ENERGY AND EXERGY EFFICIENCY OF HEAT PIPE EVACUATED TUBE SOLAR COLLECTORS

by

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In this paper, a heat pipe evacuated tube solar collector has been investigated both theoretically and experimentally. A detailed theoretical method for energy and exergy analysis of the collector is provided. The method is also evaluated by experiments. The results showed a good agreement between the experiment and theory. Using the theoretical model, the effect of different parameters on the collector's energy and exergy efficiency has been investigated. It is concluded that inlet water temperature, inlet water mass flow rate, the transmittance of tubes, and absorptance of the absorber surface have a direct effect on the energy and exergy efficiency of the heat pipe evacuated tube solar collector. Increasing water inlet temperature in heat pipe evacuated solar collectors leads to a decrease in heat transfer rate between the heat pipe's condenser and water.

Key words: *evacuated tube, solar collector, heat pipe, exergy efficiency, energy efficiency*

Introduction

Eliminating conduction and convection heat losses between absorber surface and ambient air is the main advantage of evacuated tube collectors over flat plate ones. Evacuated tube solar collectors have been designed in different models such as U-tube, direct flow, glass-glass, and heat pipe. In the last one, a heat pipe with a two-phase system is used for absorbing and transferring solar irradiance to the fluid which is to be heated [1]. Heat pipe evacuated collectors have a higher thermal efficiency than flat plate collectors, especially in cold climates [2]. Ayompe *et al.* [3] carried out an experimental comparison at similar weather condition within one year, between two different solar water heaters, one with a flat plate and the other with heat pipe evacuated solar collector. The flat plate and evacuated collector generated 496 kWh/m^2 and 681 kWh/m^2 energy per unit area, respectively. Also, annual averages of thermal efficiencies were 46.1% and 60.7%, respectively.

During the past years, the majority of studies on heat pipe evacuated collectors have been carried out mainly in order to evaluate the theoretical and empirical thermal efficiencies of these collectors. Ribot and McConnell [4] conducted a theoretical and experimental research on evacuated collectors with glass heat pipe. Hull [5] studied the thermal efficiency of heat pipe evacuated collectors. Among the important results of this study was that the collectors with less

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than ten tubes are of lower efficiency compared to conventional collectors. Ng *et al.* [6] studied the thermal efficiency of the evacuated tube heat pipe solar collectors and compared the results with experiments. It is worth mentioning that in the presented theoretical model, the correlations of fluid flow in turbulent region have been used. Jafarkazemi and Abdi [7] modeled an evacuated tube heat pipe solar collector with a circular fin and a dry condenser theoretically, and its efficiency and heat gain diagrams are compared with the results of experimental tests. Nkwetta *et al*. [8] compared the thermal performance of a direct flow and heat pipe augmented solar collectors, experimentally. Based on their results, heat pipe augmented system proved better than the direct flow one.

Energy (first law of thermodynamics) approach cannot be a sufficient criterion for thermodynamic systems assessment due to its deficiencies in internal irreversibility analysis and also qualitative analysis of energy flow. Exergy analysis combines the first and second laws of thermodynamics to measure the workability of a system based on entropy production rate and internal irreversibility [9-11]. In other words, exergy serves as a means of providing the possibility of optimizing the quality of energy of the system identifying deviations of the system from the ideal state.

As far as the authors are aware there have been few, if any, in depth research on exergy analysis of evacuated tube collectors and most of the previous works focused on flat plate collectors [12, 13], solar air collector [14], and all glass evacuated solar collector [15]. As mentioned by Kalogirou, *et al*. [16] evacuated tube solar collectors treated less often from an exergy view point. Due to lack of sufficient research, using exergy and 2nd law concept in thermal analysis of heat pipe evacuated collectors, a comprehensive model for these collectors is presented in this work. The model's accuracy is assessed by experimental tests. Using this model, the collector's performance is evaluated from both energy and exergy approaches.

