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Theoretical multi node numerical model for thermal solar collector with forced circulation

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ABSTRACT: The Thermal analysis of flat plate solar water heater is a complex task, because of a large numbers of parameters affecting its performance. In the last years, a lot of studies have been developed, using numerical approaches to describe and analyze the behavior and thermal mechanism of the thermal solar collector. In this paper, we develop a numerical model of the flat plate collector in forced circulation regime. We describe the basis of the model and show the comparisons with other bibliographic model. The main goal of this work is to simulate the efficiencies of the solar collector using an "4n-node model". The results are giving a very good agreement with the literature. The model is used to examine the impact of different parameters, which may affect the performance of the solar collector with single glass cover.

KEYWORDS: Solar collector; Simulation; forced convection.

1 INTRODUCTION

The energy demand is continuously increasing due to the acceleration of demographic increase and the economic development of emerging countries that knows the world during last years. Renewable energy has a beneficial impact on the economic and environmental policy and it's the key of the future development of our planet.

Soteris [1] recently demonstrated that the renewable energies contribute to the new job creation, the energy security and reduction of environmental pollution. Moreover, solar energy is considered one of the promising energy sources to meet the dependency on fossil energy [2-3]. Flat plate collectors are very useful systems in countries with high potential for solar radiation, as is the case of Morocco, which belongs to the Mediterranean region [4].

Many configurations are be used in numerical modeling of solar flat plate collectors. The numerical models used to find solutions without the need to build a series of prototype, once the solutions are found they should be validated experimentally. Various exertions have been made aims to develop models that contain many geometric and physical complexities.

The goal of this paper is to prove how the simulation is able to predict the parameters, the output temperature, the energy produced and the instantaneous efficiency of the flat plate solar collector. For that we configured a theoretical model such as the main theoretical elements of the model are described and enough detailed and compared with a model of the bibliography, the explanation includes a conceptual description of useful solar system in our models

2 BACKAROUND

Many numerical models being developed in the last years to describe the phenomenon and the thermal performance of the flat plate solar collector. The first initiative to implement a dynamic numerical model of the solar collector was made by Close [5] who has took into consideration the thermal capacity of the absorber, as the sum of the other thermal capacity of various elements (glass, fluid) as it is shown in the fig.1, this model is named by " 1n-node model", in the same direction Klein and al [6] changed the global model taking into account the variation of the temperature of the fluid along the solar collector.

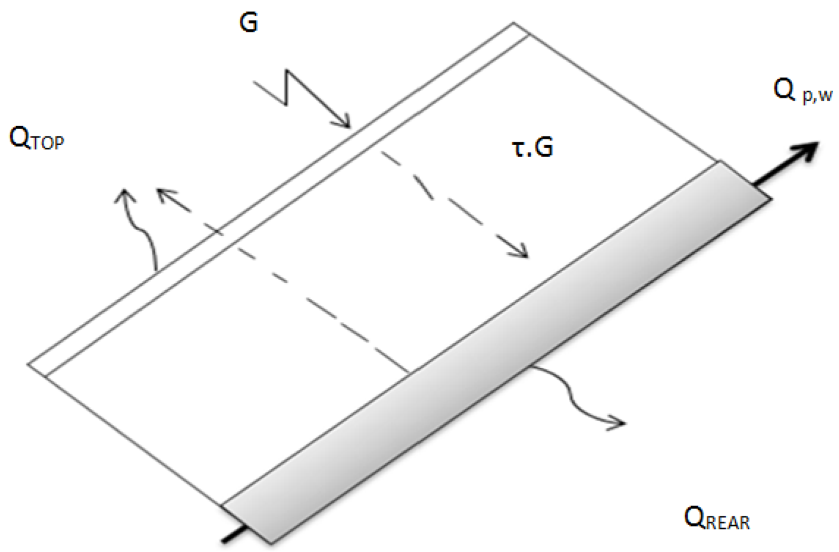


Fig.1: Energy balance of a thermal solar collector in steady state

Moreover, the authors suggested to add other nodes positioned in the transparent over of the collector. De Ron [6] developed a model of solar collector which he took into account conduction heat transfer is essential1-Dimension. In connection with the suggestion proposed by Klein [7], the author sets up one equation of the energy balance, in particular for the transparent cover and the absorber admitting that:

- The heat capacity of the air gap is neglected
- The back and collector's side are adiabatic

This model is known as the" 2n-node model" as it is exposed to the fig 2.

