

An Experimental Study of a Carbon Dioxide-filled Thermosyphon for Acquisition of Low Temperature Waste Energy

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ABSTRACT: The aim of this work is to assess the possibility and usefulness of accessing low-temperature energy using a carbon dioxide-filled two-phase thermosyphon. Various parameters such as working fluid filling ratio, coolant mass flow rate, temperature difference between evaporator and condenser affect the heat transfer of thermosyphon system. In order to investigate the performance of thermosyphon system, we monitored the temperature distribution along the thermosyphon and measured input heat to evaporator section and output heat from condenser, as well. The results show that the heat transfer rate of thermosyphon increases with increasing mass flow rate and bath temperature. The results also show that the overall heat transfer coefficient increases with increasing coolant mass flow rate but decreases with increasing bath temperature. The reason of this decreasing is partly due to the enthalpy of vaporization of the carbon-dioxide decreases with increasing temperature.

Keywords: Carbon dioxide, Waste energy, Thermosyphon, Low Temperature

1. INTRODUCTION

The thermosyphon is a promising heat transfer device with very high thermal conductivity that employs the principle of evaporation and condensation of the working fluid, which is filled in a closed metal container. The thermosyphon has been proved to have comparatively high heat transfer capability with a very small temperature difference between the heat source and heat sink. Moreover, it is simple and cheap, therefore more attractive to the thermal industry [1]. Examples included gas-gas and liquid-gas heat exchangers; recovery low grade heat in chemical and electric power plants;

pollution and emission control in smoke stack industries; collection and utilization of solar energy; domestic heating and cooling; thawing of snow on major roads; thermal control of food storage units and commercial fisheries; food drying; and thermal treatment of exterior coating in auto industries [2].

The HFC refrigerants that were once expected to be acceptable permanent fluids are now on the list of regulated substances due to their impact on the climate change, and there is growing concern about their future uses. In this situation, it is hardly surprising that the industry is looking for completely different long-term solutions. Instead of

continuing the search for new chemicals, there is an increasing interest in technology based on ecologically safe 'natural' refrigerants, i.e. fluids like water, air, noble gases, hydrocarbons, ammonia, and carbon dioxide [3]. Carbon dioxide has received renewed interest in recent years as an important alternative refrigerant. High vapor density, a surface tension lower by one order of magnitude and a low vapor viscosity drastically influence the convective and nucleate boiling characteristics of CO₂. This results in heat transfer coefficients of CO₂ that are greater than those of conventional refrigerants by 2-3 times at the same saturation temperature while its two-phase pressure drops are considerably smaller [4]. High critical pressure (73.84bar) and rather low critical temperature (31.06°C) summarize its main disadvantage. The constraints arising from the very high operating pressure make it hard to design and to manufacture basic components, but the strong point of the carbon dioxide is related to its very good environmental properties; ozone-friendly and negligible global warming potential (GWP), even zero, if the fluid is drawn from waste of industrial processes [5].

A considerable experimental and theoretical works have been done on the application and design modification to improve thermosyphon performance. Faghir [6] has emphasized the both physical significance and mathematical manipulations as well as the technology and experimental issues related to heat pipe science. Recently, Ong and Haider-E-Alahi [7] investigated the performance of an R-134a-filled thermosyphon. They studied the effects of temperature difference between bath and condenser sections, filling ratio, and coolant mass flow rates on the performance of the thermosyphon. Noie [8] considered the effect of size parameters, which are aspect ratio and filling ratio, on the thermal performance of the thermosyphon for low

temperature. Payakaruk et al. [1] investigated correlation to predict heat transfer characteristics of an inclined closed two-phase thermosyphon at normal operation conditions. They used R22, R123, R134a, ethanol and water as working fluids and filling ratio was 50, 80 and 100%. Liu et al. [9] investigated the visualization of flow pattern in an ethanol-filled thermosyphon by electrical capacitance tomography. Jiao et al. [10] investigated the effect of filling ratio on a vertical thermosyphon using nitrogen as a working fluid. Rieberer [11] investigated ground-coupled heat-pumps based on the thermosyphon principle using carbon dioxide as a working fluid. The system is equipped with two 65m boreholes. Although numerous works have been carried out using various working fluids, there is little data on the thermal performance of a carbon dioxide-filled thermosyphon for acquisition of low-temperature waste energy (temperature range from 10 to 25°C).

