A Loop Thermosyphon for Asphalt Tank Heating

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Abstract

The objective of this work is to evaluate the existing steam coils of the asphalt tanks of the Brazilian Petroleum Company as a condenser of a loop-thermosyphon. The existing asphalt tank heating system uses steam generated in a 10 bar boiler placed far from the tanks. The basic idea is to replace this system by loop-thermosyphons using the existing coils. The evaporator would be placed nearby the tank and the heat source would be gas combustion. Then the fabrication of the new system would not involve modification of the coils inside the tanks, which means minimum construction time. The main advantage of using loop-thermosyphons for asphalt tank heating is the independency from the plant boiler. Also, the loop-thermosyphon condenser temperature could be controlled by burning gas according to the desired heating power. Therefore, it would not be restricted to the maximum boiler temperature of 180°C. A small scale model of the system was built and tested in the laboratory. The results show that the system works very well when the condenser heat transfer coefficient between the condenser and the heat sink is small. However, when the heat transfer coefficient is larger, the system does not start-up completely: a large portion of the condenser is not reached by the vapor. This phenomenon could be due to either the presence of non-condensable gases or a failure in the start-up associated to the vapor flow sonic limit.

Key Words: loop-thermosyphon, asphalt heating, industrial heaters.

1. INTRODUCTION

Asphalt is the more viscous and dense product obtained from petroleum refining. Its main application is road paving. Because of its high viscosity, asphalt storage is made inside heated tanks. The higher the temperature, the lower is the viscosity and the easier is to transport it from one tank to the other. Petrobras, the Brazilian Petroleum Company, employs several large asphalt storage tanks in their plants with a capacity of more than a thousand tons. In order to keep the asphalt at the required temperature level of 140°C, the tanks are equipped with steam coils placed at the bottom. Figure 1 shows a schematic of the steam coil lay out. The steam is provided from a 10 bar boiler that is responsible for all the steam used inside the Plant. However, as the maximum vapor temperature inside the coil is 180°C, the resulting asphalt temperature in most of the tanks is of 130°C. Therefore, in order to attend the 140°C demand, a new heating system is required.

This work presents the preliminary results of an experimental study on an engineering model of a loop-thermosyphon for application on heating of asphalt tanks. The objective of the thermosyphon is to replace the actual heating system. The condenser of the loop-thermosyphon will be the existing coil. The evaporator will be placed nearby the tank and will be heated by natural gas combustion.
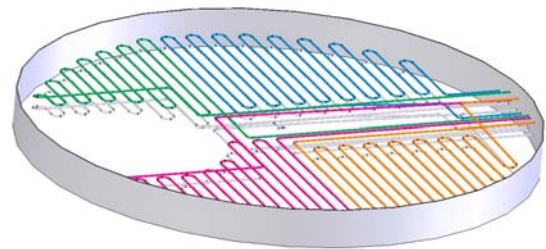

Figure 1. Asphalt tank steam coil lay out

The advantages of the loop-thermosyphon over the current equipment are basically two: independency from the plant boiler and more flexibility regarding to the temperature operation. As the actual system is limited to a 180°C coil temperature, as already mentioned, the asphalt does not reach the required temperature level of 140°C because of the heat losses. With the loop-thermosyphon, the temperature could be controlled by burning gas in the evaporator section according to power input needed. Also, the system

would be independent from the Plant central boiler and more compact, which means less heat losses to transport steam from the boiler to the tank.

Loop thermosyphon heat exchangers, also known as separated heat pipes, have been successfully applied in industrial waste heat recovery systems. Dube et al. [1] present a study on such a system. These researchers were particularly focused on the effects of non-condensable gases on the system performance. In a typical loop thermosyphon heat exchanger the geometry of the condenser differs significantly from the lay out sketched in Fig. 1. Figure 2 presents a schematic drawing of a typical loop thermosyphon heat exchanger. Both the evaporator and the condenser are geometrically very similar. They consist of two horizontal headers (upper and lower) connected by several vertical tubes in parallel. The vapor line coming from the evaporator is connected to the upper header. As vapor condenses, the liquid flows by gravity to the lower header, which is connected back to the evaporator.
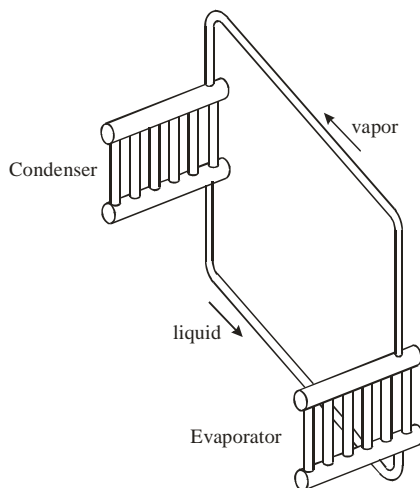

Figure 2. Loop thermosyphon heat exchanger

As for the system under investigation, the main concern is about the geometry of the condenser, which is the existing steam coil shown in Fig. 1. The active length of the condenser is mostly in the horizontal orientation, apart from the "U"-turns, which are slightly tilted towards the condensate flow direction. The main objective of this work is to verify if the steam coil of Fig. 1 can be used as the condenser of a loop thermosyphon. The main concerns are the start-up and the effect of non-condensable gases on the performance of the system.