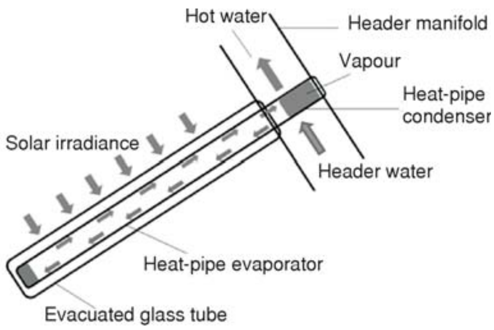

Figure 1. Schematic view of the heat pipe evacuated solar collector

Theoretical analysis

In this research a wickless heat pipe has been used in the evacuated collector. Figure 1 shows the schematic view of the collector and its operation system.

Water was used as the working fluid of the heat pipes. Due to the low pressure inside the heat pipe, boiling point of water will be around 30 °C. The evaporator part of the heat pipe will be heated by solar irradiance. Then, by conduction, this heat will be transferred to the inside surface of the evaporator and vaporizes water. The produced vapor will move to the condenser of the heat pipe. While the header water passes over the condenser, it absorbs the heat and consequently condenses the vapor within the condenser. Due to the gravity in tilted collectors, the condensed vapor goes back to the evaporator part of the heat pipe and the cycle is repeated. In order to achieve a better analysis, we have used the electrical model of the collector with details as mentioned by Jafarkazemi and Abdi [7]. Figure 2 shows this model for heat pipe evacuated collector.

The amount of absorbed solar radiation by the evaporator part of the heat pipe is equal to the optical efficiency of the collector, $I_T(\tau\alpha)$. As shown in fig. 2, a part of the absorbed heat will be dissipated to the surroundings by radiation heat loss. Radiation heat losses between the

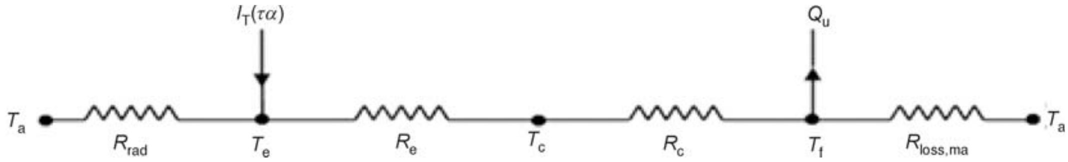

Figure 2. The electrical model for heat transfer in heat pipe evacuated collector

absorber surface and surroundings cause a part of the absorbed heat to be dissipated. This part of heat losses is shown by R_{rad} in fig. 2, and is calculated from the equation:

$$R_{rad} = \frac{T_e - T_a}{\varepsilon\sigma(T_e^4 - T_a^4)A_r} \tag{1}$$

The other part of absorbed solar radiation will be transferred to the inside surface of the heat pipe's evaporator and because of that the inside water will be heated and evaporate. The vapor moves to the condenser at the top of the heat pipe. This leads to the condenser's wall temperature reaching T_c. In fig. 2 the summation of the thermal resistances in the heat transfer path from absorber surface and outer wall of heat pipe to the inside fluid and water evaporation process is characterized by R_e. The R_e is calculated from the equation:

$$R_e = \frac{\ln\left(\frac{D_{o,e}}{D_{i,e}}\right)}{2\pi k_e L_e} + \frac{1}{h_b A_e} \tag{2}$$

As it is obvious in fig. 1, water in the header manifold works as a cooling fluid. Heat will be transferred from the condenser's outer wall to the header water. This heat transfer leads the inside vapor to be condensed. The condensed liquid goes back again to the evaporator or bottom of the heat pipe and the cycle is repeated. This process includes three stages: vapor condensing, conduction heat transfer from condenser inner surface to outer surface, and heat transfer from condenser outer surface to the condenser manifold outer layer. Each of these stages has its own thermal resistance. In fig. 2 the summation of these resistances is shown by R_c and it is calculated from the equation:

$$R_c = \frac{1}{h_c A_c} + \frac{\ln\left(\frac{D_{o,c}}{D_{i,c}}\right)}{2\pi k_c L_c} + \frac{\ln\left(\frac{D_{o,ma}}{D_{i,ma}}\right)}{2\pi k_{ma} L_{ma}} \tag{3}$$

As it seen in fig. 1, the header water flows in the header manifold and absorbs heat from heat pipe's condensers. Undoubtedly some parts of this heat will be transferred into surroundings as the manifold heat losses. In fig. 2 this part of heat losses is marked by $R_{loss,ma}$ and it consists of three terms including conduction heat losses from manifold's inner, middle and outer layers.