Owing to various above reason, we investigated the performance of a two-phase carbon dioxide-filled thermosyphon experimentally for the acquisition of low-temperature waste energy. To simulate low-temperature waste energy condition, we made an experimental water bath system which could change temperature range from 10 to 25°C. The temperature distribution along the thermosyphon was monitored, and input heat to evaporator section and output heat from condenser were measured, as well. We suggested the effects of temperature difference between evaporator and condenser section, working fluid filling ratio, and coolant mass flow rates on the performance of the thermosyphon.

2. EXPERIMENTAL SET-UP

The thermosyphon consists of a 1000 mm-long cooper tube having inside diameter 10.9 mm and outside diameter

12.7 mm, as shown in Fig.1. The 400 mm-long condenser section consists of a water-cooled jacket surrounding the pipe. The water jacket is a 52.5 mm I/D acrylic tube with 6.35 mm O/D inlet and outlet connections. The tube was plugged at one end and provided with a screwed end cap sealed with Teflon tape. A compound pressure gauge was fitted via 6.3 mm O/D stainless cross piece to measure the working pressures in the pipe. A K-type thermocouple in a stainless steel sheath was fitted through the plug to measure the inlet and outlet temperature of the coolant water. The temperature distribution along the external surface of the thermosyphon in the evaporator and adiabatic sections were measured by K-type thermocouples mechanically attached to the surface of the pipe. Temperature distribution in the condenser section was measured using K-type thermocouples mounted on the external surface of the thermosyphon. Coolant water and bath temperatures were measured using K-type thermocouples. All the thermocouples were connected to the multipoint temperature monitor connected to a personal computer. The thermosyphon was immersed in a water bath which was provided with two electrical immersion heating and cooling elements of 0.25 kW each. Temperature difference between the top and bottom of the bath was less than 0.5°C. Coolant water was supplied via a constant-chemical pump. The coolant flow rate was measured using a water flow meter. The pipe was filled with CO₂. Referred to Rieberer's study [11], we selected five kinds of filling ratios (CO₂-filled volume to total volume), namely 0.0% (base case), 8.4%, 15.7%, 25.5%, and 33.8%, respectively. In each filling ratio, experiments were performed using five different mass flow rates (0.0067 kg/s, 0.01 kg/s, 0.0133 kg/s, 0.0167 kg/s, and 0.025 kg/s) and four different bath temperatures (10°C, 15°C, 20°C, and 25°C).

In each experiment, the thermosyphon was immersed in the water bath initially at ambient temperature. The bath temperature was then progressively changed in steps until steady-state was achieved at the temperature level desired. Steady-state was considered to be achieved when four sets of temperature readings taken at 1 second intervals showed less than 0.5°C variation. The procedure was then repeated for the next temperature setting.