2. EXPERIMENTAL STUDY

2.1. Experimental Set-up

An experimental set-up was built and tested in order to evaluate the applicability of the loop-thermosyphon conception for asphalt tank heating. As the actual system is too large to be tested in a laboratory, an engineering model was designed and built. Figure 3 presents the experimental set-up. The coil diameter was built in a 1:6 scale of the actual system, which means 6 mm i.d and 11 m long. As one can see, the geometry of the experimental setup condenser is not identical to the coil of Fig. 1. However, the number of U-turns in both cases is the same, and the total length of the condenser is 1:6 of the actual system. These differences between the model and the actual coil are not expected to cause inconveniences.

Regarding to the condenser heat sink, two types of tests were performed: air under natural convection and forced convection of a thermal fluid (ethylene-glycol). In the forced convection tests, the condenser is placed inside a tank filled with the thermal fluid. The temperature of the thermal fluid is strictly controlled thought a LAUDA® PR855 controlled temperature thermal bath.

The evaporator of the model is made of a SS 316 tube, of 100 mm i.d. by 400 mm long, where the longitudinal axis is in the horizontal position. The working fluid is distilled water and the filling ratio is 80% of the evaporator volume. The vapor line is connected to the top of the horizontal cylinder, while the liquid return line is connected to the side of the cylinder, below the liquid pool level. Heat is provided by eight 20 mm o.d. cartridge type electrical heaters immersed in the liquid pool.

The evaporator of the model does not correspond to the actual system. The evaporator of the actual system should be more like the evaporator of the loop heat pipe of Fig. 2, and the heat source should be the hot exhaustion gases from a gas burner. However, the study is focused on the condenser only, as already mentioned. Electrical heaters were employed as the heat source, since they provide a better heat power control. As the entire system is well insulated with glass wool, by measuring the electrical resistance and the current, the heat power input could be accurately measured.

The condenser can also be tilted with respect to the horizontal position. During the tests, the angle "θ" of the condenser measured from the horizontal orientation (see Fig. 3) assumed values of 0°, 7°, 13°, 45° and 90°.

The condenser was instrumented with seven K-type thermocouples distributed evenly over its length. The first thermocouple was placed 50 mm from the start of the condenser. The seventh thermocouple was placed 50 mm from the end of
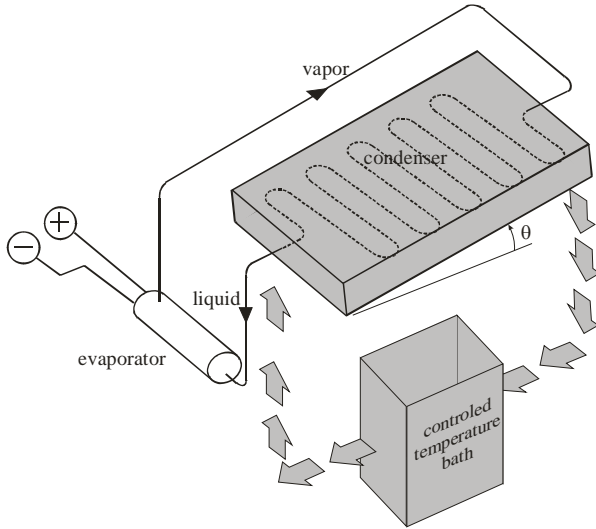
the condenser.

Figure 3. Experimental set-up

The vapor line was instrumented with two K-type thermocouples, one at the beginning and one at the end. The liquid return line was instrumented with eight K-type thermocouples evenly spread over its length.

The evaporator inside was instrumented with two K-type thermocouples: one immersed in the liquid pool and one immersed in the vapor space above the liquid pool.

The temperature, voltage and current were measured with a HP 3947-A® Data Acquisition System connected to a personal computed, which stored the data for further treatment.

