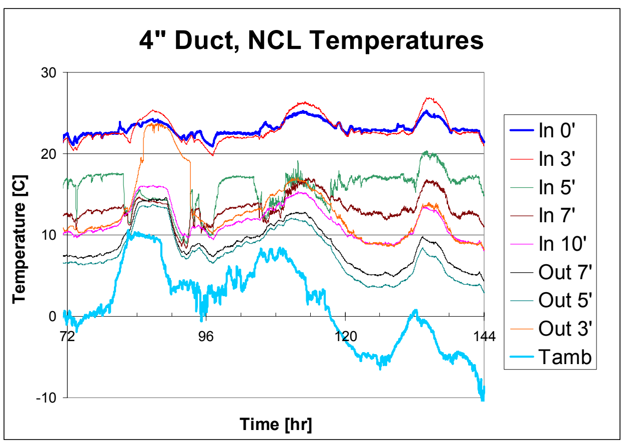

Fig. 6. Variation with temperature along the NCL. The bounding T_{room}, T_{amb} are shown as heavy lines. The temperatures decrease around the loop at night, until the pattern is broken with $T_{OUT@3'}$, which increases due to interaction with the room.

4. MODEL CALIBRATION

The model is calibrated to the prototype data using least squares to determine best-fit values of the two calibration factors C_{fric} and $C_{ht\text{-}trsf}$. The χ^2 metric used for this regression included both flow rate and a temperature:

$$\chi^2 = \Sigma_i\{[(T_{mod,top,i} - T_{data,top,i})/\sigma_{\Delta T}]^2 + [(m_{mod,i} - m_{dat,i})/\sigma_m]^2\}/(N-2) \qquad (4)$$

where i labels the different measured conditions. the modeled quantities T_{mod} and m_{mod} are functions of (C_{fric},$C_{ht\text{-}trsf}$). Contours of χ^2 as a function of (C_{fric},$C_{ht\text{-}trsf}$) are shown in Fig. 7 for the complete 4” diameter data set. The minimum value χ^2_{min} is about 1. The “true” parameters could exist within the χ^2 contour bounded by ~$2\chi^2_{min}$, or about ± 30% of the values. A more extensive data set with a more accurate means of measuring flow, such as with tracer gas, could yield more precise values of (C_{fric},$C_{ht\text{-}trsf}$). Nonetheless, the fit of the model to the data appears reasonable, as shown in Fig. 8 for NCL flow rate.

In the case of the 8” diameter pipe, a similar analysis was performed, yielding (C_{fric},$C_{ht\text{-}trsf}$) = (~8, 2), unfortunately indicating that the parameter adjustments are dependent on diameter. This result is somewhat unexpected. As a result, one should use the model with default calibration factors only for diameters that are reasonably close to 4”, e.g., 3”-6”.

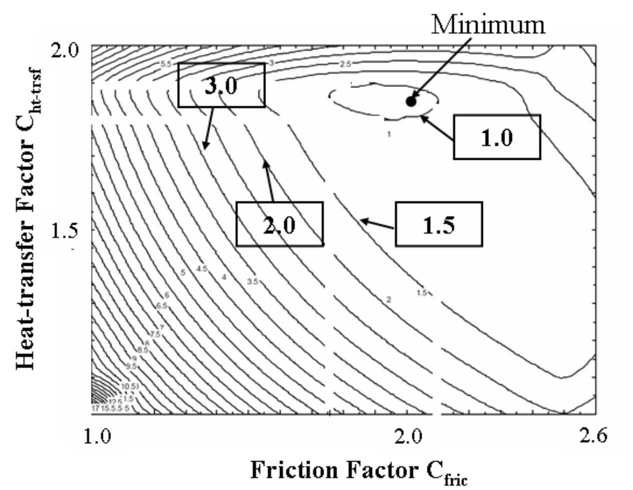

Fig. 7. Contours of χ^2 as a function of (C_{fric},$C_{ht\text{-}trsf}$) for all the data on the 4” diameter room-air NCL. Contour values are shown in the boxes, and the minimum is χ^2_{min} ~ 1, located at (C_{fric},$C_{ht\text{-}trsf}$) = (2.05,1.8).