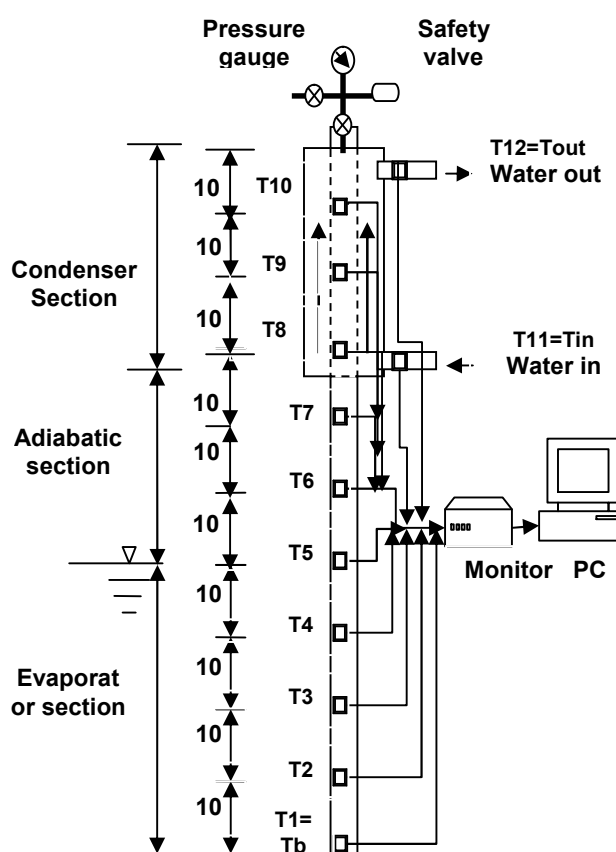


Figure 1: Schematic diagram of experimental rig. (length scale unit, cm)

In the absence of any heat loss from the pipe to the surroundings, the amount of heat absorbed at the evaporator section is equal to the amount of the heat rejected at the condenser. Thus, the performance of a thermosyphon operating between bath temperature (T_b) and condensing

temperature (T_c) may be expressed by the general heat transfer rate equation [7].

$$Q = U_e A_e (T_b - T_c) \quad (1)$$

Where Q is heat transfer rate (W), U_e is Overall heat transfer rate ($W/m^2/^\circ C$) and A_e is cross section area of thermosyphon (m^2). The condenser temperature (T_c) is approximated by the arithmetic mean of inlet (T_i) and outlet (T_o) coolant water temperatures, viz.

$$T_c = (T_i + T_o) / 2 \quad (2)$$

The heat transmitted from the heat pipe is equal to the heat rejected to the coolant water.

$$Q = \dot{m} c_p (T_o - T_i) \quad (3)$$

Where $\dot{m}$ is coolant water mass flow rate (kg/s), c_p is specific heat of water ($J/kg/^\circ C$).

3. RESULTS AND DISCUSSION

Fig. 2 shows a typical temperature distribution along the length of the thermosyphon at the filling ratio of 15.7% and coolant mass flow rate of 0.0167 kg/s for various bath temperatures (from 10°C to 25°C). As the bath temperature increased, there was a slight fall in the temperature towards the adiabatic section. This indicated a two-dimensional heat flow along the pipe wall towards the condenser section. In the condenser section, the pipe wall temperature increased towards the end where the coolant outlet pipe was located. This increase was attributed to the coolant water temperature which increased as it flowed towards the outlet. These results were similar to those from Ong and Haider-E-Alahi's study [7] for the R-134a-filled thermosyphon.

To compare CO₂-filled case with base case (vacuum = -85 kPa, filling ratio = 0.0%), temperature distributions along the length of the thermosyphon for four different filling ratios at coolant mass flow rate of 0.025 kg/s are shown in Fig. 3. Four different CO₂-filled thermosyphons showed apparently different temperature distribution compared with the base case. However, four different thermosyphons had very similar temperature distributions along the length of the thermosyphons.

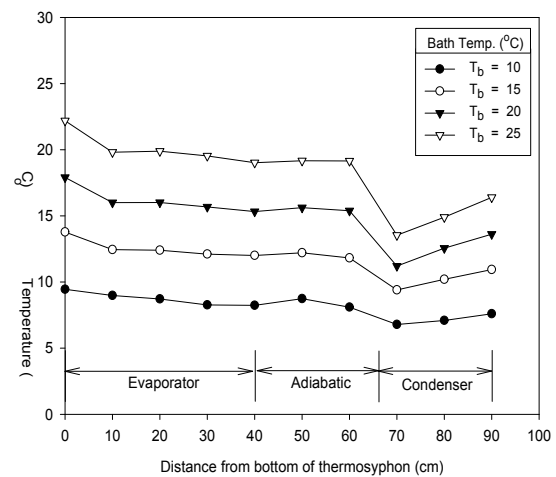


Figure 2: Temperature distribution along the length of the thermosyphon for various bath temperatures (Filling ratio = 15.7%, Mass flow rate = 0.0167 kg/s).

