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Performance Comparison of Single-Stage and Cascade Refrigeration Systems Using R134a as the Working Fluid

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Abstract

This study presents an experimental comparison of single-stage and cascade vapour-compression refrigeration systems using R134a as the refrigerant. The experimental plants employ a vapour-compression refrigeration cycle serving as a base unit, a cooling tower and another vapour-compression refrigeration cycle serving as a higher-temperature unit in the cascade operation. In the single-stage operation the condenser of the base unit was connected to the cooling tower, whereas in the cascade operation it was thermally coupled to the evaporator of the higher-temperature unit via a water stream. Using data obtained from steady-state test runs, the performance characteristics of both systems, namely evaporating and condensing temperatures, refrigerant mass flow rate, compressor power, coefficient of performance (COP), compressor discharge temperature, compressor volumetric efficiency and the ratio of compressor discharge to suction pressures, were evaluated. The results show that, for a given refrigeration capacity, the cascade system provides a lower evaporating temperature, lower compressor discharge temperature, lower ratio of discharge to suction pressures and higher compressor volumetric efficiency at the expense of a lower COP.

Key words: Refrigeration, Cascade system, Single-stage system, Coefficient of performance, R134a.

Introduction

A vapour-compression refrigeration cycle with only one stage of compression is called a single-stage refrigeration system. A cascade refrigeration system, on the other hand, employs 2 or more individual refrigeration cycles operating at different pressure and temperature levels. The duty of the lower-temperature cycle is to provide the desired refrigeration effect at a relatively low temperature. The condenser in the lower-temperature cycle is thermally coupled to the evaporator in the higher-temperature cycle. Thus, the evaporator in the higher cycle only serves to extract the heat released by the condenser in the lower cycle. Then this heat is rejected into the ambient air or a water stream in the condenser of the higher cycle.

Since high ratios of pressure across the compressor cause undesirably high discharge temperatures, low volumetric efficiencies and excessive stresses on compressor parts, the maximum allowable pressure ratio for reciprocating compressors is limited to about 9 (ASHRAE, 1990). If the pressure ratio exceeds this limit for a specific application requiring a relatively low refrigeration temperature, it can be decreased using a cascade refrigeration system instead of a single-stage one.

A cascade refrigeration system operates with a lower evaporating temperature, smaller compression ratio and higher compressor volumetric efficiency when compared with a single-stage one. In a cascade system, the lower cycle may use a high-pressure refrigerant providing refrigeration at a low temperature with moderate evaporating pressures, while the higher cycle may use a low-pressure refrigerant rejecting heat at a high condensing temperature with-out extremely high condensing pressures. However, due to the fact that a cascade system requires at least 2 refrigeration cycles, it is more expensive to build and more complicated than a single-stage system. Moreover, the overlap of the condensing temperature of the lower cycle and the evaporating temperature of the higher cycle, which is caused by the heat transfer between the 2 cycles, reduces the efficiency of a cascade system.

An alternative to a cascade refrigeration system is a compound refrigeration system, which uses 2 or more compressors connected in series in the same refrigeration cycle. A variation of compound refrigeration systems, named as 2-stage refrigeration system with intercooler, utilises intercooling between the compression stages in order to reduce the compression power. However, unlike cascade systems, all compound systems suffer from the problem of oil return to the compressors. In order to assure equal oil return to each compressor connected in series, some extra equipment such as oil separators and float valves are utilised. Cascade refrigeration systems can be considered when an evaporating temperature below –18 °C is required (Dossat, 1991). These systems can provide temperatures down to –100 °C, which are generally required for some industrial processes involving the liquefaction of gases.

The literature on low-temperature vapour-compression refrigeration is usually focussed on compound refrigeration systems. A study on the thermodynamic analysis of a 2-stage R134a refrigeration system with intercooling determined that the optimum inter-stage pressure is very close to the saturation pressure corresponding to the arithmetic mean of the condensing and evaporating temperatures and that most of the irreversibility losses stem from low compression efficiency (Zubair et al., 1996). Another study on the performance of a 2-stage R22 refrigeration system with intercooling determined the effect of condensing, evaporating, refrigerated medium and environment temperatures on the system's irreversibility rate (Nikolaidis and Probert, 1998). Molenaar (1992) investigated the performance of a cascade refrigeration system using 2 different refrigerant couples, namely R502/R13 and R22/R23, to find a replacing couple with a lower ozone depleting potential for R502/R13. A cascade heat pump system used for providing a hot water stream and utilising R12 refrigerant was developed and experimentally analysed in another study (Hasegawa et al., 1996). The effect of evaporating temperature on the performance of a cascade system using R22/R23 refrigerants was also examined (Cho et al., 2001). Kanoğlu (2002) performed an exergy analysis of a cascade refrigeration system consisting of 3 individual cycles and used for natural gas liquefaction. Kilicarslan (2004) presented the experimental performance of a cascade refrigeration system using R134a in both lower-temperature and higher-temperature cycles and relying on a water stream to exchange heat between the cycles.