2.2. Test Procedure

For each set of tests, i.e., air natural convection or thermal fluid forced convection, and for each of the condenser tilt angles, the heater was turned on, starting from the thermal equilibrium with the ambient. The power input was first set to 42 W. After the system reached steady state, the power input was increased by 6 steps up to 2000 W. In each power step, the system was left to reach steady state. In this condition, the thermal resistance R [°C/W] of the condenser is computed as:

$$R = \frac{T_e - \overline{T_c}}{\dot{Q}}, \tag{1}$$

where T_e [°C] is the temperature of the vapor above the evaporator liquid pool, $\overline{T_c}$ [°C] is the average of the seven thermocouples attached to the condenser and $\dot{Q}$ [W] is the heater power input.

2.3. Experimental Uncertainty

After calibration of the system, the uncertainty in temperature measurement is ±0.60°C in the range of temperatures of interest (ambient to 350°C). The uncertainty in voltage measurement is ±0.1%, and the uncertainty of the heater electrical resistance value was found to be ±4%.

Following the procedure described by Holman [2], the uncertainty in the thermal resistance measurement during the tests had a maximum value of ±8 %.

3. RESULTS AND DISCUSSION

Previous experience with conventional stainless steel-water thermosyphons showed that under certain circumstances the vapor does not reach the end of the condenser, especially for large values of the condenser external heat transfer coefficient (Refs. [3] to [5]). When the heat transfer coefficient is small, such as air in natural convection, the vapor generally gets very close to the condenser end. On the other hand, when the heat transfer coefficient is large, such as cooling by a liquid under forced convection, the vapor only wets a small part of the condenser. In this case, the temperature of the region of the condenser that is not reached by the vapor becomes equal to the heat sink temperature, i. e., there is no heat transfer between that portion of the condenser and the heat sink, which means that part of the condenser is inactive. This behavior can be attributed to non-condensable gases as described by Dube et al. [1], Peterson [6], Busse et al. [7], among others, or by the fact that the start up is not complete, as described by Mantelli et al. [3] and by Silverstein [8]. Non-condensable gases could be present because of either the hydrogen generated by chemical reaction between the water and the tube walls (stainless-steel) [1], [6], [7] and/or by undetected leakage of air into the system. The incomplete start-up of the system could happen when the vapor flow reaches the sonic limit [8].

With the knowledge of this behavior, as already mentioned, the condenser were cooled by both natural convection and forced convection. Based on the success of past laboratory experiments, the natural convection tests were performed first, in order to check the system proper operation, especially when the setup is subjected to the several different condenser tilt angle "θ" (see Fig. 3). However, air in natural convection is not a good representation of the heat transfer coefficients between the loop and the asphalt. For

the tube employed (8 mm o.d.) and for air under natural convection at an ambient temperature of 25°C, the global heat transfer coefficient between the thermosyphon working fluid and the heat sink (air) was estimated according to Incropera and De Witt [9] to be approximately 8 W/m²·°C. For the actual system of Fig. 1 with asphalt, the global heat transfer coefficient is approximately 80 W/m²·°C. On the other hand, the heat transfer coefficient for the forced convection test with ethylene-glycol, is approximately 170 W/m²·°C, which is a better representation of the actual data. Therefore, if the condenser worked correctly under the two extreme values of 8 and 170 W/m²·°C, it would also work for the asphalt.

3.1. Natural Convection Tests

During these tests, the tank of the condenser was open and the condenser was cooled by natural convection of air. Figure 4 presents the collected temperature data as a function of time for a typical test. Only the thermocouple placed inside the evaporator to measure the vapor temperature and the seven thermocouples spread on the condenser external surface are displayed here. As the system was turned on with a power input of 42 W, one can see that it did not start up since only the evaporator thermocouple was above the room temperature (approximately 25 °C). As the power input was increased to 307 W, the first five condenser thermocouples started to feel the presence of the vapor, one after the other. The readings of these five first thermocouples were practically identical and approximately 5 °C below the evaporator temperature. As the power input was increased to 871 W the vapor reached quickly the entire length of the condenser and all the seven condenser thermocouples read practically the same temperature.

Figure 5 presents the values of the thermal resistance of the condenser, defined according to Eq. (1), as a function of the heat power input, for all the condenser tilt angles tested during the natural convection tests. As one can see, the thermal resistance decreases as the power input increases. Also, the differences among the thermal resistance values for different tilt angles are large for small values of power input (less than 1700 W). In this case, the smaller values of thermal resistance were found for tilt angles of 7 to 13°. Interestingly, the maximum tilt angle, i. e. 90°, in general yielded larger thermal resistances. However, for power input of 1700 W and more, the effect of the tilt angle is negligible on the condenser thermal resistance. It also can be noted that a further increase in the power input would not lead to a significant decrease in the condenser thermal resistance.
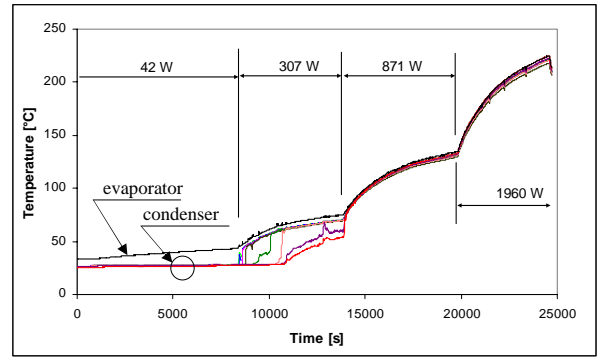