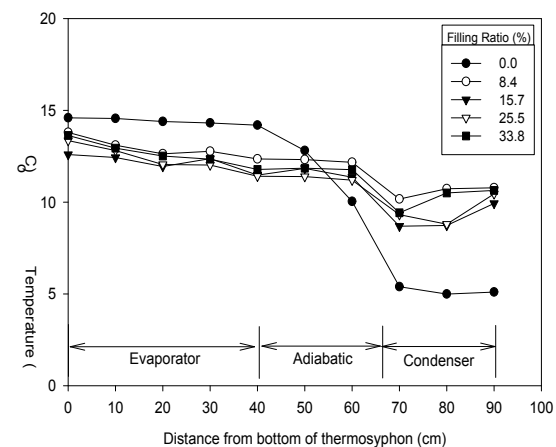


Figure 3: Temperature distribution along the length of the thermosyphon for a low

bath temperature (Mass flow rate = 0.025 kg/s, Bath temperature = 15°C).

Fig. 4 shows the difference between bath and evaporator temperatures plotted against bath temperature flow at a filling ratio of 15.7% for five coolant mass flow rates. Temperature differences from 1°C to 2.5°C were experienced. In the absence of any convective film between the water in the bath and heat pipe outer wall, there would not be any difference. As shown in the study of Ong and Haider-E-Alahi [7], the difference arises from the presence of the natural convective film heat transfer. The difference in temperatures increased with increasing bath temperatures.

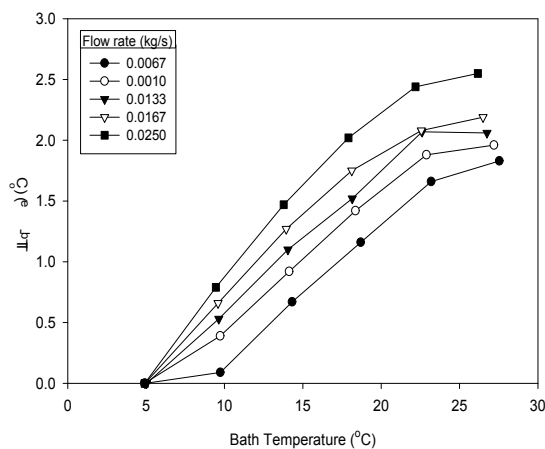


Figure 4: Difference between bath and evaporator temperature vs. bath temperature (Filling ratio = 15.7%).

As shown in Fig. 5, above filling ratio of 15.7%, the temperature between bath and evaporator section was stably increased with increasing bath temperature. In addition, temperature difference was shown to be the maximum at the filling ratio of 15.7%.

Fig. 6 shows the transient plots of calculated heat transfer rates from the hot water in the water bath to the condenser water flow for a fill ratio of 15.7% and a coolant mass flow rate of 0.025 kg/s under bath temperature from 10 to 25°C. The

results showed that heat transfer rates increased with increasing bath temperatures. The results obtained at 1 min intervals for the various bath temperatures from 10 to 25°C showed that steady-state was achieved almost instantaneously, within 1/2 min.

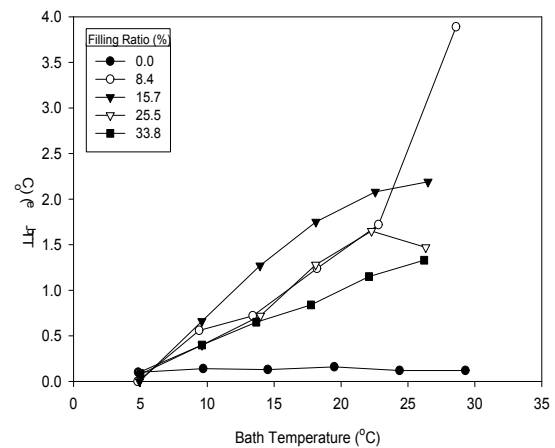


Figure 5: Bath temperature vs. temperature difference between bath and evaporation section (Mass flow rate = 0.0167 kg/s).