As seen from the literature survey outlined above, a thorough comparison of the experimental performances of single-stage and cascade refrigeration systems has not been made yet. Therefore, the main objective of this study is to compare the performance characteristics of these systems using R134a as the working fluid while the secondary objective is to investigate the effect of using a refrigerated condenser water stream on the performance of a refrigeration system. For this purpose, 2 experimental plants were developed and instrumented. These plants employ a refrigeration cycle serving as a base unit for each system, a bench-top cooling tower and another refrigeration cycle serving as a higher-temperature unit for the cascade system. The performance of the experimental single-stage system using 3 different types of condensers, namely air-cooled, water-cooled and evaporative condensers, was presented in a previous study (Hosoz and Kilicarslan, 2004). In the present study, the single-stage operation was achieved using only a water-cooled condenser. The base unit and higher-temperature unit of the cascade system were thermally connected to each other by means of a water stream. Each refrigeration system was tested by varying refrigeration capacity in the base unit and the water flow rate passing through the condenser of the base unit. Then, the performance characteristics of both systems, namely condensing and evaporating temperatures, refrigerant mass flow rate, compressor power, coefficient of performance (COP), compressor discharge temperature, compressor volumetric efficiency and the ratio of compressor discharge to suction pressures, were determined and compared with each other.

Experimental refrigeration systems

Schematic diagrams of the experimental single-stage and cascade refrigeration systems are shown in Figures 1 and 2, respectively. The base unit of both systems consists of a reciprocating compressor, a shell-and-coil type water-cooled condenser, an internally equalised thermostatic expansion valve and an electrically heated evaporator. The base unit was charged with 600 g of R134a. The single-stage operation was performed by connecting the condenser in the base unit to a cooling tower, whereas the cascade operation was achieved by coupling the condenser in the base unit to the evaporator in another refrigeration cycle called the higher-temperature unit.

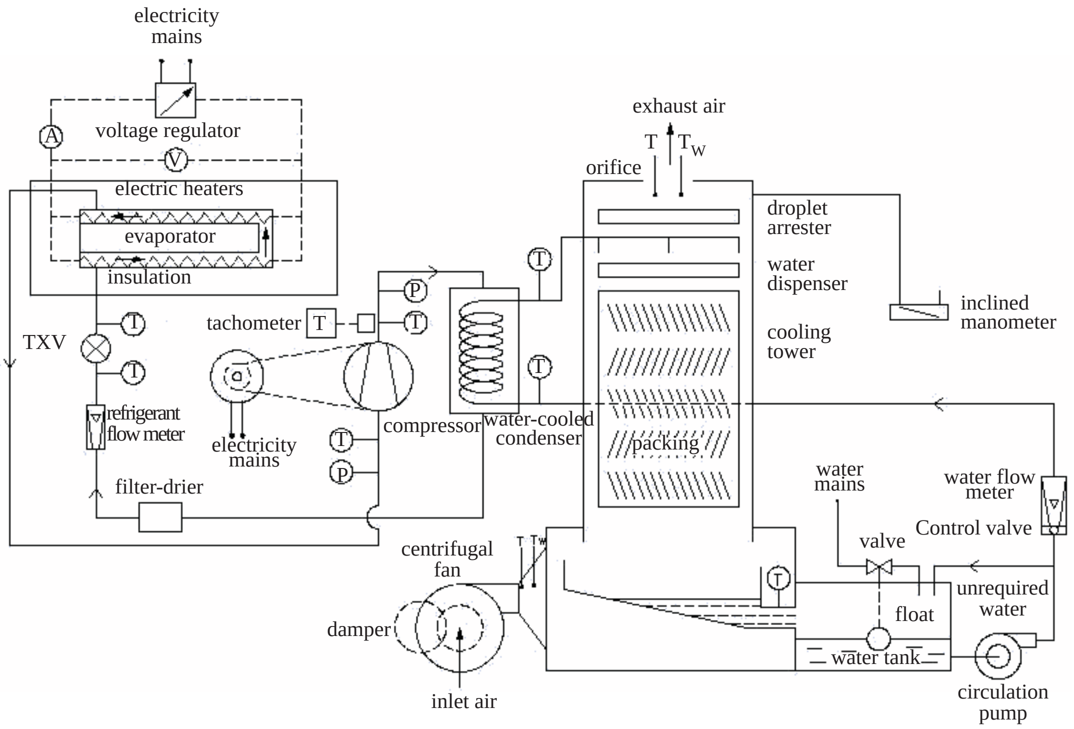

Figure 1. Schematic diagram of the experimental single-stage refrigeration system with a cooling tower.

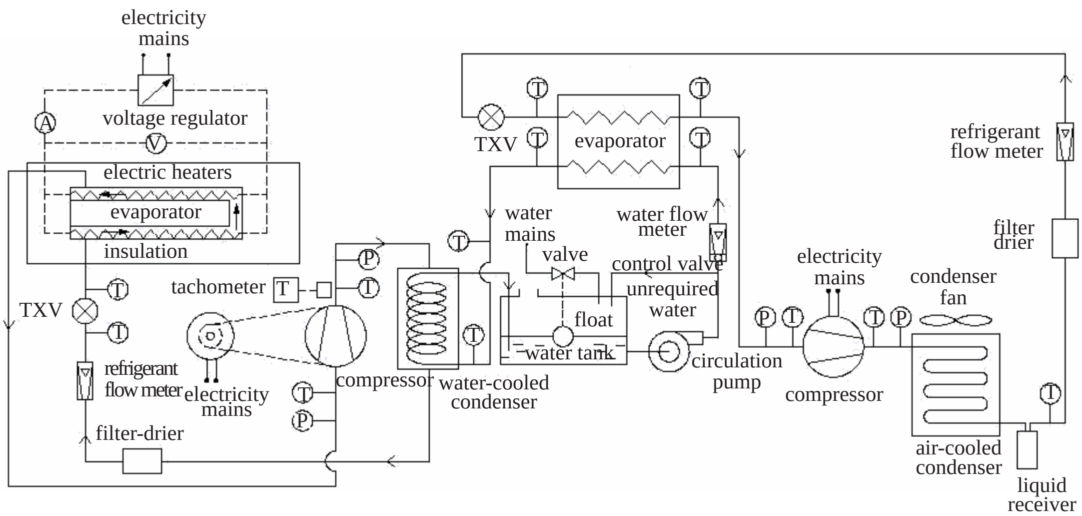

Figure 2. Schematic diagram of the experimental cascade refrigeration system.

