

Influence of structural design condensing part of NH₃ heat pipe to heat transfer

Martin Vantúch^{1,a}, and Milan Malcho¹

¹University of Žilina, Faculty of Mechanical Engineering, Department of Energy Technology, Univerzitná 1, 010 26 Žilina, Slovak Republic

Abstract. The article describes influence design heat exchangers to efficiency condensation liquid ammonia in the gravitational heat pipe. Analyse adverse factors in the operation and flow of ammonia in heat pipe. Also describes heat transfer characteristics of heat pipe in low-potential geothermal heat transport simulations.

1 Introduction

Ground source heat pumps which use vertical heat exchangers do not require large areas for installation and not depend on intensity of solar radiation that reaches surface of the earth. Vertical heat exchangers work effectively in virtually all geological environments, except for the soil with low thermal conductivity, such as dry sand or gravel.

When the gravitational heat pipe (GHP) is used as a vertical heat exchanger, heat transport takes place in phase change of working medium. Heat pipes now generally used in solar collectors come to forefront in scope of cooling, transmission as well as the acquisition of geothermal heat. During the last decade of refrigerant as working substance in GHP, our research has focused mainly on the working substance NH₃.

Heat exchanger on condensation section of the heat pipe is one of the most important parts of the equipment. The heat gained by GHP is collected in this part, because its structure must be designed to transfer heat produced with the best performance possible.

2 Proposal of the heat pipe heat exchanger

In solving condensation of vapor on the wall at heat exchanger laminar flow condensation we assume that:

- temperature of free surface layer is the same as temperature of steam substance t_v ,
- condensate temperature at a wall spiral pipe is equal to temperature of a wall, t_b, C ,
- density of vapor is to density of a liquid phase negligibly small,

- heat transfer in condensate layer is determined only by heat conduction in direction perpendicular to the wall HP.

Thermal resistance of vapor-liquid interface, which depends on difference in the number of vapor molecules absorbed to surface of colder fluid and released to surface in a steam space-thermal resistance is compared to the condensate layer negligible.

Of the said it results that the operation temperature of condensate layer is given by equation:

$$\bar{t}_L = t_{i,C} + \frac{\Delta t_{i,C}}{\delta_x} y, \quad (1)$$

where

$$\Delta t_{i,K} = t_p - t_{i,K}. \quad (2)$$

The heat exchanger was designed and manufactured from stainless steel (AISI 304), which is resistant to corrosion and does not react with ammonia. It is cylindrical in shape. Inside the cylinder there is a spiral pipe in which flows the medium (50% mixture of ethylene glycol) of primary heat pump circuit. The spiral is rotated with a uniform pitch so that a face of each individual thread touches. The bottom and top of the spiral is ended by threaded part for connection to heat pump primary circuit. This spiral is bathed by vapor working fluid (ammonia) of heat pipe, which condenses the whole surface of the spiral and a water flows back towards the heel of GHP where there is a phase change again.

Heat exchanger is a hollow cylinder and do not contain any obstacles except for spiral. The heat transfer medium vapor flows freely in whole cylinder, which will ensure vapors free movement and their effective condensation to walls of the spiral pipe. At the bottom of heat exchanger are welded four flanged adapters for another connection to evaporating section of the heat

^a Corresponding author: martin.vantuch@fstroj.uniza.sk

pipe, which consists of two parallel-connected U pipes from stainless steel.

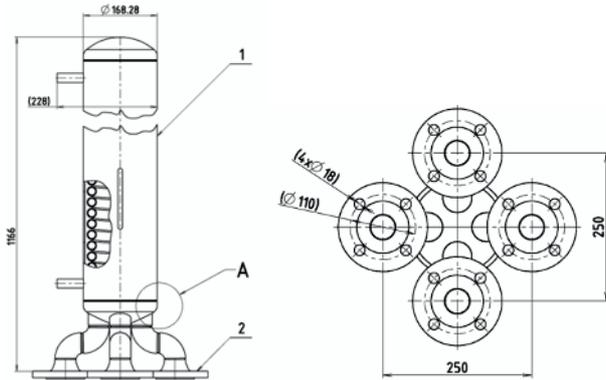


Figure 1. Flanged joints and detailed views of the heat exchanger

2.1 Heat transfer coefficient during condensation of NH_3

Were calculated heat transmission values of the heat exchanger model. As working substances of HP were selected ammonia (NH_3), and in forced circulation of the primary circuit was used mixture of 50% ethylene glycol (thermal G). In the calculation we used heat transmission properties of saturated vapour condensing ammonia. The input values for the calculation were chosen to the basis of substances and materials properties.

