Condensation in an annulus with spiralled wires

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There are different types of heat transfer enhancement techniques that can be used in air-conditioning and refrigeration systems to improve heat transfer, reduce heat transfer area or weight and to decrease the logarithmic mean temperature difference. Some of the heat transfer techniques are expensive and cannot be afforded by small manufacturing firms. An easy and affordable technique considered in this paper has a wire spiralled in the annulus of a water-cooled, tube-in-tube condenser, where R-22 is condensing in the annulus while the water flows through the inner tube. The purpose of this paper was to determine the heat transfer and pressure drop characteristics of the refrigerant R-22 in the annulus. Measurements were taken on four different heat exchangers, three with wires spiralled in the annulus at different angles and one without any wires used as a reference. It could be concluded that the use of the wires improved the overall heat transfer coefficient by up to 350% (45° spiral). However, as expected the pressure drop also increased, by 225%.

Nomenclature

A	heat transfer area, m^2
C_p	specific heat, J/kg K
D	diameter, m
f_p	pressure drop coefficient, dimensionless
i	enthalpy, J/kg
l	length of heat exchanger, m
m	mass flow rate, kg/s
p	pressure, Pa
ΔT_{lm}	logarithmic mean temperature difference, K
T	temperature, K
Q	heat transfer rate, W
U	overall heat transfer coefficient, W/m^{2}K
v	specific volume, m^3/kg
θ	wire angle (Figure 1), degrees

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Subscripts

a	annulus cross-section
c	condensing
f	fluid
g	vapour
h	hydraulic
in	inlet of condenser
o	outside
out	outlet of condenser
r	refrigerant
w	water

Introduction

Energy is currently being used in the United States at a rate roughly equivalent to 6 300 000 m^3 of oil per day.[6] More than a third of this energy is used for residential and commercial space heating and air-conditioning, residential and commercial water heating, and industrial petrochemical processing.[9] An efficiency improvement of 10% in the cited applications by using heat transfer augmentation would save about 238 000 m^3 of oil per day, and reduce atmospheric CO_2 emissions by about 400 million metric tons per year.[8] Therefore, heat transfer augmentation of evaporation and condensation for refrigeration, air-conditioning and heat pump applications has received increasing attention in recent years from an energy efficiency point of view.[14,11] This is due to not only an emphasis on energy efficiency, but also a need for more compact and lighter heat exchangers, as well as possible reductions in the charges of refrigerants, and the phasing out of chlorofluorocarbon (CFC) refrigerants programme since the end of 1995 as stipulated by the Montreal Protocol , the Copenhagen and Vienna amendments[2] and the recent Kyoto agreement.

Tube-in-tube heat exchangers are inexpensive and easy to manufacture but not very effective. When R-22 hot-water heat pumps are operating near their maximum safe condensing temperature of 55° C, it is difficult to reach hot-water outlet temperatures of 60° C and higher for tube-in-tube heat exchangers. A temperature higher than the condensing temperature is possible by absorbing heat from the refrigerant superheat before condensation starts.

On the other end of the scale of heat exchangers are fluted tubes, which are efficient, but expensive, and not easy to manufacture. Hot-water temperatures of 60° C and higher can be reached at condensing temperatures of 55° C, the reason being that the heat transfer coefficient in the annulus of fluted tubes is higher than that of plain tube-in-tube heat exchangers. A need therefore exists to investigate easy-to-manufacture and cheap alternatives for heat transfer augmentation of tube-in-tube heat exchangers. R-22 was used in this study as complete cessation of the production ofHCFCs is only called for by January 1, 2030.[1]

Wires spiralled in the annulus (on the inside tube) will establish swirl flow in the annulus and it was observed from literature that devices that establish swirl in the flowing fluid are particularly attractive augmentative schemes for forced convective systems.[7,10,5,13] The intensity of the swirl in the annulus will be altered with the number, thickness, and angle of the wires. Figure 1 shows a schematic representation of this type of tube-in-tube heat exchanger. The purpose of this paper was to determine the heat transfer and pressure drop characteristics of the refrigerant R-22 (Monochlorodifluoromethane, $CHClF_2$) in the annulus of the proposed method (Figure 1) of heat transfer improvement.

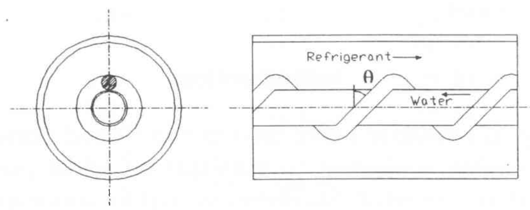

Figure 1 Schematic representation of the spiralled heat exchanger (not to scale)

Experimental set-up

The tube-in-tube heat exchanger consisted of two concentric, soft-drawn, copper refrigeration tubes. The inner tube had an outer diameter of 9.53 mm and an inner diameter of 8.11 mm. The outer tube had an outer diameter of 15.9 mm and an inner diameter of 14.3 mm. The total length of the heat exchange areas was 12.4 m.