The compressor employed in the base unit is a twin-cylinder open type one with a swept volume of 75.7 $cm^3 rev^{-1}$ and a nominal speed of 460 rpm. It was belt-driven by a single-phase electric motor. The water-cooled condenser consists of a vertical coil enclosed in a welded steel shell and has a heat transfer area of 0.075 m^2. The evaporator was made from copper tube with 2 separate electric resistance heaters rolled inside the tube. The refrigeration load was provided to the evaporator by varying the voltage across the electric heaters via a variable transformer.

The cooling tower coupled to the base unit in the single-stage operation consists of air and water circuit elements, and a column of packing material through which the 2 streams are brought into contact with each other. The column is 150 mm × 150 mm × 600 mm high and fabricated from clear PVC. It contains 8 decks of inclined and wettable plastic plates with a total transfer area of 1.14 m^2. Ambient air is pulled into the tower by means of a centrifugal fan at a rate determined by adjustment of the damper setting. After absorbing heat from the hot water stream coming from the condenser, the air stream discharges into the atmosphere through an orifice used for measuring the airflow rate. A circulation pump draws the cooled water from the tank of the tower, and sends it to the condenser through a hand-operated water control valve determining the water flow rate circulated in the circuit.

The higher-temperature refrigeration unit coupled to the base unit in the cascade operation consists of a hermetic compressor, an air-cooled condenser, a liquid-receiver, a filter-drier, an internally equalised thermostatic expansion valve and a tube-in-tube evaporator. This unit was charged with 750 g of R134a. The compressor in this unit has a swept volume of 8.85 $cm^3 rev^{-1}$ and a nominal speed of 2800 rpm. The condenser was made from copper tubing attached to aluminium fins.

The water circuit between the condenser in the lower unit and the evaporator in the higher unit contains a circulation pump, a small water tank, plastic tubing and a hand-operated valve controlling the water flow rate in the circuit. All elements in the refrigeration and water circuits of both experimental systems, and the pipelines were insulated with either polyurethane foam or elastomeric insulator.

Both single-stage and cascade refrigeration systems were located in an air-conditioned space where dry and wet bulb temperatures of the air could be maintained at desired values. This was achieved by employing heating and cooling coils and supplying ambient air to the space continually.

Figures 1 and 2 also indicate the locations where mechanical and electrical measurements were performed. Mechanical measurements consist of temperature, pressure and mass flow rate measurements conducted on both units including the cooling tower, while electrical measurements are the voltage across the electrical heaters and current flow through the heaters in the evaporator of the base unit. The heat input to the evaporator in the base unit was determined from the product of voltage and current draw. Furthermore, the rotational speed of the compressor in the base unit was continually monitored using a photoelectric tachometer. Some features of the instrumentation are summarised in Table 1.

All temperature measurements were performed using K-type thermocouples. Thermocouples for refrigerant temperature were soldered to the copper tube, while thermocouples for water and air temperatures were in direct contact with the fluid streams. Both dry and wet bulb temperatures of the air stream at the inlet and outlet of the cooling tower were measured. Compressor suction and discharge pressures in the base unit and in the higher-temperature unit were measured using Bourdon tube

Table 1. Characteristics of the instrumentation.

Measured Variable	Instrument	Range	Accuracy
Temperature	Type K thermocouple	-50/100 °C	0.3 °C
Pressure	Bourdon gauge	-100/600, 0/2000 kPa	5,20 kPa
Refrigerant flow rate	Variable area flow meter	0-20 $g\ s^{-1}$	5%
Water mass flow rate	Variable area flow meter	0-50 $g\ s^{-1}$	5%
Air mass flow rate	Orifice-inclined manometer	0-40 mmH_2O	1 mmH_2O
Compressor speed	Photoelectric tachometer	0-999 rpm	5 rpm
Voltage	Analogue voltmeter	0-250 V	2 V
Current	Analogue ammeter	0-10 A	0.1 A

gauges. Because the refrigeration lines are relatively short, it was assumed that evaporating and condensing pressures in both units were equal to the measured ones.