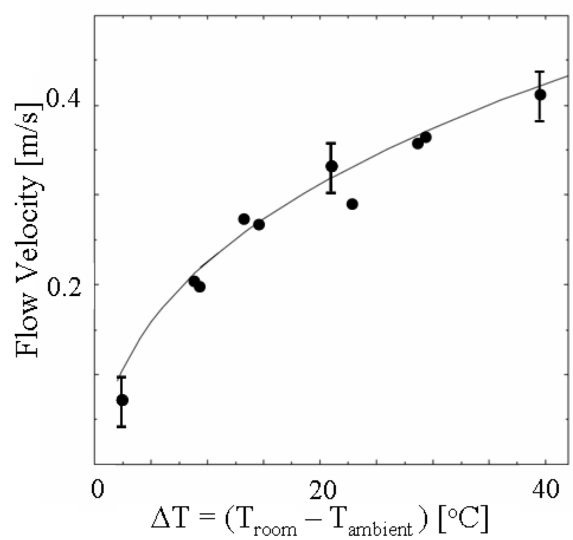

Fig. 8. Average duct flow velocity vs. $\Delta T = T_{room} - T_{ambient}$ Measured flow rates are shown as the solid dots. The model results with (C_{fric},$C_{ht\text{-}trsf}$) = (2.05,1.80) are the solid curve.

5. DESIGN GUIDELINES

The calibrated model can be used to design the geometry of the room-air NCL. The design criterion is that at the record minimum temperature $T_{amb,min}$, the exit temperature of the loop must be above 0 °C. Fig. 9 shows $T_{exit}(T_{amb})$ for the 4" diameter/10' long case. The graph shows that if $T_{amb,min}$ is -10 °C, the insulation must be at least .5". A greater level of insulation than this minimum may be desired to limit the thermal losses, as below.

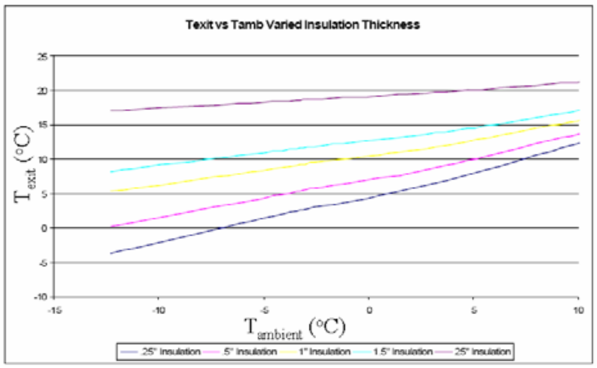

Fig. 9. Exit temperature versus ambient temperature, for a vertical 4" diameter duct 10' long, with insulation thicknesses the curve parameter, with values .25", .5", 1", 1.5", and 2.5" (respectively, from bottom to top curve).

Fig. 10 shows the variation of T_{exit} with total effective length L_{eff} when keeping the vertical height fixed at 10'. [L_{eff} is greater than the vertical height because of horizontal components of the duct run and effective lengths added in due to fittings and bends. If the length becomes too long for a given diameter, the flow will slow and cool down below 0 °C when $T_{amb} = T_{amb,min}$. A 30' length would be just safe at the 4" diameter when $T_{amb,min}$ = -13 °C.

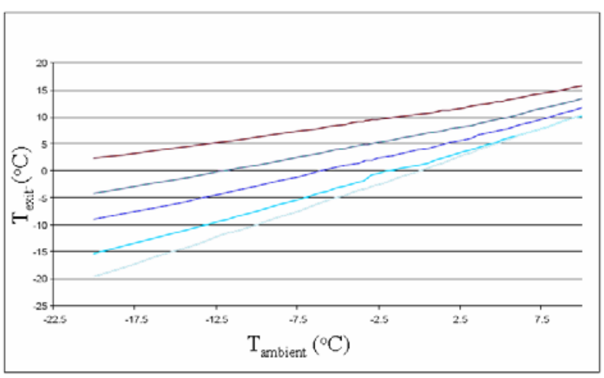

Fig. 10. Exit temperature versus ambient temperature, for a 4" duct with 1" insulation, 10' in vertical height, and lengths of 100', 60', 40', 30', and 20' (from bottom to top curve).