Considering the electrical model in fig. 2, the following correlations are obtained:

$$I(\tau\alpha)A_r = \frac{T_e - T_a}{R_{rad}} + \dot{Q}_u + \frac{T_f - T_a}{R_{loss,ma}} \tag{4}$$

$$I(\tau\alpha)A_r = \frac{T_e - T_a}{R_{rad}} + \frac{T_e - T_c}{R_e} \tag{5}$$

$$\dot{Q}_{\mathrm{u}} = \frac{T_{\mathrm{c}} - T_{\mathrm{f}}}{R_{\mathrm{rad}}} - \frac{T_{\mathrm{f}} - T_{\mathrm{a}}}{R_{\mathrm{loss,\,ma}}} \tag{6}$$

Subtracting eq. (5) from eq. (4) leads to:

$$\dot{Q}_{\mathrm{u}} = \frac{T_{\mathrm{e}} - T_{\mathrm{c}}}{R_{\mathrm{e}}} - \frac{T_{\mathrm{f}} - T_{\mathrm{a}}}{R_{\mathrm{loss,\,ma}}} \tag{7}$$

where T_{f} is the header water temperature and it is considered to be the average of inlet and outlet water temperature. The following correlation is also used for the useful energy gain by the working fluid:

$$\dot{Q}_{\mathrm{u}} = \dot{m} c_p (T_{\mathrm{out}} - T_{\mathrm{in}}) \tag{8}$$

Using eqs. (4), (6), (7), and (8), it is possible to calculate T_{out} and $\dot{Q}_{\mathrm{u}}$.

Energy efficiency

Energy (1st law) efficiency of the collector is calculated from the equation:

$$\eta_{\mathrm{en}} = \frac{\dot{Q}_{\mathrm{u}}}{A_{\mathrm{r}} I_{\mathrm{T}}} \tag{9}$$

Exergy efficiency

For calculating the exergy efficiency in thermodynamic systems, it is necessary to find out the exergy sources and exergy users (sinks). In solar collectors,exergy of the Sun's radiation considered as the source of exergy and the header water which is heated in the top manifold is considered as the exergy sink.

Using the thermal reversible system's correlation, transferred exergy in a heat transfer process at a location temperature, T, is calculated from eq. 10 [17]:

$$\dot{E}x_{\mathrm{heat}} = Q\left(1 - \frac{T_{\mathrm{a}}}{T}\right) \tag{10}$$

Exergy of thermal radiation can be calculated using eq. (10):

$$\dot{E}x_{\mathrm{heat}} = A_{\mathrm{p}} I_{\mathrm{T}}\left(1 - \frac{T_{\mathrm{a}}}{T_{\mathrm{s}}}\right) \tag{11}$$

where T_{s} is the temperature of the exergy source and is considered to be 75% of the black body temperature of the Sun which is assumed to be 4500 K [18].

The exergy of the inlet and outlet water are calculated from the equations:

$$\dot{E}x_{\mathrm{w,in}} = \dot{m}_{\mathrm{w}} c_p \left[(T_{\mathrm{in}} - T_{\mathrm{a}}) - T_{\mathrm{a}} \ln\left(\frac{T_{\mathrm{in}}}{T_{\mathrm{a}}}\right)\right] \tag{12}$$

$$\dot{E}x_{\mathrm{w,out}} = \dot{m}_{\mathrm{w}} c_p \left[(T_{\mathrm{out}} - T_{\mathrm{a}}) - T_{\mathrm{a}} \ln\left(\frac{T_{\mathrm{out}}}{T_{\mathrm{a}}}\right)\right] \tag{13}$$

Consequently, the rate of exergy increase in water can be calculated by subtracting eq. (13) from eq. (12):

$$\Delta \dot{E}x_{\mathrm{w}} = \dot{m}_{\mathrm{w}} c_p \left[(T_{\mathrm{out}} - T_{\mathrm{in}}) - T_{\mathrm{a}} \left(\ln \frac{T_{\mathrm{out}}}{T_{\mathrm{in}}} \right) \right] \tag{14}$$