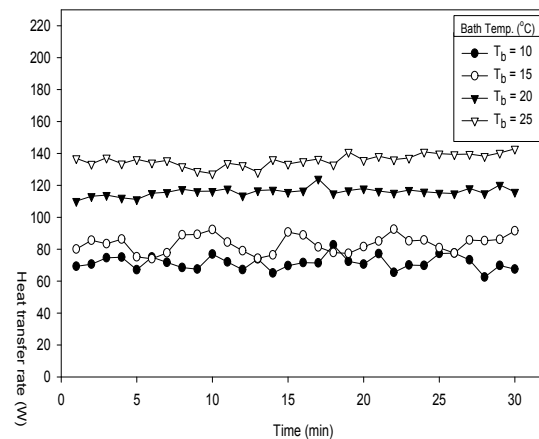


Figure 6: Heat transfer rate vs. time (Filling ratio = 15.7%, Mass flow rate = 0.025 kg/s).

Heat transfer rate plotted against temperature difference between bath and condenser regions at a fixed mass flow rate was shown in Fig.7. The results showed that heat transfer rate increased with increasing temperature difference

between bath and condenser regions. However, it seemed that heat transfer rate was independent of filling ratio.

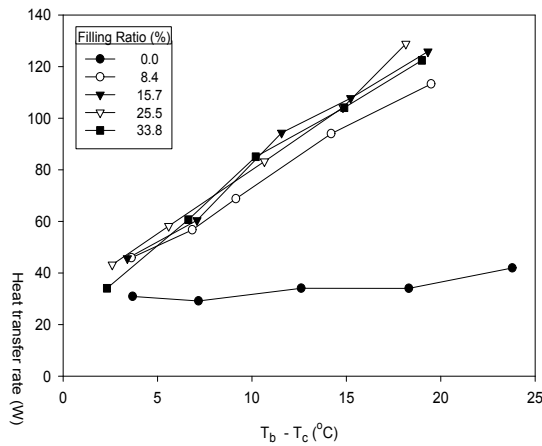


Figure 7: Heat transfer rate plotted against temperature difference between condenser region (Mass flow rate = 0.0167 kg/s).

Fig. 8 shows heat transfer rate plotted against temperature difference between bath and condenser regions at a fixed filling ratio. The result showed the heat transfer rate increased with temperature difference between bath and condenser regions. The result also showed the heat transfer rate increased with increased mass flow rate.

In Fig. 9, the overall heat transfer coefficient (U_e) is plotted against the difference between bath and mean condenser water temperatures for the five filling ratios used from 0.0% to 33.8% under coolant mass flow rate 0.025 kg/s.

Overall heat transfer coefficient was found to decrease with increased operating temperature difference. These results were greatly different from those of water- or R134a-filled thermosyphon, which showed heat transfer coefficient increased with increasing operating temperature difference [7, 8]. These decreasing overall heat transfer coefficient tendency is partly due to the enthalpy of vaporization (latent heat) of the carbon-

dioxide decreases with increasing temperature.

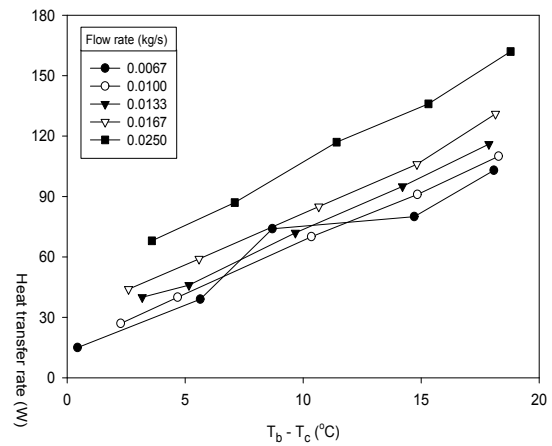


Figure 8: Temperature difference between bath and condenser region vs. heat transfer rate (Filling ratio = 25.5%).