The heat exchangers considered in the analysis had copper wires of 1.3 mm diameter spiralled in the annulus. The first heat exchanger had no spiralled wires, while the rest of the heat exchangers had the wires spiralled at 30°, 45°, and 60°, respectively. The heat exchanger was co-axially coiled to a diameter of 400 mm and operated in a counter-flow arrangement. It was insulated with 50 mm of glass wool to minimise losses to the atmosphere. The water flowed in the inner tube and the condensing refrigerant in the annulus.

The experimental set-up (Figure 2) was a closed system, using R-22 in a vapour compression cycle together with a hot-water cycle and a cold-water cycle. The vapour compression system comprised a compressor, followed by a water-cooled condenser, filter, sight-glass, expansion valve, and a water-heated evaporator. Pressure dial gauges were placed before and after the condenser, and before and after the evaporator. The accuracy of the pressure gauges was ±3%. The compressor-input power was measured with a wattmeter with an accuracy of ±3%.

Pt 100s were placed to measure the inlet and outlet temperatures of the water and the R-22 in the condenser, respectively. The accuracy of the Pt 100s for measuring temperature differences after calibration was ±0.1°C with a 99% confidence level.[3] All the Pt 100s were installed tightly onto the tube surfaces with aluminium tape as well as insulation material on top of the tape to minimise heat transfer to the atmosphere. A high thermal conductivity paste was used to increase the heat transfer between the tube and the Pt 100s. The smallest temperature difference measured between inlet and outlet was 10°C and this was large enough to employ only surface-mounted temperature detectors. The cold-water tank (1000 ℓ) was connected to a chiller to cool the water if necessary.

Hot water from an on-site, hot-water storage reservoir (1000 ℓ), fitted with an electrical resistance water heater, was pumped through the condenser inner tube. By using a resistance heater inside the reservoir, the water was heated

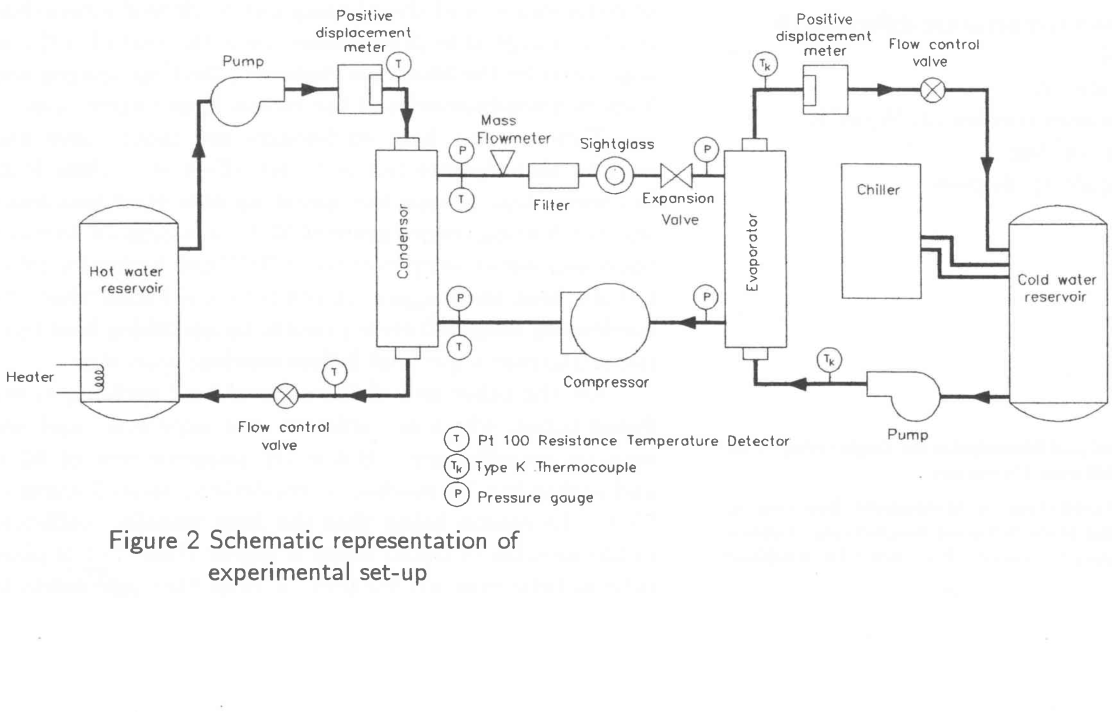

Figure 2 Schematic representation of experimental set-up

to provide an increase in the hot-water temperature, so that readings could be taken over a range of hot-water temperatures. Two positive displacement watermeters calibrated to a ±1% accuracy were installed on the hot- and cold-water sides.