The ratio of absorbed exergy in the exergy sink delivered to the exergy source is the exergy efficiency of the thermodynamic system. Assuming eq. (14) as the exergy sink and eq. (11) as the exergy source leads to the following equation for calculating the exergy efficiency of the solar collector:

$$\eta_{\mathrm{ex}} = \frac{\dot{m}_{\mathrm{w}} c_p \left[(T_{\mathrm{out}} - T_{\mathrm{in}}) - T_{\mathrm{a}} \left(\ln \frac{T_{\mathrm{out}}}{T_{\mathrm{in}}} \right) \right]}{A_{\mathrm{p}} I_{\mathrm{T}} \left[1 - \left(\frac{T_{\mathrm{a}}}{T_{\mathrm{s}}} \right) \right]} \tag{15}$$

Experimental set-up and procedure

A solar collector test laboratory has been developed based on ISO 9806-1 standard [19] to evaluate the results, experimentally. The laboratory is at the solar energy laboratory of Islamic Azad University, South Tehran Branch (latitude 35.70°N, and longitude 51.42°E). Figure 3 shows a schematic diagram of the experimental set-up. The heat pipe evacuated tube solar collector's specifications are shown in tab. 1.

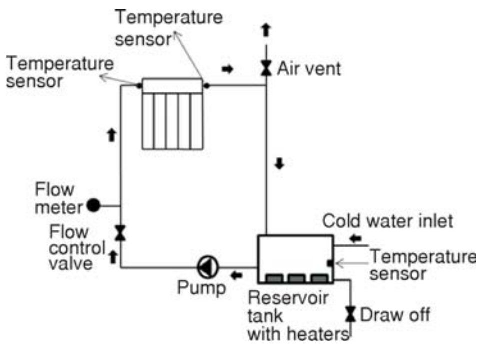

Figure 3. Schematic view of the investigated system

Mass flow rate and inlet water temperature were considered as variables. Mass flow rate of inlet water, inlet and outlet water temperature, incident solar energy per unit area of the absorber surface, and ambient temperature were directly measured by the existing instruments at the test site. The specifications of the system components are described below. The reservoir tank volume is 0.15 m^3, and the tank is made of galvanized steel. Temperature measurement is made by calibrated Pt-100 temperature sensors to measure the inlet and outlet fluid temperatures of the collector and the reservoir tank. A calibrated flow meter with a maximum flow range of 0.04 kg/s is used to measure the water flow rate. The heaters are controlled by a commercial solid state relay controller. A proportional integral derivative temperature controller is used for this purpose. The pyranometer, ambient temperature sensor and wind velocity probe are all calibrated against reference instruments. A data logger with 12 bit resolution is used to log the data every minute. The rated accuracy of measurement instruments is shown in tab. 2.

Table 1. The heat pipe evacuated solar collector's specifications

Parameter	Value
Number of evacuated tubes	4
Aperture area	0.342 m^2
Valid absorption length	1.715 m
Transmission of the glass tube	≥91%
Absorptivity coefficient of the absorber	≥94%
Emissivity coefficient of the absorber	≤7%
Material of absorber pipes	Copper
Effective length of heat pipe evaporator	1700 mm
Outer diameter of heat pipe evaporator	8 mm
Inner diameter of heat pipe evaporator	6.8 mm
Collector tilt	45º
Thickness of insulation in the header	40 mm

Table 2. Test instruments used to measure weather and temperature parameters

Instrument	Accuracy
Temperature sensor	±0.25 °C
Flow meter	±0.0006 kg/s
Anemometer	±0.5 m/s
Pyrnaometers	±3%

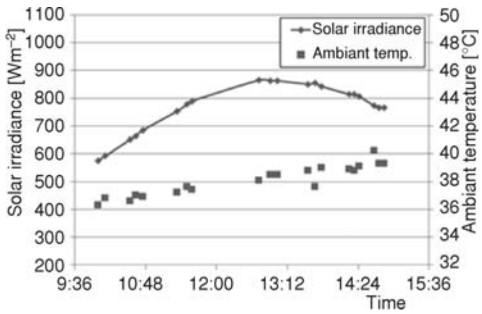

Figure 4. Variations of solar irradiance and ambient temperature during the experiments

Figure 4. shows the variation of solar irradiance and ambient temperature on the day which tests have been done.

Results and discussion

Experimental evaluation of theoretical analysis

Figure 5 and tab. 3 show a comparison between theoretical and experimental values of outlet water temperature. As it is seen the theoretical and empirical results are in a good agreement except in the mid-day. The difference could probably have been improved if there was more time before the data gathering, as this could make the process steadier.