Refrigerant mass flow rates in both cycles were measured using individual variable-area R134a flow meters, located in the liquid line of each unit. The mass flow rate of the water stream through the condenser in the base unit was measured using another variable-area flow meter with a needle control valve. Air mass flow rate through the cooling tower was determined by measuring pressure difference across the orifice (ΔP) using an inclined manometer, finding density of the air leaving the tower (ρ_e) with the help of dry and wet bulb temperatures, and evaluating them in the following equation:

$$\dot{m}_a = \rho_e K_0 A_0 Y \sqrt{2\ \Delta P/\rho_e} \tag{1}$$

where K_0 is the flow coefficient, A_0 is the orifice cross-section area and Y is the expansion factor. Inserting values for these 3 constants into Eq. (1) and defining ΔP as a function of h_m, which denotes the orifice differential in mmH_2O, gives:

$$\dot{m}_a \cong 0.0137 \sqrt{h_m\, \rho_e} \tag{2}$$

Thermodynamic analysis of the experimental systems

The evaporator load on the base unit, i.e. the refrigeration capacity of the single-stage and cascade refrigeration systems, can be evaluated for the refrigerant and the heaters side. Assuming that the evaporator in the base unit was insulated perfectly, the evaporator loads for both sides can be equated:

$$Q_{evap} = \dot{m}_{r,base}\,(h_{evap,base,e} - h_{evap,base,i}) \cong V\,I \tag{3}$$

As seen in Eq. (3), the evaporator load for the refrigerant side utilises refrigerant mass flow rate and refrigerant enthalpies at the exit and inlet of the evaporator in the base unit, while that for the heaters side relies on the results of voltage and current measurements. Refrigerant enthalpies were evaluated using a software package for refrigeration (CoolPack, 2004). The evaporator load deviations between the 2 sides were usually within 5%, and only the heaters side results were used as the evaporator load due to their yielding lower uncertainties, as revealed by uncertainty analysis presented in the next section. Then the refrigerant mass flow rate in the base unit based on evaporator load for the heaters side can be determined from

$$\dot{m}_{r,base} = \frac{V\,I}{h_{evap,base,e} - h_{evap,base,i}} \tag{4}$$

The accuracy for the refrigerant mass flow rate measurements performed by the variable-area flow meter was equal to $\pm 5\%$, which was poorer than the uncertainty for the refrigerant flow rates evaluated from Eq. (4). Therefore, only the results of this equation were used as the refrigerant flow rate in the base unit, while the results of direct measurements were used for checking purposes.

Assuming that there is no heat transfer to or from the compressor, the power absorbed by the refrigerant during the compression process in the base unit can be determined from

$$W_{comp,base} = \dot{m}_{r,base}\,(h_{comp,base,e} - h_{comp,base,i}) \tag{5}$$

where $h_{comp,base,e}$ and $h_{comp,base,i}$ are the enthalpies of the refrigerant at the exit and inlet of the compressor, respectively.

The ratio of the evaporator load to the compressor power gives the energetic performance of the single-stage refrigeration system and that of the lower unit of the cascade system, i.e.

$$COP_{base} = Q_{evap}/W_{comp,base} \tag{6}$$

Assuming that the compression process in the higher-temperature unit is also adiabatic, the compressor power absorbed by the refrigerant in this unit of the cascade system can be expressed as

$$W_{comp,high} = \dot{m}_{r,high}\,(h_{comp,high,e} - h_{comp,high,i}) \tag{7}$$

where $\dot{m}_{r,high}$ is the refrigerant mass flow rate measured by the variable-area flow meter, and $h_{comp,high,e}$ and $h_{comp,high,i}$ are the enthalpies of the refrigerant at the exit and inlet of the compressor in the higher-temperature unit, respectively.

Then the energetic performance of the cascade refrigeration system can be determined from the overall coefficient of performance defined as

$$COP_{cas} = Q_{evap}/(W_{comp,base} + W_{comp,high}) \quad (8)$$

Volumetric efficiency of the compressor in the base unit can be defined by (Stoecker and Jones, 1982)

$$\eta_v = \frac{\dot{m}_{r,base}\ v_{comp,base,i}}{V_s\, n_{comp}} * 100 \quad (9)$$

where $v_{comp,base,i}$ is the specific volume of the refrigerant at the compressor inlet, V_s is the swept volume of the compressor, which is equal to 75.7 $cm^3 rev^{-1}$, and n_{comp} is the compressor speed.

Although the performance parameters of both refrigeration systems can be evaluated from Eqs. (3)−(9), energy balance equations for the condenser/cooling tower combination in the single-stage operation and for the lower unit condenser/higher unit evaporator combination in the cascade operation will also be presented below.

The heat rejection rate in the condenser of the base unit can be determined from

$$Q_{cond,base} = \dot{m}_{r,base}\ (h_{cond,base,i} - h_{cond,base,e}) \quad (10)$$

where $h_{cond,base,i}$ and $h_{cond,base,e}$ are the enthalpies of the refrigerant at the condenser inlet and exit, respectively. Assuming that the cooling tower is perfectly insulated and applying the principle of energy conservation to the water-cooled condenser/cooling tower combination in the single-stage system, the heat rejected by the refrigerant can be related to the heat absorbed by the air stream passing through the tower as follows:

$$Q_{cond,base} \cong \dot{m}_a[(h_a + w\,h_g)_e - (h_a + w\,h_g)_i] - \dot{m}_a[(w_e - w_i)\,h_f] - |W_p| \quad (11)$$

where $|W_p|$ is the power absorbed by water in the circulation pump, which is equal to about 0.1 kW, and $(h_a + w\,h_g)_e$ and $(h_a + w\,h_g)_i$ are the enthalpies of the moist air at the exit and inlet of the cooling tower, respectively. These enthalpies and specific humidities of the air at the exit and inlet of the tower, w_e and w_i, and specific enthalpy of the make-up water, h_f, were obtained from Coolpack (2004) using the results of temperature measurements. Thus, Eq. (11) offered a means to evaluate the heat rejection rate in the condenser of the single-stage system based on airside measurements. The results show that deviations between condenser heat rejection rates obtained from Eqs. (10) and (11) were usually within 5%, thus indicating the reliability of the experimental data based on refrigerant side measurements and calculations involving them.