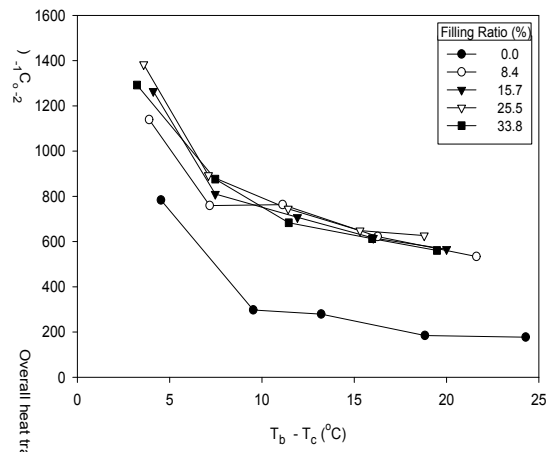


Figure 9: A Filling ratio vs. overall heat transfer coefficient at a fixed mass flow rate (Mass flow rate = 0.025 kg/s).

Fig.10 shows the variation in overall heat transfer coefficient plotted against filling ratio. Overall heat transfer coefficient increase with mass flow rate, but there is no significant increase with increased filling ratio. At the coolant mass flow rate of 0.025 kg/s, the highest value of overall heat transfer coefficient was about 400 Wm^2C^{-1} for a filling ratio of

8.8% and about $900 \text{ Wm}^{-2}\text{C}^{-1}$ for a filling ratio 33.8%.

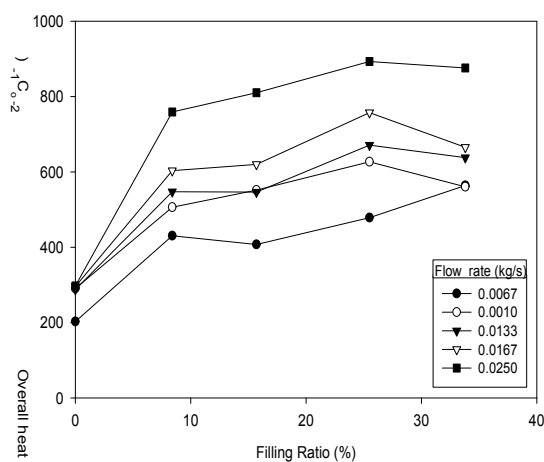


Figure 10: Overall heat transfer coefficient vs. filling ratio for various mass flow rates (Bath temperature = 15°C).

5. CONCLUSION

To investigate for the acquisition of low-temperature waste energy using CO_2 -filled thermosyphon, we investigated performance of a two-phase CO_2 -filled thermosyphon experimentally. The main results of this study were summarized as follows.

- (1) Compared with the case of zero filling ratio (vacuum = -85kPa), CO_2 -filled thermosyphon had apparent different temperature distribution along the length of the thermosyphon.
- (2) Heat transfer rates of CO_2 -filled thermosyphon increased with increasing bath temperatures and coolant mass flow rate.
- (3) Overall Heat transfer coefficient was found to decrease with increased operating temperature difference. This decreasing is partly due to the enthalpy of vaporization of the carbon-dioxide decreases with increasing temperature.
- (4) Overall heat transfer coefficient increase with mass flow rate, but there is

no significant increase with increased filling ratio.

(5) We can conclude that an environmentally friendly CO_2 -filled thermosyphon system can be used for the acquisition of low-temperature waste energy (temperature range from 10 to 25°C) under appropriate filling ratio and operating condition.

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