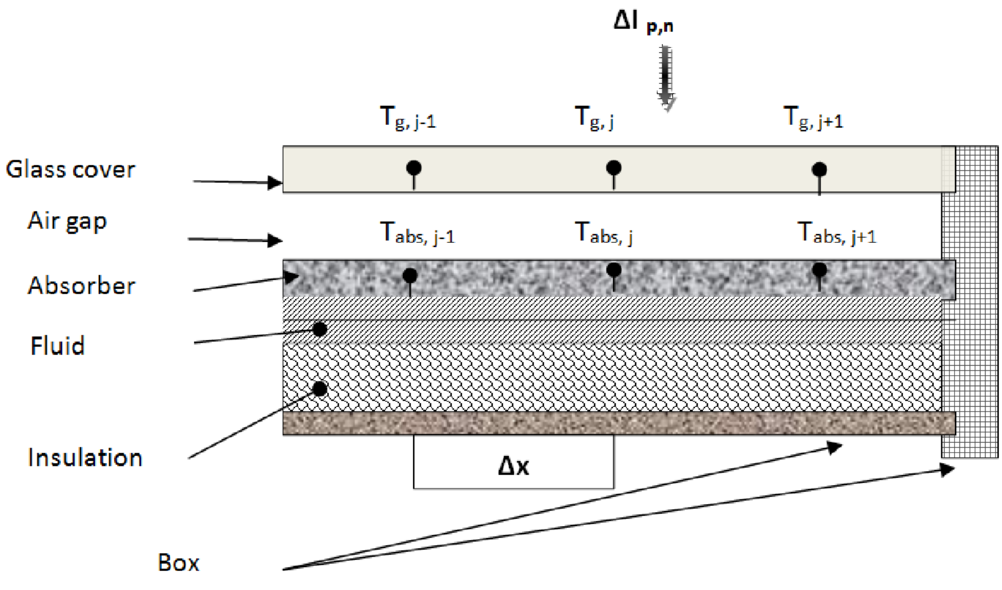


Fig. 2: The basic 1n-node schema with $T_{p,j}$ is extended with further 1n-node with $T_{g,j}$ on the collector glass cover, realizing a 2n-node model

In the same area, Wijeysundera [8] developed the model''2n-node model'' where the solar collector is modeled as two zones, one corresponds the absorber and the fluid and the second relate to the transparent cover. Moreover, the author showed that the model approach"2n-node" gives good results in the prediction of the solar collector parameters. Kamminga [9] considered the capacity of the fluid, together with those of the cover and the absorber to develop the model ''3n-nodemodel''as it is shown in the fig. 3. The transfer of heat along flows is due only by the movement of the fluid (conduction is neglected), also Morrison and Ranatunga [10] developed a model "3n-node model" differencing between the thermal capacity of the fluid and the absorber, they set up three characteristic equations, in particular of the transparent cover, absorber and of the fluid.

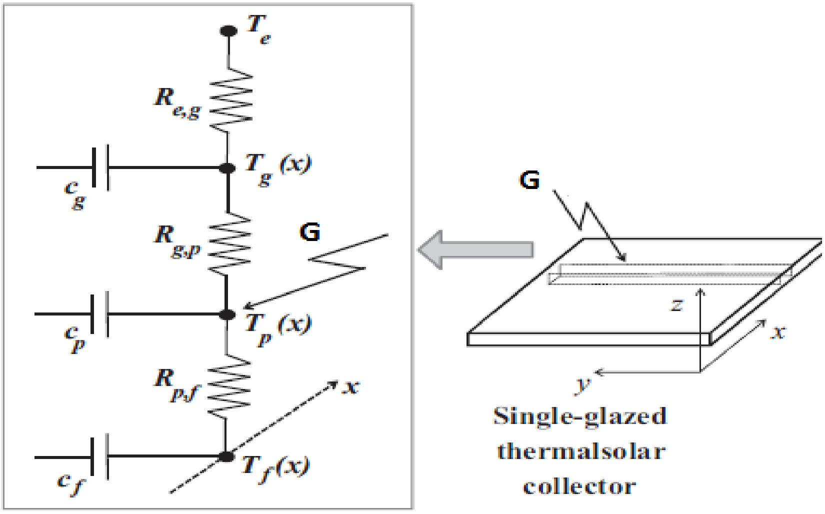


Fig.3: Electrical analogy of a 3n-nodes model of a solar thermal collector [11]