During experiments, the condensation temperature was maintained at a value of 60°C and the evaporation temperature at −0.7°C. The water inlet temperatures changed as a result of the heater in the hot-water reservoir and the chiller connected to the cold-water reservoir. As the temperatures changed, the flow rates had to be changed accordingly, to maintain the desired pressure (and thus temperature) in the condenser and evaporator. Measurements were only taken once all the readings had stabilised, which represented steady state conditions. Energy balances between the water and refrigeration sides of the condenser were within ±5%, where the average value was taken as the correct value. The tube-side Reynolds number of the water was kept constant at approximately 6 300.

Processing of results

To determine the heat transfer in the condenser, the total heat transfer in the condenser was calculated as the average between the refrigerant heat loss $(m_r(i_{in} - i_{out}))$ and the water heat gain $(m_w C_{p,w}(T_{w,out} - T_{w,in})$:

$$Q = 0.5\left\{m_w C_{p,w}\left(T_{w,out} - T_{w,in}\right) + m_r\left(i_{in} - i_{out}\right)\right\} \quad (1)$$

The specific heating capacity, $C_{p,w}$ of the water was calculated at the mean water temperature. The enthalpy values were obtained from property tables of R-22[12] by using the measured temperature and pressure values. According to the compressor curves the refrigerant mass flow is 0.0156 kg/s at a condensing temperature of 60°C and an evaporation temperature of −0.7°C. This estimation was used originally, as a refrigerant mass flowmeter was not available for this study. After the experiments were completed, the refrigerant mass flow was measured at an evaporating temperature of −0.7°C and a condensing temperature of 60°C by using a coriolis mass flowmeter with an error of ±0.1%. The mass flowmeter was installed between the condenser and filter. The refrigerant mass flow measured was 0.0156 kg/s as stated in the manufacturer's compressor curve. This could be expected as the difference in heat transfer on the waterside and refrigerant side of the condenser was always less than ±5%.

With the average heat transfer known from eq. (1), the logarithmic mean temperature difference over the condenser heat exchanger was determined, whereafter the overall heat transfer coefficient could be determined for each geometry with eq. (2).

$$Q = U_o A_o \Delta T_{lm} \quad (2)$$

The overall heat transfer coefficient was based on the outer area of the inner tube and the area of the wires was not taken into consideration.

As the density of the R-22 changes during condensation, the pressure drop coefficient[3] was determined from:

$$f_p = \frac{2\Delta p D_h A_a^2}{m_r^2 l_c\left(v_f + 0.5 v_{fg}\right)} \quad (3)$$

The hydraulic diameter was taken as the difference between the inside diameter of the outer tube and the outside diameter of the inner tube. As the wire diameter was so small, its influence was not taken into account. The detail on estimating the length of condensation is given by Coetzee *et al.*[4]

Results

All results are given at a refrigerant Reynolds number (based on the hydraulic diameter) of approximately 14 000. The overall heat transfer coefficients during condensation in the annulus for the different spiralled angles, compared to the case with no wires, are shown in Figure 3. The heat exchanger with wires spiralled, at an angle of 45° in the annulus, had the highest overall heat transfer characteristics. All the spiralled heat exchangers showed a large improvement in heat transfer when compared to the normal tube-in-tube heat exchanger with no wires spiralled in the annulus. The heat transfer increased by 350% from a tube-in- tube heat exchanger with no wires to a heat exchanger with 45° spiralled wires.

