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CALCULATION MODEL FOR PROGRESSIVE SPIRAL HEAT EXCHANGERS

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Abstract: *Common types of geothermal exchanger both the surface and the depth is characterized by uneven soil application. Uniform thermal load of the massive earth could be an energy optimization in the sense of optimizing storage capacity and therefore reduced surface / volume of land used.*
Less expensive additional measures such as homogenization and modification insulation material storage enclosure and heat exchanger can be obtained within limits determined on equivalent in terms of heat sources similar to the vertical.
The following is the constructive principle and calculation method.

Key words: *geothermal exchanger, variable spatial geometry, calculation model.*

1. Introduction

Common types of geothermal exchanger both the surface and the depth is characterized by uneven soil application. Uniform thermal load of the massive earth could be an energy optimization in the sense of optimizing storage capacity and therefore reduced surface / volume of land used [1].

In this idea proposing one solutions for achieving its original surface heat exchangers, release / take- uniform heat throughout the system. Exchangers are designed modular elements having variable diameter or length respectively and thermal load evenly distributed.

One solution atypical geothermal heat exchangers, shallow, variable spatial geometry using cylindrical or tapered. By comparison with the probes/or pilots Geothermal wells solution presents significant from the point of view of recovery of the thermal capacity of the soil.

If geothermal probes and heat transfer is directly proportional uneven drilling depth and temperature difference, corresponding variable amount of heat transferred / taken from agent working agent circulation loop length work, equipping the well [2].

In the case of surface heat exchanger with cylindrical geometry coil, the phenomenon is similar to the observation that the spiral shape of the circulation pipe, the equivalent depth required is significantly reduced compared to vertical wells. For pipe coils made with constant diameter is maintained disadvantage loading / unloading uneven ground.

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The situation is radically altered when using spiral geometry of tapered or cylindrical modular loading surface to which the transfer is directly proportional to the reduction temperature of the work and its evolution, leads to a transfer of that charge / discharge uniformly [3].

2. Model Calculation

The exchanger consists of semicircular elements enveloping a tapered surface.

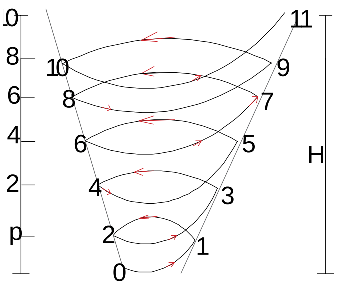

Fig. 1. *Progressive spiral heat exchanger*

Length of modules is determined by the principle of equal quantity of heat transferred / received by each element so that the high demands of earth to be uniform [4].

The final dimensions result from the total heat load (Q), the number of elements (n), soil characteristics and parameters of thermo geometry, small base diameter and up spiral .

Suitable principles**,** specific failure is constant [5]:

$$q = \frac{Q}{n} = KL_i(T_i - \theta) = const. \qquad (1)$$

where: K [w /m^2 K] = global coefficient of heat transfer exchanger.

n= number of modules, n=$\frac{H}{p}$.

H - total height

p- step spiral

$T_i = \frac{T_{i1} + T_{i2}}{2}$ - average temperature of the work on the section considered.

θ=considered constant soil temperature during the process.

For water- ground heat pumps, reversible temperature difference $\Delta T = T_{tur} - T_{retur}$, which should be done instead [6].

Seasonal difference is 15^{o}C -20^{o}C cooling mode in heating mode .

The uniform Representative (n) elements, resulting Q for each mode, the difference between the inlet temperature (T_{i1}) and output (T_{i2}) is the same:

$$\{\Delta T = T_{tur} - T_{retur} = T_{11} - T_{12} \qquad (2)$$

For the first element in length (L_1) required outlet temperature will be:

$$\begin{cases} T_{12} = T_{11} \pm \delta \\ \overline{T}_1 = \frac{1}{2}\left[T_{11} + T_{12}\right] = T_{11} \pm \frac{\delta}{2} \end{cases} \qquad (3)$$

The observation that the modules are connected in series, for entering into an element is equal to the temperature out of the previous item [7].