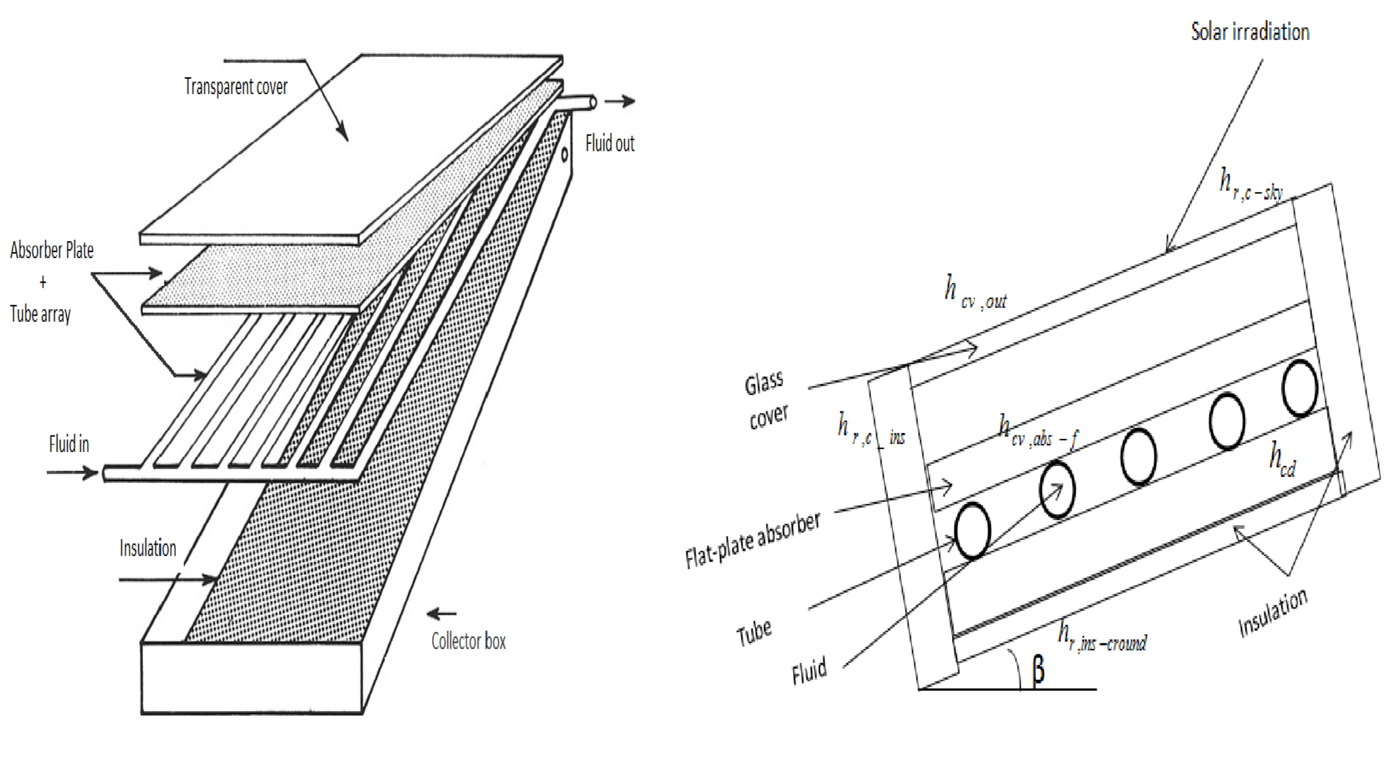
Moreover, the addition of the thermal capacity of the fourth element (back insulation) drives to the implementation of a model "4n-node model ". According to the previous work, the authors applied the model "4n-node model" taking into account the thermal inertia of the components of the solar collector (glass, absorber, fluid and insulation).