For Jacob's number must be observed $Ja \geq 10$:

$$Ja = \frac{l_{vA}}{c_p(t_v - t_s)} = 99.92 \geq 10. \quad (3)$$

From the calculation of Jacob's numbers have been validated criterion equations which can be used to calculate the Nusselt number

$$Nu = 0.729 (Ar Pr Ja)^{0.25}. \quad (4)$$

Heat transfer coefficient α ($\text{W m}^{-2} \text{K}^{-1}$) we obtain from equation

$$\alpha_A = \frac{Nu_A \lambda_A}{D} = \frac{476.01 \times 0.54}{0,0213} = 12067.85.$$

3 Experimental measurements on the geothermal borehole

The proposed facility is to simulate the transport of low-potential heat from rocks to heat pipe with working substance NH_3 during the phase change. The device consists of a cylinder filled with sand, bentonite and is inserted in the middle with a heat pipe.

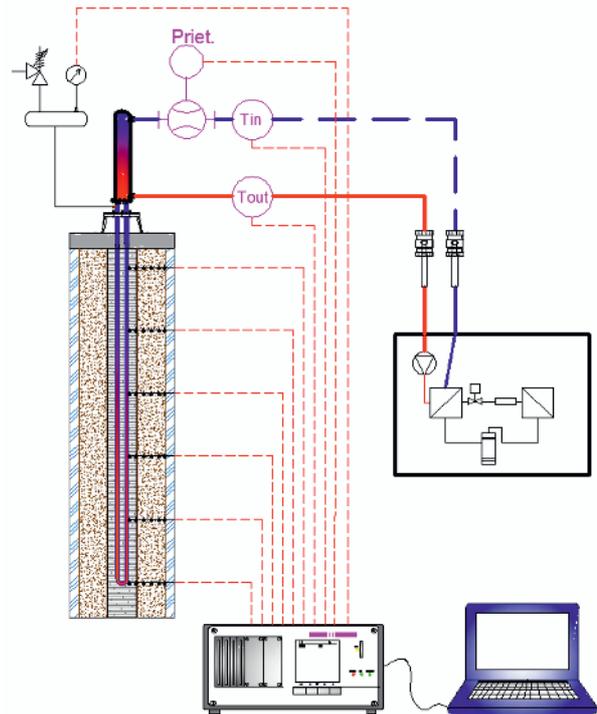


Figure 2. Measurements schematic system with NH_3 heat pipe

The proposed device is to simulate transport of low-potential heat from the ground to heat pipe with NH_3 working substance at the phase transition. The device consists of a cylinder filled with sand, bentonite that is placed in the middle of a heat pipe. On the surface of cylinder is wound DEVI heating cables in order to control heat flux into the soil (sand). At the top of condenser HP is equipped a heat exchanger, where there is heat transfer (cooling) in working condensation substance HP. To determine the thermal performance of heat pipe has been designed cooling circuit. Cooling circuit flows substance Thermal G (ethylene glycol 50/50). The inlet temperature to the heat exchanger we regulate with circulatory thermostat Julabo FL 2503. On the inlet and outlet pipe heat exchanger was fixed PT 100 thermocouples which panned circulating temperature (cooling) fluid.

Measurement were initiated at 12°C temperature of simulator ie. at a temperature of the ground at a depth of 150 meters in real borehole. Cylinder heating was set to persistent 60 W / m in depth, which represents the approximate performance of dry rocks. After starting the circulation thermostat, that reference temperature was set at -15°C we observed output heat performance obtained by the condensation part of the GHP and also were recorded of temperature fields in the simulation cylinder. Status simulator at the beginning of measurement:

- NH_3 working substance pressure in the heat pipe $p = 4,4 \text{ MPa}$,
- average temperature in the simulator $t = 12^\circ\text{C}$,
- total amount of NH_3 working substance in HP $m = (20\%) 5 \text{ kg}$,
- length of the heat pipe 5 m .