Figure 6 shows a comparison between theoretical and experimental results for energy and exergy efficiencies of the collector. Due to the similar variations of energy and exergy efficiencies in fig. 6 as well as measurement errors, it is possible to confirm the proposed theoretical model within a reasonable approximation.

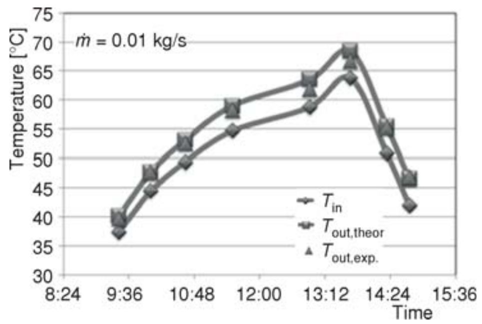

Figure 5. Comparison between theoretical and experimental values of outlet water temperature at $\dot{m}$ = 0.01 kg/s

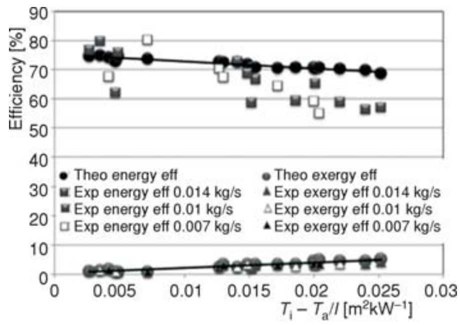

Figure 6. Comparison between theoretical and experimental energy and exergy efficiencies

Table 3. Comparison between theoretical and experimental values of outlet water temperature at $\dot{m}$ = 0.01 kg/s

Time of the day	Inlet temperature [ºC]	Theoretical outlet temperature [ºC]	Experimental outlet temperature [ºC]	Absolute error [ºC]
09:25 a. m.	37.4	40.46	39.7	0.76
10:00 a. m.	44.4	47.74	47.7	0.04
10:38 a. m.	49.4	53.03	52.5	0.53
11:30 a. m.	54.7	58.76	58.3	0.46
00:55 p. m.	58.8	63.20	61.8	1.4
01:40 p. m.	63.8	67.71	66.8	0.91
02:20 p. m.	50.9	55.57	55.3	0.27
02:45 p. m.	42.0	47.04	46.8	0.24

Discussion

Obviously the temperature and mass flow rate of inlet water are the most effective operating parameters in solar collectors. In heat pipe evacuated tube collectors, the heat transfer to the water is mainly done while the water is passing through the manifold at the top of the collector which touches the condenser wall. Consequently the variations of T_c can be considered as one of the effective parameters on collector performance. Figure 7 shows collector's energy efficiency versus $(T_i - T_a)/I$ and mass flow rate of inlet water. As it is seen in this figure, energy efficiency decreases with increasing inlet water temperature. An increase in inlet water temperature leads to a decrease in the temperature gradient between water and the outer wall of the condenser. Therefore, the heat transfer rate between the condenser and the header water will decrease.

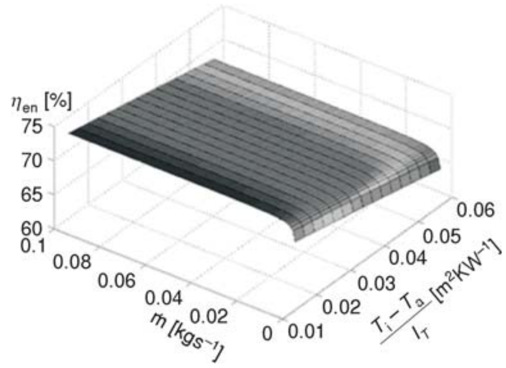

Figure 7. Energy efficiency *vs.* water mass flow rate and ($T_{in} - T_a/I_T$)

As it is obvious in the other axis, an increase in the mass flow rate between 0 and 0.01 kg/s leads to an increase in heat transfer rate between condenser wall and header water. Therefore, the energy efficiency also increases. After that, in higher flow rates the convection heat transfer coefficient between water and the heat pipe's condenser is almost constant. Therefore, the energy efficiency would also be constant.