Similarly, the heat rejected by the refrigerant in the lower unit condenser of the cascade system can be related to the heat absorbed by the refrigerant in the higher unit evaporator. Assuming that both the condenser in the lower unit and the evaporator in the higher unit were perfectly insulated, the heat rejection rate in the lower unit condenser can be evaluated from

$$Q_{cond,base} = \dot{m}_{r,high}\ (h_{evap,high,e} - h_{evap,high,i}) - Q_{gain} - |W_p| \quad (12)$$

where $h_{evap,high,e}$ and $h_{evap,high,i}$ are the enthalpies of the refrigerant at the exit and inlet of the higher unit evaporator, respectively, and Q_{gain} is the heat gain through the components of the water circuit. Consequently, Eq. (12) offered a means to evaluate the heat rejection rate in the lower-temperature unit of the cascade system using measurements on the higher-temperature unit. However, Eqs. (10) and (12) were used for evaluating Q_{gain} by equating the results since it is too difficult to determine Q_{gain} precisely using heat transfer correlations.

Description of the experimental procedure

Single-stage and cascade refrigeration systems were tested in steady state by varying the evaporator load (refrigeration capacity) and water mass flow rate passing through the condenser of the base unit. The evaporator load supplied to the evaporator in the base unit was changed between 170 and 486 W for the single-stage system, while it was changed between 247 and 805 W for the cascade system. Since higher evaporator loads caused elevated discharge pressures that may be detrimental to compressor valves, the single-stage system was not tested at loads over 486 W. The water flow rates in both systems were kept at 8 and 12 g s^{-1} for each evaporator load. During tests, the dry and wet bulb temperatures of the space containing refrigeration systems and the cooling tower were kept at 22.5 ± 0.3 °C and 19.5 ± 0.3 °C, respectively. Tests on the single-stage system were performed by opening wide the air damper at the cooling tower inlet, which yielded an air flow rate of about 60 g s^{-1}.

Steady-state conditions were assumed to have been reached when changes in temperature and pressure at the key points of the systems had ceased. It was accepted that when temperature deviations at the points considered were lower than 0.3 °C for 10 min, the steady state was achieved. Both systems were usually brought to steady state within 30 min of the input conditions being changed. As soon as stabilised conditions were achieved, data were collected to analyse the performance of the system being tested.

Uncertainty analysis

Uncertainty analysis for the calculated performance parameters of both refrigeration systems were performed using the method given by Moffat (1988). According to this method, the function R is assumed to be calculated from a set of totally N measurements (independent variables) represented by

$$R = R\,(X_1,\ X_2,\ X_3, ...,\ X_N) \tag{13}$$

Then the uncertainty of the result R can be determined by combining uncertainties of individual terms using a root-sum-square method, i.e.

$$\delta R = \left\{ \sum_{i=1}^{N} \left(\frac{\partial R}{\partial X_i}\,\delta X_i \right)^2 \right\}^{1/2} \tag{14}$$

Using accuracies for various measured variables presented in Table 1, uncertainties of the calculated parameters were determined with the evaluation of Eqs. (3)−(9) in Eq. (14). The uncertainties of Q_{evap}, $\dot{m}_{r,base}$, $W_{comp,base}$ and $W_{comp,high}$ estimated by the analysis are 4.4%, 4.4%, 16.2% and 16.5%, respectively. On the other hand, those of the COP_{base}, COP_{cas} and η_v are 16.8%, 7.9% and 6.9%, respectively.

Results and Discussion

Figure 3 indicates the evaporating temperature in the base unit as a function of the refrigeration capacity at 2 different condenser water flow rates for each refrigeration system. It is seen that the cascade system yields lower evaporating temperatures for any given refrigeration capacity due to better heat rejection in the condenser. In other words, the cascade system results in a greater refrigeration capacity for any given evaporating temperature. Moreover, the higher the water flow rate, the lower the evaporating temperature for both systems. Because the temperature of the refrigerated medium is directly related to the evaporating temperature, the cascade system can provide lower medium temperatures. For a water flow rate of 12 g s^{-1}, the ratio of the refrigeration capacity of the cascade system to that of the single-stage system ranges from a minimum of 1.21 (at −13.0 °C evaporating temperature) up to a maximum of 1.42 (at −27.5 °C evaporating temperature).

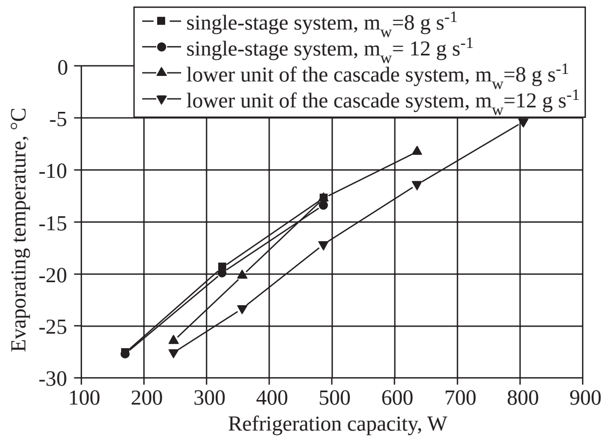

Figure 3. Evaporating temperature as a function of refrigeration capacity.