Figure 3. Experimental simulator boreholes with NH₃ heat pipe

During the measurement, the working substance was continuously cooled and temperature fell steadily. At the beginning it was clear that the substance on primary side is cooled very rapidly and abnormally heated with minimum temperature gradient.

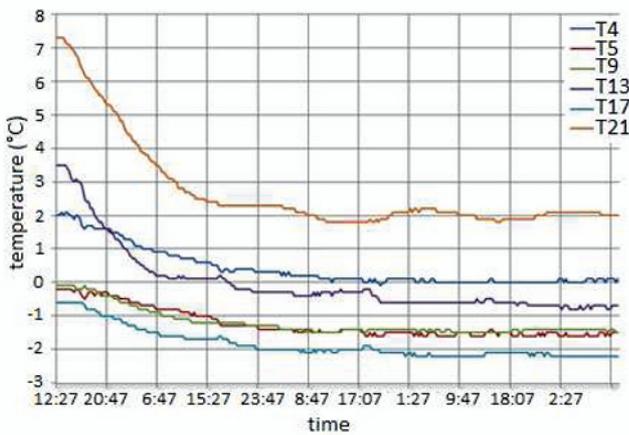


Figure 4. Temperature changes in the borehole during the measurement

During the measurement, the working substance was continuously cooled and temperature fell steadily. At the beginning it was clear that the substance on primary side is cooled very rapidly and abnormally heated with minimum temperature gradient. For 30 minutes decreased the temperature at -4 °C and end of the day reached a value up to -14 °C at the output of primary circuit of the cooling thermostat. During period of measurement, mass flow was maintained at a constant value of 10 kg / h. Temperatures in various depths showed none resp. minimal changes in the start of test (which evidence to suggest that there has been a boiling of liquid phase in HP). Later temperatures stabilized at constant values throughout the period of measurement. From the measurement was recognizably that the heat pipe did not

show required operation and heat transfer was minimal only. During the measurement was released a part of ammonia vapour volume in order to clear any unwanted gas mixture which could penetrate to GHP in filling process. This would prevent some NH₃ vapour flow to the condensation section and condensation had not carried. After ammonia deflation, the heat transfer was increased but the rise was only for a step and later it dropped again to a minimum value. The thermal performance has stabilized at a constant 25 W, representing only heat loss of heat exchanger.

These facts may result from several causes:

- Inadequate roughness resp. wettability of the surface - this is an important factor in heat transfer (α) of soil to the heat pipe. If the condensed working fluid does not leak evenly (distribution of NH₃ layer at a larger area with smaller thickness) it may cause decreased performance of the heat pipe during heat transportation. If leaking layer cover only a small percentage of surface pipe GHP, mentioned phenomenon may greatly affect obtained performance.

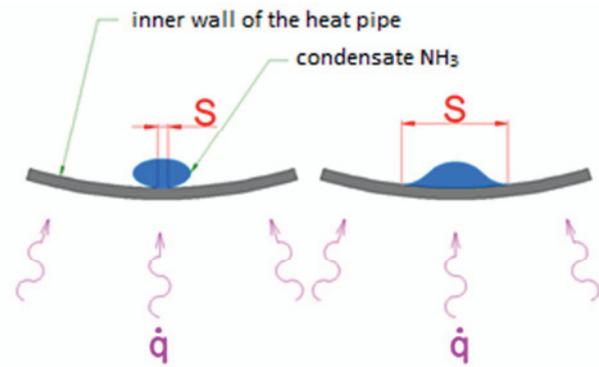


Figure 5. The wettability of heat pipe inner surface

The amount of shared heat expresses equation:

$$Q = \alpha (t_w - t_k) S, \quad (3)$$

where t_w is the temperature of the heat pie inner wall and t_k is the temperature change in the physical state of condensate corresponding value of pressure in HP.