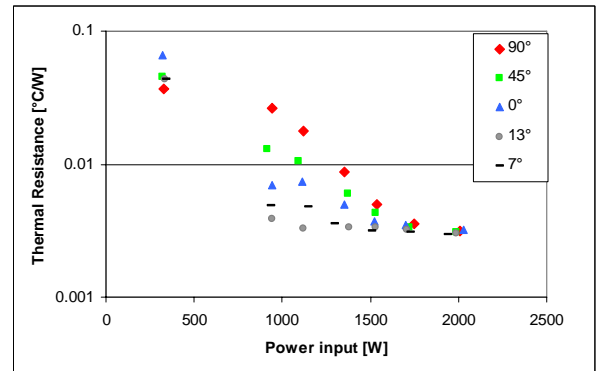Figure 4. Evaporator and condenser temperature readings as a function of time

Figure 5. Condenser thermal resistance as a function of the heat power input

3.2. Forced Convection Tests

In order to study the effect of the condenser external heat transfer coefficient on the system performance, the thermosyphon was tested under forced convection cooling, as already mentioned. Only tests for tilt angles of 0 and 7° were performed. The controlled thermal bath temperature was tested for 20, 80 and 120°C. In all cases, the vapor front never went beyond the second thermocouple of the condenser, which means at least 70% of the condenser length was inactive. Only a small portion of the condenser was enough for the vapor to condense. This observation agrees with previous works on conventional stainless steel-water thermosyphons [3] to [5].

According to references [1] and [6], conventional thermosyphons can experience similar behavior in the presence of non-condensable gases. According to Busse et al. [7], stainless steel-water thermosyphons generate hydrogen, which obstruct the condenser, leading to the appearance of a cold zone. Those authors studied several cleaning and fabrication procedures, including passivation by addition of oxygen in order to avoid the appearance of non-condensable gases. They concluded that none of these procedures were helpful in avoiding the presence of non-

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condensable gases. Dube et al. [1] also acknowledge the presence of non-condensable gases in stainless steel-water thermosyphons. As there is no way to avoid their presence, they studied experimentally the best position to place a gas reservoir to accommodate the non-condensable gases during operation of a loop-thermosyphon similar to the one shown in Fig. 2.

4. SUMMARY AND CONCLUSIONS

A coil type condenser geometry is analyzed in this work for application as a condenser of a loop thermosyphon for asphalt tank heating. This type of geometry differs significantly from the geometry generally employed for loop thermosyphon heat exchangers.

A small scale prototype of the condenser was built and tested in order to check the system operation conditions. Natural convection of air and forced convection of ethylene-glycol were used to remove heat from the condenser, in order to test a large range of possible heat transfer coefficients between the condenser and the heat sink.

The results showed that under a poor external heat transfer coefficient (natural convection of air), the vapor spreads through the entire condenser length. On the other hand, when the external heat transfer coefficient is larger (forced convection of ethylene-glycol), only a small portion of the condenser is active. According to the literature, this behavior could be either due to the presence of non-condensable gases or by a failure in the start-up of the system.

The presence of an inactive portion of the condenser is undesired for the application under study. It is necessary that the entire condenser length be wetted by the vapor, so the asphalt be uniformly heated inside the tank. This problem could be solved by installing a reservoir to trap the non-condensable gases, similarly to the work of Dube et al. [6], provided the source of the problem is indeed the presence of non-condensable gases. Further work is needed to investigate this phenomenon. A loop thermosyphon made of glass, similar to the one used in this study, is already being built to better understand this issue.

Could this problem of limited active condenser length be solved, the existing steam coils could be used as condensers of loop thermosyphons to heat the asphalt tanks. The optimum condenser tilt angle appears to be approximately 7°, according to experimental results, which is the tilt angle of the actual system. Therefore, the fabrication of the new system would not involve modification of the coils inside the tanks, which means minimum construction time. The main advantage of using loop-thermosyphons for asphalt tank heating is the independency from the plant boiler (10 bar). With the loop-thermosyphon, the temperature could be controlled by burning gas according to the desired heating power.

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