Figure 4 shows the condensing temperature as a function of the refrigeration capacity for each refrigeration system. It is clear that the cascade system results in lower condensing temperatures for any given refrigeration capacity as a result of lower water temperatures at the condenser inlet. Due to better heat rejection, a cascade system can absorb more heat than a single-stage system, thus providing a higher refrigeration capacity for any given evaporating temperature. Furthermore, as the water flow rate increases, the condensing temperature decreases for both systems.

Figure 5 depicts the refrigerant mass flow rate through the single-stage system and through both units of the cascade system for 2 different water flow rates. It is seen that the lower unit of the cascade system experiences slightly lower refrigerant flow rates than the single-stage system for any given refrigeration capacity. This originates from the fact that the cascade system yields lower evaporating temperatures and the increase in enthalpy of the refrigerant in vaporisation (h_{fg}) increases with decreasing evaporating temperatures, thus giving lower refrigerant flow rates for a given refrigeration capacity. The results show that the higher unit of the cascade system operates with higher refrigerant flow rates than the lower unit, which is due to a higher evaporator load on this unit. Moreover, the higher the water flow rate, the lower the refrigerant flow rate for both systems. Because increasing the condenser water flow rate promotes subcooling in the condenser, the refrigerant entering the expansion valve at a lower temperature has a lower enthalpy. As this refrigerant expands to the evaporating pressure, it has a lower quality. Refrigerant of low quality can absorb more heat per unit mass in the evaporator, thus yielding a lower refrigerant flow rate for a given refrigeration capacity.

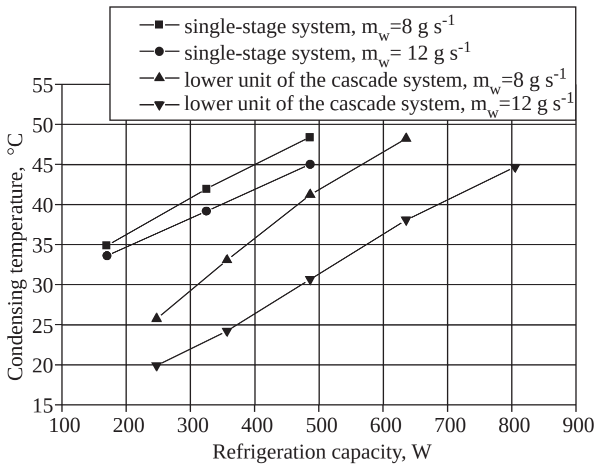

Figure 4. Condensing temperature as a function of refrigeration capacity.

Figure 6 exhibits the compression power against refrigeration capacity for both refrigeration systems. It is obvious that the lower-temperature unit of the cascade system absorbs less power than the single-stage system. This originates from the fact that the pressure ratio across the compressor in the lower unit of the cascade system is less than that in the single-stage system for the same refrigeration capacity. However, the higher-temperature unit absorbs greater compressor power than the lower one. Although compressor power absorbed in the lower unit of the cascade system falls with decreasing refrigeration capacity, the higher unit still absorbs large amounts of compressor power, thereby still causing high total compression power inputs to the entire system at low refrigeration loads. This is mainly attributable to the heat leakage into the water circuit between the condenser in the lower unit and the evaporator in the higher unit. By comparing the results of Eqs. (10) and (12), it is found that up to 30% of the refrigeration load on the evaporator in the higher unit is due to heat gain through the pipeline and circulating pump.

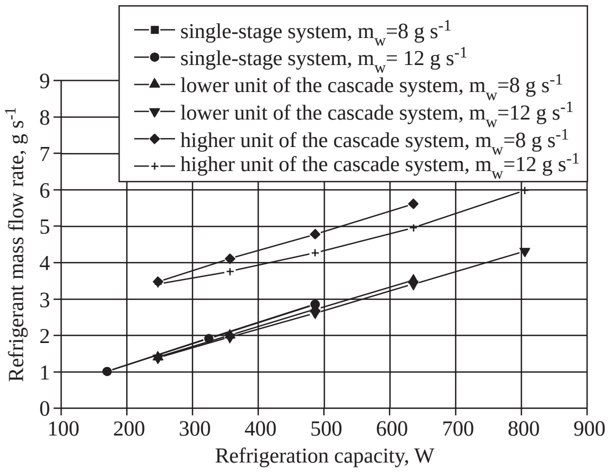

Figure 5. Refrigerant mass flow rate as a function of refrigeration capacity.

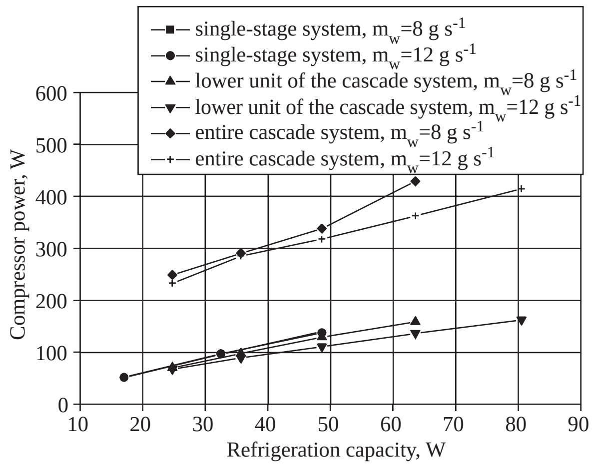

Figure 6. Compressor power as a function of refrigeration capacity.