- Because the condensing section of the heat exchanger was in form a hollow cylinder the junction of two U pipes, uneven working fluid dispensing for each U-GHP is one option that has occurred in our case. This may cause the condensate from the heat exchanger pour-down to lower part of the GHP has drained to only one U-tube. This may result degradation performance of GHP due to:
 - The loss function one of the U-tube (whereas the one of HP is empty, it will not run phase transformation).
 - Increases hydrostatic pressure due to amount of NH₃ liquid column of next full U-tube.

$$p = \rho_{NH_3} g h. \quad (4)$$

Said possibilities affecting the functionality of GHP were highly probable and caused the heat pipe cannot reached expected performances of the heat generated from the earth compared to forced circulation of the working substance.

In next step was designed heat exchanger with different condensation chamber. In this heat exchanger the condensation parts U pipes are hermetically sealed and separated without mutual common space. All of them has filler valve in which was brought same volume of NH₃ as in the previous case – 2 × 2.5 kg for each U tube. Condensation parts of U tubes are washed coolant supplied from the top of the cylinder to the bottom see figure 6. Condensation part of HP is held this time straight pipeline, so that not to break continuous stream of flowing condensate in the HP. This will ensure greater area of contact condensate with a wall of HP and thus intensified heat transfer.

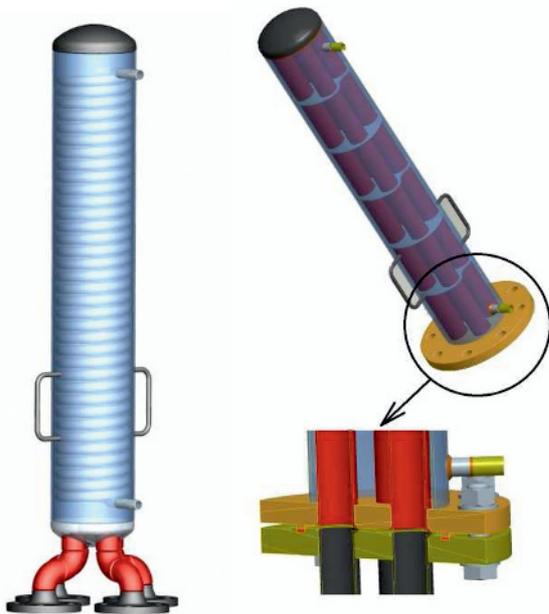


Figure 6. The original concept with curved pipes (left), the new concept of heat exchanger with straight pipes (right)

After starting cooling process occurred instantaneous phase transition in HP. There has been an immediate decrease in temperature in the vicinity of HP at all sections of the evaporation part. An immediate boiling of ammonia in HP in the cooling condensation part causes a pressure drop in HP. The decrease in temperature was recorded in other hours and following days. Temperature nonlinearly fell, mainly in the bottom of HP. In these parts, the drop in temperature was the most intense. Temperature at 4 meters fell to the lowest minimum during the measurement to -8.5 °C. After about 24 hours, the coolant temperature reached the reference value of -14 °C. From this moment, the cooling rate simulation well was dependent on the ability of heat transport by HP at a constant temperature of the coolant at the inlet to the heat exchanger. The graph shows that the heat transfer process from soil to HP did not take place in all sections as well. E.g. temperature *T4* showed a higher value compared to *T5* and *T9*, which caused hydrostatic

pressure column height levels of NH₃, which is in standby mode, reaching up to 0.6 meters. This has greatly influenced transmission properties in the lowest part of HP. Assume that the liquid surface contact with the inner wall was the largest at the bottom of HP, where was located the greater part of the liquid volume of ammonia.