In [12-13] an algorithm DFA(Dynamic Fitting Algorithm) was used, developed by spirkl [14-15] to identify the principal parameters of the solar heat thermal collector (effective area, loss coefficient, etc). The collector discredited in nodes and each one of them associate with an equation of the energy balance, in order to calculate the temperature of the fluid. Cadfalch [16] presented a detailed numerical approach which can be considered as an extension of the dynamic model 1n-nodes developed by [17]. These models are translated by a set of partial differential equations which can be simplified and resolved by transforming them to obtain differential equations ordinary and by using a numerical method to solve it, such as the method of Rang Kutta 4th order [18-19].

3 THE 4N-NODE MODEL:

The present model is based as the other models on mass and energy conservation laws. The fig.3.a shows the geometry of our model which contains essentially, a single glass, absorber, fluid and insulation.

The convection exists between the solar collector and the atmosphere which surrounded it, also it exists inside the collector, between the fluid and the absorber.



a) b)

Fig.4: Flat-Plate solar collector geometric, subject to the solar and thermal analysis

The radiative exchange was made between, the glass cover and the sky and the glass cover with the absorber, and finally between the collector and the ground. Conduction is dominant at the rear losses through the insulation which is modeled as a multiple layer (Fig.3.b).Before starting calculations, it is necessary to specify certain calculation assumptions:

- Irradiation is uniform for all receiving surfaces of the solar collectors
- The ambient temperature is the same around the collector
- The physical properties and thermal fluid data are based on the average temperature
- The heat transfer fluid is incompressible and doesn't undergo change of state

3.1 HEAT BALANCE OF THE TRANSPARENT COVER:

The energy comes from the sun in the form of the rays solar penetrate in the solar collector passed by the transparent cover of the solar collector. The Eq. 1 shows the heat balance of this piece (Eq.1):

$$(m.Cp)_g.\frac{dT_g}{dt} = (\alpha_g.G).S + h_{r,g-sky}.S(T_{sky} - T_g) + (h_{r,g-ground}. + h_{cv,out}).S(T_{am} - T_g) + h_{r,g-abs}(T_{abs} - T_g) \qquad \text{(Eq.1):}$$

Where:

The transfer coefficient $h_{cv,out}$ given by the empirical relationship given by Adams and Woertz.

$$h_{cv,out} = 5{,}67 + 3{,}86.V_w \qquad \text{(Eq.2)}$$

The coefficient of radiation transfer between the glass cover and the sky [17-20]:

$$h_{r,g-sky} = \varepsilon_v \sigma (T_g + T_{sky})(T_g^2 + T_{sky}^2) \qquad \text{(Eq.3)}$$

The temperature of the sky is correlated by the formula [17]:

$$T_{sky} = 0{,}0552.T_{am}^{1,5} \qquad \text{(Eq.4)}$$

Also the coefficients of transfer by radiation respectively between glazing and the ground and the transparent cover with the absorber are given by [17-20]:

$$h_{r,g-ground} = \varepsilon_g \sigma (T_g + T_{ground})(T_g^2 + T_{ground}^2) \qquad \text{(Eq.5)}$$

$$h_{r,g-abs} = \frac{\sigma (T_g - T_{abs})(T_g^2 - T_{abs}^2)}{\frac{1-\varepsilon_g}{\varepsilon_g} + 1 + \frac{1-\varepsilon_{abs}}{\varepsilon_{abs}}} \qquad \text{(Eq.6)}$$

3.2 HEAT BALANCE OF THE ABSORBER

The absorber receives energy coming from the solar radiation and at the same time, it transfer this energy to fluid, knowing that a part of this energy is lost because of heat exchange by convection with the glass cover and by conduction with the insulator (Eq.7)

$$(m.Cp)_{abs} . \frac{dT_{abs}}{dt} = (\alpha_{abs} .\tau_g \alpha_g .G).S + h_{r,g-abs} .S(T_g - T_{abs}) + h_{cv,abs-f} .S(T_f - T_{abs}) + h_{cd.,abs-ins} .S(T_{ins} - T_{abs})$$

$h_{cv,abs-f}$ Represent the coefficient of transfer by forced convection between the absorber and the fluid, it was calculated according to the empirical relations given by the works of Haussen and Sider-Tate.