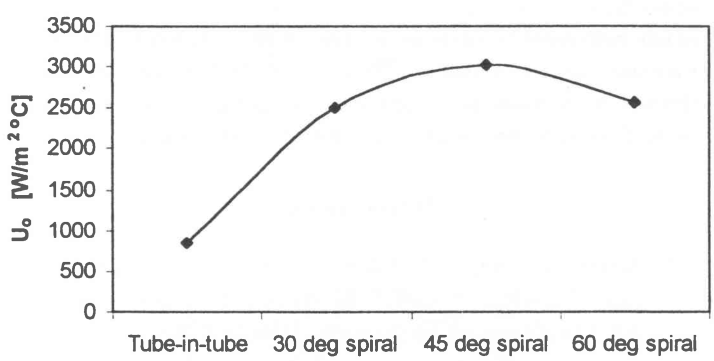

Figure 3 Overall heat transfer coefficients for the different geometries

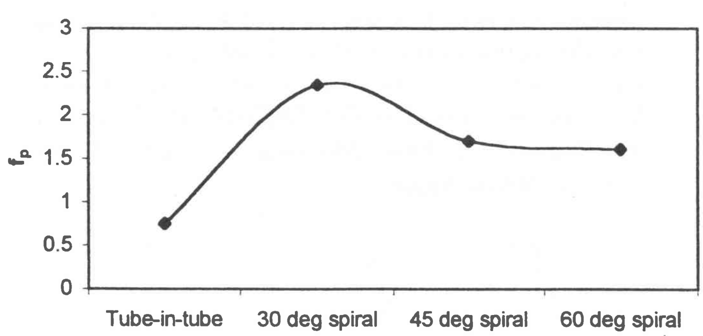

Figure 4 Pressure drop coefficients for the different geometries

A comparison between the pressure drop coefficients of each geometry considered is shown in Figure 4. The spiralled wires in the annulus increased the resistance to flow, as was expected. The heat exchanger with the densest spiralled wires ($\theta = 30°$) in the annulus had the highest value of pressure drop coefficient. The increase was 310% when compared with the tube-in-tube heat exchanger with no wires. However, the pressure drop coefficients are substantially lower at 45° and 60° (225% and 210% if compared to tube-in-tube). Taking into account the high overall heat transfer coefficients it appeared that if wires are used at angles of 45°, good performance could be expected from the heat exchanger / condenser.

Conclusions

The use of the spiralled wires resulted in a substantial improvement in heat transfer, with a larger pressure drop as penalty. The heat exchanger with wires spiralled at 45° showed the highest overall heat transfer and a relatively low pressure drop coefficient. This method of heat transfer augmentation is very attractive as an inexpensive alternative. Further tests can be conducted on other geometries to investigate the existence of an optimum wire angle (θ). With the present data, it could be concluded that a wire angle of 45° would give the best performance for air-source, hot-water heat pumps. It is also recommended that the measurements be repeated for other evaporating and condensing temperatures and therefore for other refrigerant mass flows. The measurements should also be repeated for other refrigerants such as R134a, R407c, R404A, R410A, propane, and isobutane. Measurements can also be conducted with more than one wire spiralled in the annulus, as well as spiralled wires with different diameters.

References

1. ASHRAE 1997. *ASHRAE Handbook – Fundamentals.* American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta GA.

2. Cavallani A 1996. Working fluids for mechanical refrigeration. *International Journal of Refrigeration*, **19**, pp.485–496.

3. Coetzee S 1998. Condensation of R-22 during heat transfer augmentation with spiralled wires in the annulus of a tube-in-tube heat exchanger for hot-water heat pumps. Project report, Department of Mechanical Engineering, Rand Afrikaans University, Johannesburg, South Africa.

4. Coetzee S, Da Veiga WR & Meyer JP 1999. Condensation of R-22 during heat transfer augmentation with spiralled wires in the annulus of a tube-in-tube heat exchanger for hot-water heat pumps. *Proceedings of the International Conference on Compact Heat Exchangers and Enhancement Technology for the Process Industries*, Banff, Canada, pp.377–384.

5. Gupta RK & Raja Rao M 1974. Heat transfer and friction characteristics of Newtonian and power law fluids in smooth and spirally corrugated tubes. *Advances in enhanced heat transfer*, **2**, pp.239–243.

6. Hedley D 1986. *World Energy: The facts and the future*, 2nd edn. Facts on File Publications, New York.

7. Hong SW & Bergles AE 1978. Augmentation of laminar flow heat transfer by means of twisted tape inserts. *Journal of Heat Transfer*, **98**, pp.251–256.

8. Hu X & Jacobi AM 1993. Local heat transfer behaviour and its impact on a single-row, annularly finned tube heat exchanger. *Journal of Heat Transfer*, **115**, pp.66–74.

9. Krenz JH 1984. *Energy Conversion and Utilisation*, 2nd edn. Allyn and Bacon Inc., Boston.

10. Marner WJ & Bergles AE 1978. Augmentation of tube side laminar flow heat transfer by means of twisted tape inserts, static mixer inserts and internally finned tubes. *Proceedings of the 6th Int. Heat Transfer Conference*, **2**, pp.578–581.

11. Meyer JP 1998. Evaluation of energy efficient and environmental acceptable pure and zeotropic refrigerants in air-conditioning and refrigeration. *Proceedings of the USA-RSA Bi-national Energy and Environment Workshop*, University of Durban-Westville, South Africa, pp.133–144.

12. REFPROP 1993. *Thermodynamic Properties ofRefrigerants and Refrigerant Mixtures Database*, Ver. 4, US Department of Commerce.

13. Sethumadhaven R 1982. Studies in high performance heat exchange surfaces. PhD thesis. University of Bombay, India.

14. Van der Vyver S & Meyer JP 1997. Heat transfer augmentation in the annulus of a heat exchanger consisting of a round tube inside a twisted square tube. *R & D Journal*, **13**, pp.72–82.