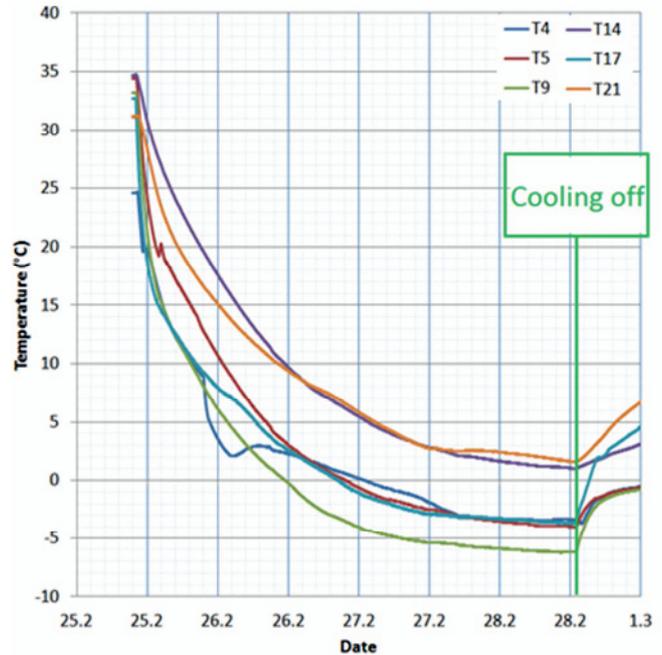


Figure 7. Temperature degrees in the borehole during the measurement

Condensate flowing down from the heat exchanger had a discontinuous flow toward the bottom due to viscosity and wettability. That the evaporation surface area of flow condensate with the inside wall of HP will be less, resulting in reduced heat transfer from the surrounding (soil) to HP.

Thermal power obtained on the condenser of HP in the heat exchanger at the beginning of the measurement outreach values through 1 kW because of the high particulate thermal yield of the borehole. During the measurement, the obtained capacity decreased to around 200 W. At this value, the power stabilized and the temperatures in different depths have not fallen more. We have reached a state of heat flows balance between supply and obtained heat from the borehole.

Before starting to measure, the absolute pressure in the pipe reached a value of 0.68 MPa. At this pressure, it is the boiling point of ammonia at 13 °C. At the time of start cooling, a pressure drop in the HP occurred, and during the 24 hours, it dropped to 3.1 MPa. In the next stage of measurement until its completion, it dropped by only two tenths to 3 MPa, due to achieving the reference value of cooling -14 °C and gradual cooling simulation of the cylinder (decrease in heat capacity of soil).

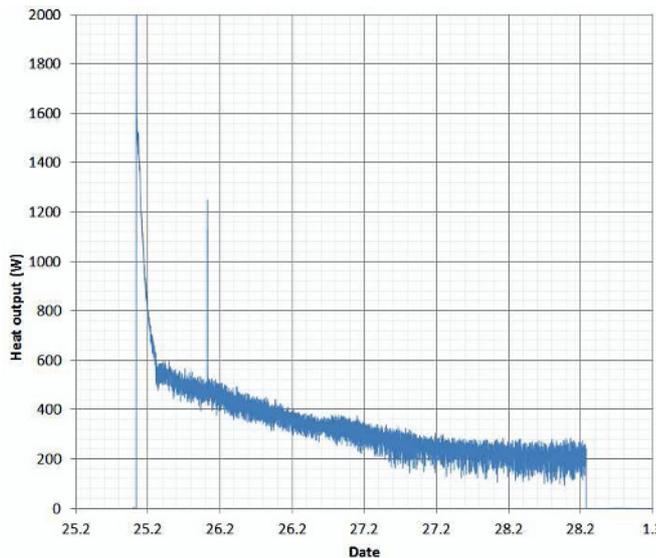


Figure 8. Proceedings of change thermal power obtained by HP

4 Conclusion

Measurement has shown that the design of HP and its condensation part has significant influence on performance of HP when transporting low-potential heat. Furthermore, has been shown that performance of HP affects wettability and shape of the inner wall surface which directly impact on condensate way flowing condensate. Is important that the contact surface of condensate with the inner surface of condensate HP was the greatest possible. Another important factor is to determine the correct amount of working fluid is in HP. The ideal amount is a continuous layer of condensate, which covers the entire surface of cylinder without residual amount of condensate in the bottom HP. If the volume of working fluid is excess, may get an increase in hydrostatic pressure at the bottom HP due to height of condensate column in HP. then there is no boil working fluid and heat transfer at the bottom of HP is nil.

4.1 Acknowledgement

Article was prepared under the Operational Program Research and Development ITMS – 26220220057 “device uses low-potential geothermal heat without forced circulation of heat carrier in a deep borehole”.

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