Figure 7 shows the COP for both systems. The values regarding the cascade system are reported not only for the entire system but also for its lower unit. Although COPs for the lower unit of the cascade system are higher than those for the single-stage system, COPs for the entire cascade system fall below those for the single-stage system. This stems from increasing compression power due to the second compressor used in the higher unit and extra refrigeration load on the evaporator in the higher unit owing to heat leakage into the water circuit. It is clear that using a refrigerated water stream in the condenser of a refrigeration system causes a moderate increase in the refrigeration capacity at the expense of a severe increase in the total power consumption of the system, thus lowering the overall COP.

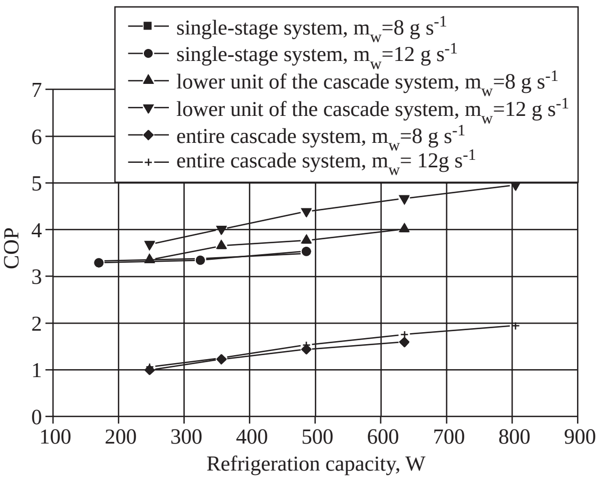

Figure 7. Coefficient of performance as a function of refrigeration capacity.

Figure 8 indicates the compressor discharge temperature as a function of refrigeration capacity. When compared with the single-stage system, the lower unit of the cascade system experiences lower discharge temperatures due to lower temperatures of the heat-rejected medium, while discharge temperature decreases with increasing water flow rate. It is known that the higher the discharge temperature, the higher the possibility of thermal destruction of the lubricating oil, consequently causing excessive wear and decreasing durability of the compressor.

The ratio of compressor discharge pressure to suction pressure against refrigeration capacity is shown in Figure 9. Because the lower unit of the cascade system operates with lower condensing temperatures, the pressure ratio across the compressor of this unit is lower than that of the single-stage system. This low pressure ratio lowers the compressor power and discharge temperature and decreases the possibility of compressor failure. The results show that the higher the water flow rate the lower the pressure ratio for both the single-stage system and the lower unit of the cascade system. However, because the refrigeration capacity increases with water flow rate, higher flow rates cause a higher amount of heat rejection in the condenser of the lower unit, thus increasing the evaporator load on the higher unit of the cascade system. Consequently, the pressure ratio in the higher unit slightly increases with water flow rate.

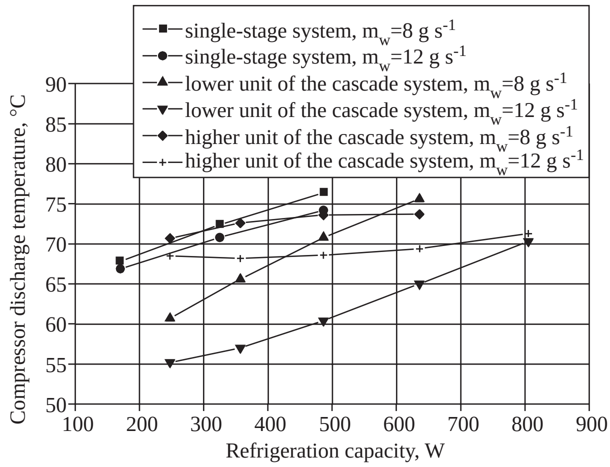

Figure 8. Compressor discharge temperature as a function of refrigeration capacity.

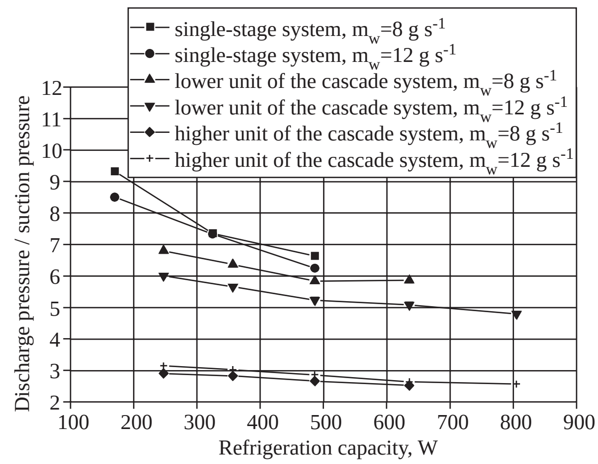

Figure 9. The ratio of compressor discharge pressure to suction pressure as a function of refrigeration capacity.

Figure 10 depicts the compressor volumetric efficiency as a function of refrigeration capacity for both the single-stage system and the lower unit of the cascade system. Because it is too difficult to measure the actual speed of the compressor in the higher unit due to its sealed structure, its volumetric efficiency was not determined. Volumetric efficiency, defined as the ratio of the refrigerant volume flow rate entering the compressor to the displacement rate of the compressor, is an important factor for predicting the performance of reciprocating compressors. The results show that volumetric efficiency increases with increasing refrigeration capacity and increasing water flow rate. Comparing the graphs in Figures 9 and 10, it is seen that volumetric efficiency increases with decreasing pressure ratio over the compressor. This means that the lower the pressure ratio the lower the compressor power for a given refrigerant flow rate.