Therefore the measurement of success for the last one can write relations:

$$\begin{cases} T_{n1} = T_{11} \pm (n-1)\delta \\ T_{2n} = T_{11} \pm \delta = T_{11}(n-1) \pm \delta \qquad \textit{and} \\ \overline{T}_n = \frac{T_{n1} + T_{2n}}{2} = T_{11} \pm (n-1)\delta \pm \frac{\delta}{2} = T_{11} \pm \left(n - \frac{1}{2}\right)\delta \end{cases} \qquad (4)$$

From equation equivalency spread the heat transferred / received results:

$$KL_1 = (\overline{T}_1 - \theta) = KL_n\left(\overline{T}_n - \theta\right) \qquad (5)$$

Accordingly: $L_1 = \frac{\pi D_1}{2} = \frac{\pi p}{2\sin\beta}$ and

$$L_n = \frac{\pi p}{2\sin\beta} - \frac{\overline{T_1}-\theta}{\overline{T_n}-\theta} \Rightarrow L_n = \frac{T_{1p}}{2\sin\beta}\,\frac{\left(T_{tur}\pm\frac{\delta}{2}\right)-\theta}{T_{tur}\pm\left(n-\frac{1}{2}\right)\delta-\theta} \quad (6)$$

Substituting in (6) according to average temperatures (3) and (4) resulting length of the second module (Ln) depending on the assumptions adopted:

$$L_n = L_1\left[\frac{T_{11}\pm\frac{\delta}{2}-\theta}{T_{11}\pm\left(n-\frac{1}{2}\right)\delta-\theta}\right] = L_1\left[\frac{(T_{11}-\theta)\pm\frac{\delta}{2}}{(T_{11}-\theta)\pm\left(n-\frac{1}{2}\right)\delta}\right] =$$

$$L_1\frac{1\pm\frac{\delta}{2}\Big/(T_{11}-\theta)}{1\pm\left(n-\frac{1}{2}\right)\delta\Big/(T_{11}-\theta)} \quad (7)$$

The final configuration of the virtual volume corresponds to a given pitch (p) and a required number of modules inserted (n) is determined by the characteristic angles [8].
α- between α frustocone generator vertical and horizontal elements β-between diameter.
The apparent diameter L value determined last way:

$$D_n = \frac{2L_n}{T_1} \quad (8)$$

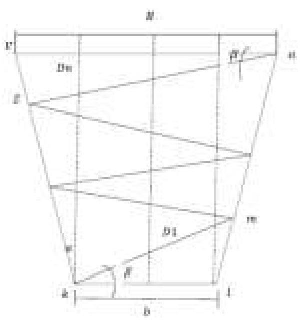

Fig. 2. *The middle section of the truncated cone.*

The middle section of the truncated cone generator results:

$$B = b \pm 2(mp)tg\alpha \quad (9)$$

The similarity of triangles:

$$\frac{\overline{Kl}}{\overline{uv}} = \frac{Km}{\overline{uz}}, \frac{b}{B} = \frac{D_1}{D_n} \Rightarrow B = b\frac{L_n}{L_1} \quad (10)$$

Substituting:

$b + 2(np)tg\alpha = b\frac{L_n}{L_1}$, where:

$$tg\alpha = \frac{b}{2np}\left(\frac{L_n}{L_1}-1\right) \quad (11)$$

Specifying Ln under (6) obtaining final form:

$$tg\alpha = \frac{b}{2np}\left[\frac{1\pm\frac{\delta}{2}\Big/(T_{11}-\theta)}{1\pm(n-1)\delta/(T_{11}-\theta)}\right] \quad (12)$$

Defining the geometry changes depending on parameters known, with meanings:
b- small base
p- step coil
n- number of modules
δ- spread extreme temperatures on the way
T11 = T tur -agent working temperature entry into exchanger
θ- massive ground temperature.

In relation computing intervening algebraic sum sign (-) applicable under the air conditioning / cooling and sign (+) for the heating.

Calculated for both hypotheses, adopting values covering.

3. Conclusions

In terms of functional and energy, the solution is obvious solutions superior to any surface or deep usual, the heat transfer in the heat exchanger is variable .

Heat flow of heat transferred / received, remain constant at each level corresponding transfer surfaces change in proportion to the temperature variation spread.

The particularity of the solution presented are favorable argument optimize surface geothermal exchangers, used in hybrid realization of heating / cooling using geothermal and solar energy.

Leveraging the advantages mentioned, it is possible to reduce the amount of land used for seasonal storage of energy.

Less expensive additional measures such as homogenization and modification insulation material storage enclosure and heat exchanger can be obtained within limits determined on equivalent in terms of heat sources similar to the vertical.

The following is the constructive principle and calculation method.

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