$$h_{cv,abs-f} = \frac{Nu.\lambda_f}{D} \qquad \text{(Eq.8)}$$

Where λ and D are the thermal conductivity and the hydraulic diameter of the fluid respectively. The Nusselt number, *Nu*, is given by the following relation [18-19]:

$$Nu = 3{,}66 + \frac{0{,}085.Gz}{1 + 0{,}047..Gz^{\frac{2}{3}}} \qquad Gz< 100 \quad (Re< 2100) \qquad \text{(Eq.9)}$$

$$Nu = 1{,}86.Gz^{\frac{1}{3}} + 0{,}87(1 + 0{,}015.Gr^{\frac{1}{3}}) \qquad Gz> 100 \quad (Re< 2100) \qquad \text{(Eq.10)}$$

$$Nu = 0{,}116(\mathrm{Re}^{\frac{2}{3}} - 125).\mathrm{Pr}^{\frac{1}{3}}.(1 + \frac{D}{N.L})^{\frac{2}{3}} \qquad 2100< Re< 10000 \qquad \text{(Eq.11)}$$

3.3 HEAT BALANCE IN THE FLUID

The fluid is in exchange thermal only with the absorber; it is a transfer of heat by forced convection.

$$(\rho.V.Cp)_f . \frac{dT_f}{dt} = h_{cv,abs-f} .S(T_{abs} - T_f) \qquad \text{(Eq.12)}$$

3.4 HEAT BALANCE OF THE INSULATION

Heat exchange concerning this element is those between insulator with the absorber and also with the ambient.

$$(m.Cp)_{ins} . \frac{dT_{ins}}{dt} = h_{cd,abs-ins} .S(T_{abs} - T_{ins}) + (h_{cv,out} + h_{r,ins-ground}).S(T_{am} - T_{ins}) \qquad \text{(Eq.13)}$$

Because of the complex structure and multiplicity system of equations characterize our model, the method of Rang-Kutta 4th order [18-19] was adopted for the numerical solution of this system of equations.

The instantaneous efficiency of flat plate collectors is defined as the report of useful energy on incident energy of the solar radiation multiplied the collector surface:

$$\eta = \frac{Qu}{G.S} \quad \text{(Eq.14)}$$

Where:

$$Qu = \dot{m}.C_p(T_f - T_{am}) \quad \text{(Eq.15)}$$

A lot of parameters influenced on the instantaneous efficiency, as the building materials of the collector, the geometry of absorber, the properties of the transparent cover, the weather conditions and the position of the collector, it can be represented by the relation [23-24]:

$$\eta = F_R(\tau\alpha) - F_R U_L \frac{(T_f - T_{am})}{Ps} \quad \text{(Eq.16)}$$

Where, the term $F_R(\tau\alpha)$ expressed how the energy of the incident solar radiation is absorbed and the term $F_R U_L$ determine how much energy is lost.

4 Results And Discussion

This section is devoted to the comparison between our numerical model and Klein model [28]. We treat the curves, values together with the errors. The characteristics of the site and the parameters used in the simulation are given in Table 1.

Table.1: Characteristics of the site used in the numerical simulation

Parameters	Value	Unit
Latitude	34,15	degrees
Longitude	-6,58	degrees
Wind speed	1,5	m/s

Fig.4 shows the comparison between the efficiency curves presented by model of Klein [25] and that of the present study. As it is shown in the fig. 4, the curve of the present study has a slope less than that of Klein [24].

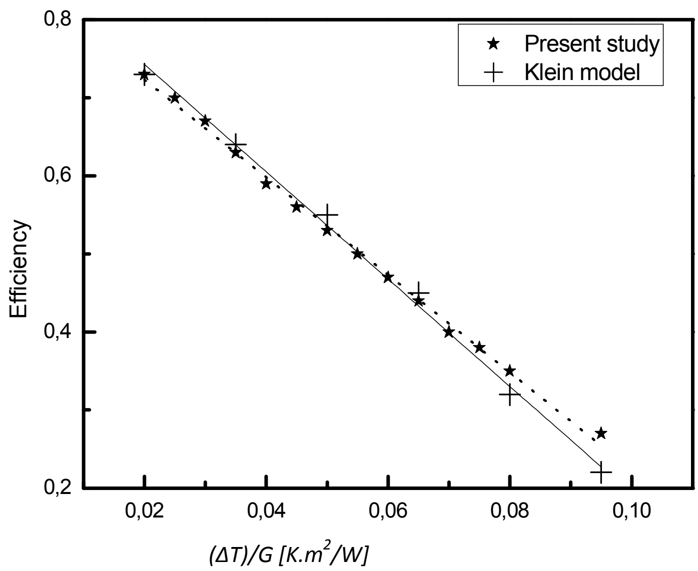


Fig 5: Instantaneous efficiencies of solar collector obtained by the present study and by the method of Klein versus (T_{abs}-T_{am})/G