Figure 8 shows collector's exergy efficiency *vs*. $(T_i - T_a)/I$ and mass flow rate of inlet water. Considering this figure, increasing inlet water temperature leads to an increase in exergy efficiency. The main part of exergy losses in solar collector is due to the temperature difference between the temperature of solar radiation as an exergy source and temperature of the collector's absorber surface. Increasing the water inlet temperature in heat pipe evacuated collectors leads to a decrease in heat transfer rate between the heat pipe's condenser and water. Consequently T_c increases. As it is seen in fig. 2, T_c and T_e have a direct correlation so any increase in T_c leads to an increase in T_e. An increase in the absorber's surface temperature leads to a decrease in the temperature difference between the absorber and temperature of solar radiation. As a result of this, exergy losses will decrease. As it has been mentioned, an increase in the mass flow rate between 0 and 0.01 kg/s leads to an increase in heat transfer rate between condenser wall and header water. The T_e will also increase. Consequently, the exergy efficiency of the collector will decrease. At higher flow rates, constancy of the convection heat transfer coefficient between water and the heat pipe's condenser leads to a stabilization of exergy efficiency.

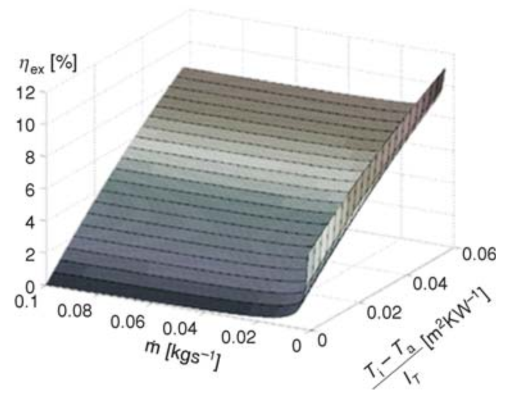

Figure 8. Exergy efficiency *vs.* water mass flow rate and ($T_{in} - T_a$)/I_T

Conclusions

As far as the authors are aware there have been few if any, in depth research on exergy analysis of heat pipe evacuated tube solar collectors and most of the previous works focused on the flat plate collectors. Due to lack of sufficient research, using exergy and 2nd law concept in thermal analysis of the heat pipe evacuated collectors, a comprehensive model for these collectors was presented in this work. The model's accuracy was assessed by experiment.

While increasing the difference between water inlet and ambient temperature leads to a decrease in energy efficiency, it leads to an increase in exergy efficiency. The same process is observed about the variations of water inlet mass flow rate. It is also concluded that increasing water inlet temperature besides decreasing water mass flow rate results in a better exergetic performance.

Absorbing more solar radiation leads to better performance from both energy and exergy approaches. Making an improvement in the transmittance of tubes and absorptance of the absorber surface has a direct effect on collector performance.

Acknowledgments

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Nomenclature

A	– area, [m^2]
A_r	– solar collector aperture area, [m^2]
A_p	– solar collector absorber area, [m^2]
c_p	– heat capacity of fluid, [Jkg^{-1}K^{-1}]
D	– diameter, [mm]
$\dot{E}x$	– exergy flow rate, [W]
h	– convection heat transfer coefficient, [Wm^{-2}K^{-1}]
I_T	– solar irradiance, [Wm^{-2}]
k	– thermal conductivity, [Wm^{-1}K^{-1}]
L	– length, [m]
$\dot{m}$	– fluid mass flow rate, [kgs^{-1}]
$\dot{Q}$	– heat transfer rate, [W]
R	– thermal resistance, [KW^{-1}]
s	– specific entropy, [kJkg^{-1}K^{-1}]
T	– temperature, [°C]
W	– water, [–]
U	– heat loss coefficient, [Wm^{-2}K^{-1}]

Greek symbols

ε	– emissivity, [–]
σ	– Stefan-Boltzmann constant, [Wm^{-2}K^{-4}]
$\tau\alpha$	– effective transmittance – absorptance product of absorber

Subscripts

a	– ambient
b	– boiling
c	– condenser
e	– evaporator
f	– fluid
i	– inside
in	– inlet
ma	– manifold
o	– outside
out	– outlet
rad	– radiation
T	– total
u	– useful
w	– water
0	– dead (reference) state

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