The loop dumps room air heat to the outside, a parasitic loss that must be quantified to assess the viability of the loop. The model was used to compute the parasitic, which depends on temperature difference ($T_{room} – T_{amb}$) in a complex manner. Losses from the NCL were computed for six cities using a bin method (10), with results given in Table 1. Values are given with and without a thermostatic control that stops the flow when the air temperature rises above 1 °C. The parasitic losses with no control range between ~1%-10% of savings. The losses in northern climates could be reduced with added insulation. The control significantly reduce the losses, ranging from nearly 100% reduction in mild climates, but only about 35% reduction in the severest climate. A detailed cost study was not done, but it is estimated that the NCL could be installed for \$100-\$200. The cost advantage of going to a PSWH vs. an active system should be larger than sum of the NCL cost plus the monetary value of the parasitic losses from the NCL plus the monetary value of any decreased savings from going to the PSWH vs. an active system.

Table 1. Room-air NCL Thermal Losses vs. Location.

	Loss, GJ/yr		**Loss, %[1]**	
Location	No cntrl[2]	Cntrl[2]	No cntrl[2]	Cntrl[2]
Albuquerque	0.26	0.07	2.80%	0.79%
Boulder	0.36	0.15	4.58%	1.94%
Memphis	0.24	0.05	3.31%	0.63%
Minneapolis	0.65	0.42	9.84%	6.27%
Phoenix	0.13	0.00	1.39%	0.01%
Sacramento	0.26	0.00	3.40%	0.00%

1. Loss expressed as a % of the savings at that location.
2. "Cntrl" is a damper closing above 1 °C.

6. CONCLUSIONS AND FUTURE WORK

If pipes are protected from freezing, the available geographical for PSWH can be extended northward. A form of primary pipe freeze protection has been investigated, using warm room air circulating in a NCL through a divided duct containing the supply/return pipes of the PSWH. A simple model for the NCL was developed which balances buoyancy and friction, predicting flow rate and NCL temperatures. A prototype room-air NCL was built and tested. The model agreed well with the data from a 4" duct if the model's friction and heat transfer coefficients were roughly doubled [$(C_{fric},C_{ht-trsf}) = (2.05, 1.8)$]. The calibrated model was then used to calculate annual parasitic loss from the NCL and to generate design guidelines.

Future work depends on available funding. Field trials of the room-air NCL are needed to develop practical details and resolve any issues that may arise. Other configurations should be investigated, such as circularly-concentric ducts with water pipes in the inner duct, and return air flow in the outer annulus (12). This configuration will allow lower flow rates than the split duct configuration, while still providing adequate protection. The uni-directional pipe model, where the flow exits into the attic at the top of the loop, should also be developed and validated, and its projected costs and losses compared to bi-directional NCL.

7. ACKNOWLEDGMENTS

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8. NOMENCLATURE

Symbols

C	Scale factor for friction or heat transfer
D	Diameter of air duct
f	Friction factor in Darcy-Weisbach Eqn.
g	gravity
H	Vertical height of natural convection loop
K	Hydraulic friction coefficient
L	Duct length
v	Average velocity in the duct
P	Pressure
ρ	density
χ^2	Metric of deviation between model and data
Δ	Difference
σ	Error
Σ	Summation

Subscripts

amb	Ambient
buoy	buoyancy
cold	Cold side of the NCL
data	data
exit	exit from the NCL
fric	friction
hot	Hot side of the NCL
ht-trsf	Heat transfer
hydr	Hydraulic
i	Index for fittings or data points
min	Record minimum (temperature)
mod	model
room	Room air
shear	Wall shear

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15. SUBJECT TERMS
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