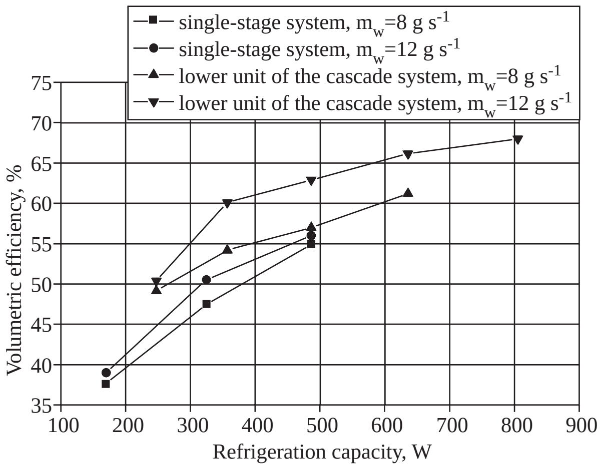

Figure 10. Compressor volumetric efficiency as a function of refrigeration capacity.

Conclusions

The performance characteristics of single-stage and cascade vapour-compression refrigeration systems were experimentally determined and compared with each other. Moreover, the effect of using a refrigerated water stream in the condenser of a refrigeration system was investigated. Considering the experimental information gathered in this study, it is possible to draw the following conclusions.

- Due to lower water temperatures at the condenser inlet and resulting lower condensing temperatures, the cascade system yields greater refrigeration capacities for any given evaporating temperature or it provides lower evaporating temperatures for any given refrigeration capacity compared with the single-stage system.
- The cascade system yields a lower compressor power and a lower refrigerant mass flow rate in the lower-temperature unit compared with the single-stage system. Although compressor power in the lower unit decreases with a decrease in condensing temperature, total compressor power in the cascade system is greater than that in the single-stage system due to the second compressor utilised in the higher-temperature unit.
- The COP for the lower unit of the cascade system is higher than that for the single-stage system. However, the cascade system experiences lower overall COP values due to the elevated power demand of the higher unit compressor, meaning that it requires more power to provide the same refrigeration capacity. Moreover, the COPs for both systems increase with increasing water flow rate passing through the condenser in the base unit.
- Although the use of a refrigerated water stream in the condenser of a refrigeration system increases refrigeration capacity significantly, the COP for the overall system drops drastically, thus causing energy inefficiency.
- The compressor discharge temperature for the single-stage system is higher than that for the lower unit of the cascade system. Similarly, the pressure ratio across the compressor in the single-stage system is higher than that in the lower unit of the cascade system. Lower discharge temperatures and pressure ratios imply an increased durability for the lower unit compressor of the cascade system.
- The volumetric efficiency of the compressor in the single-stage system is lower than that in the lower unit of the cascade system, implying a higher compressor power for a given refrigeration capacity compared with the lower unit compressor.

The poor energy efficiency of the cascade system is attributable to not only the employment of 2 compressors but also heat leakage into the water circuit coupling the condenser in the lower unit to the evaporator in the higher unit. This leakage can partially be avoided by using a single heat exchanger, the cascade condenser/evaporator, instead of utilising 2 separate heat exchangers, thus cancelling out the water stream. Another reason for the inefficiency of the cascade system is the overlap of the condensing temperature in the lower unit and the evaporating temperature in the higher unit. This can also be avoided by using a 2-stage refrigeration system with an intercooler at the expense of the oil return problems mentioned before. Furthermore, the cascade system will perform better if the lower and higher units utilise a higher-pressure and a lower- pressure refrigerant, respectively.

Nomenclature

A_0	orifice cross-section area (m^2)
COP	coefficient of performance
h	specific enthalpy of the refrigerant (J kg^{-1})
h_a	specific enthalpy of the air (J kg^{-1})
h_f	specific enthalpy of the make-up water (J kg^{-1})
h_{fg}	increase in specific enthalpy in vaporisation (J kg^{-1})
h_g	specific enthalpy of the water vapour (J kg^{-1})
h_m	orifice differential (mmH_2O)
I	current flow through the heaters (A)
K_0	flow coefficient
$\dot{m}$	mass flow rate (kg s^{-1})
N	total number of independent variables in function R
n_{comp}	compressor speed (rps)
Q_{cond}	heat rejection rate in the condenser (W)
Q_{evap}	refrigeration capacity (W)
Q_{gain}	heat gain through the components of the water circuit (W)
R	a function of independent variables
v	specific volume of the refrigerant (m^3 kg^{-1})
v_e	specific volume of the air at the exit of the cooling tower (m^3 kg^{-1})
V	voltage across the heaters (V)
V_s	swept volume of the compressor (m^3 rev^{-1})
w	specific humidity
W_{comp}	compressor power (W)
$\|W_p\|$	power absorbed by water in the circulation pump (W)
X	independent variable
Y	expansion factor
η_v	compressor volumetric efficiency
ρ_e	density of the air at the exit of the cooling tower (kg m^{-3})
ΔP	orifice pressure drop (Pa)

Subscripts

a	air
base	base unit
cas	cascade system
comp	compressor
cond	condenser
e	exit
evap	evaporator
high	higher-temperature unit of the cascade system
i	inlet
r	refrigerant

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