The lower slope means smaller - $F_R U_L$ = - 6,9 for our study and -6,2 for Klein model [25] and this means smaller heat losses from the collector. In reality, the losses of the solar collector are due to a combination between the transfer by radiation and transfer with convection [26].In particular the fig. 4 shows that the curve of instantaneous efficiency is higher than that of [24] in the region of high temperatures, this means that the present collector can work in high temperatures. In other hand, the fig.5 [26] proved that the model of the present study illustrate in the fig.4 situated well in the regions corresponding the flat plate solar collector a single glass.

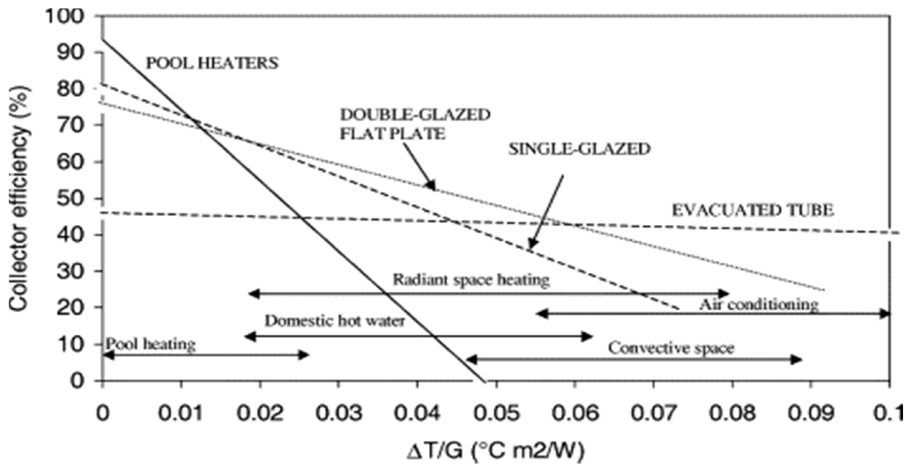


Fig.5: Collectors efficiencies of various liquid collectors [27]

5 CONCLUSION

The paper presents a detailed model of multi-node (4n-node model) for flat plate solar collectors a single glass cover. Moreover, the model results were compared to previous results bibliographic; it was demonstrated that there is a very good agreement between them.

The profile of the curve of the yield obtained by this model was found not linear in shape, also it is noted that the efficiency of the solar collector decreases according to the increase of the ratio between the different temperature and solar irradiation. In addition to this, it is shown that the model has been well situated in the area corresponding the solar collector of same type.

It should be noted that the experimental validation of this model will be publish in other separated articles.

NOMENCLATURE

Cp	specific heat at constant pressure (J/kg.K)
D	tube diameter (m)
F	collector factor (dimensionless)
G	solar radiation (W/m^2)
Gr	Grashoff number
Gz	Graetz number
h_{cv}	convection heat transfer coefficient (W/m^2. K)
h_{cd}	conduction heat transfer coefficient (W/m^2. K)
h_r	radiation heat transfer coefficient (W/m^2. K)
L	lengthof thetube (m)
N	number of tube
Nu	Nusselt number
Pr	Prandtl number
Q_u	useful energy (W)
Re	Reynolds number
R	Thermal resistance (K.W^{-1})
S	surface collector
T	temperature (°C)
U	heat coefficient (W/m^2.K)
V	volume (l)
V_w	Wind speed (m/s)

Greek letters

λ	thermal conductivity (W/m.K)
ρ	density (Kg/m^3)
α	absorbance coefficient
τ	solar transmittance coefficient
ε	emissivity
δ	Stephan –Boltzmann coefficient
η	Instantaneous efficiency

subscripts

abs	absorber
am	ambient
e	external
f	fluid
g	glass
ins	insulation
L	overall
P	plate
R	heat removal
w	water

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