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DESIGN AND THERMAL ANALYSIS OF A CONDENSER WASTE HEAT RECOVERY VAPOUR COMPRESSION REFRIGERATOR WITH AUGMENTED ACCELERATED FLOW EVAPORATOR

Praveen Maruthur, G. Ragul , Mohammed shebin K B, Yadhukrishnan T M, Muhammed Anzari T, Naveen R and Salim Jahanger
Department of Mechanical Engineering, Nehru College of Engineering & Research Centre, Thrissur, India
E-Mail: ragulme90@gmail.com

ABSTRACT

The performance of an alternative evaporator design for household applications along with condenser waste heat recovery is investigated in this paper. In this novel concept condenser waste heat from refrigerator is trapped and utilized while the geometric parameter of the evaporator is changed where outlet air area is reduced progressively from inlet. Thus reducing material cost of evaporator. Experiments have been conducted in an optically accessible test rig using R 134-a refrigerant to determine the performance of varying area accelerated flow evaporator and condenser waste heat recovery and finding the dependency of performance of the test rig on the mass flow rate of air, refrigerant and varying contact area. The procedure relies on the plane tube exchanger surfaces and mass and heat balances to determine the flow rate of air and coefficient of performance enhancement.

Keywords: vapour compression refrigeration, varying area augmented evaporator, coefficient of performance, waste heat recovery.

INTRODUCTION

Waste heat is generally the energy associated with the waste streams of air, gases and liquids that leaves the boundary of the system and enter into environment. The essential quality of heat is not the amount but its value. Waste heat recovery and utilization is the process of capturing and reusing waste heat for useful purposes. Not all waste heat is practically recoverable. The strategy of how to recover this heat depends on the temperature of the waste heat sources and on the economics involves behind the technology incorporated. Cooling generates considerable quantities of heat. If not utilized, this energy simply becomes waste heat. The cooling may be for the process cooling, air conditioning or other use. Vapor compression Refrigeration system is an improved type of air refrigeration system. The ability of certain liquids to absorb enormous quantities of heat as they vaporize is the basis of this system. Compared to melting solids (say ice) to obtain refrigeration effect, vaporizing liquid refrigerant has more advantages. To mention a few, the refrigerating effect can be started or stopped at will, the rate of cooling can be predetermined, the vaporizing temperatures can be governed by controlling the pressure at which the liquid vaporizes. Moreover, the vapor can be readily collected and condensed back into liquid state so that same liquid can be re-circulated over and over again to obtain refrigeration effect. Thus the vapor compression system employs a liquid refrigerant which evaporates and condenses readily. The System is a closed one since the refrigerant never leaves the system.

The coefficient of performance of a refrigeration system is the ratio of refrigerating effect to the compression work; therefore the coefficient of performance can be increased by increasing the refrigerating effect or by decreasing the compressor work. Thus all the heat removed from the process, plus most of the energy added by the compressor is rejected to the local environment. The evaporator has an important role in the determination of the system performance because it is responsible for providing the cooling capacity required for preserving the goods stored in the refrigerator at the desired temperatures. Hence, improving the performance of the evaporator is potentially significant as a means of improving the performance of the whole system and, consequently, as a means of promoting material cost savings.

Cur and Anselmino (1992) proposed an alternative configuration of a tube-fin evaporator for 'no-frost' domestic appliances, the so-called Accelerated Flow Evaporator (AFE). The main purpose of this concept is to reduce the size of the evaporator (and hence the volume of aluminum) by enhancing the local air-side heat transfer coefficient. This method helped in achieving reduced material usage with reduction in air-side cross sectional area causing an increase in mean velocity of air stream and local Reynolds number which is factor of velocity of flowing fluid and it plays the major role in convection heat transfer processes.

This paper makes an experimental investigation on recovering waste heat given out from the condenser of the refrigerator, improving heat transfer rate across the walls of the evaporator by increasing the mean air stream velocity and thereby increasing mean air steam velocity and directly enhancing the air-side heat transfer coefficient.

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EXPERIMENTAL SETUP

Refrigerant

Refrigerant chosen for this test rig is R134a due to its properties depicted below includes the economic consideration, latent heat of vaporization, specific volume of vapor, compression ratio and specific heat of refrigerant in both liquid and vapor states. Considering the above factors refrigerant R-134a (Tetrafluoroethane, CH_2F-CF_3) is selected as the refrigerant for combo-refrigerator.

Table-1. Properties of R-134a.

Name	Boiling point (K) @ 1 bar	Freezing Point (K) @ 1 bar	Critical Temp (K) @ 1 bar	Critical Pressure (bar)
R-134a	247	176.55	374.25	40.67

Evaporator

The evaporator coil was made from the copper tubes of diameter 5 mm and length 3.5 m.

Properties of Copper

Density - 8954 kg/m^3

Thermal diffusivity - 0.404 m^2/h

Specific heat - 0.091 keal/kg ^{0} c or 0.3kj/kg k

Thermal conductivity - 386 w/mk

The refrigerant circuitry was designed to provide a full counter-flow heat exchanger configuration.

Figure 1 shows a side view of the evaporator sample illustrating the cross-sectional area reduction in the direction of the flow.

Figure-1. Accelerated flow evaporator.

Condenser with waste heat recovery

The condenser is a heat transfer surface where heat from the hot refrigerant vapor is rejected to the condensing medium. The vapor refrigerant rejects its latent heat of vaporization and gets condensed into liquid state. There are different types of condensers and selection of condensers depends upon the capacity of the system, refrigerant used and medium of cooling available. The following factors are considered in the design of condensers. Specific heat of the cooling medium, Type of refrigerant, Velocity of cooling medium, Area of condensing surface._

Figure-2: Condenser waste heat recovery with blower.

Refrigerant has been chosen, depending the working pressures condenser and evaporator. These two pressures are integral to the design of the system, and there are several constraints that govern their selection. It is known that the saturation temperature of the refrigerant at the condenser pressure should be at least 10-15 degrees above the temperature of the environment. From that criterion, the condenser pressure chosen.

$$P_{condenser} = 10.10^5 \text{ pa}$$

The choice of evaporator pressure is not arbitrary; rather it is based on volumetric efficiency and pressure ratio. The compression pressure ratio is the ratio of condenser pressure to evaporator pressure and will have an effect on the volumetric efficiency. Our design uses a hermetically sealed reciprocating compressor, about which several factors are known:

V_s $= 15\frac{cm3}{rev}$ swept volume

V_c $= 0.05\ V_s$ clearance volume

n $=1.05$ polytrophic exponent

ω_{motor} $= 2800\frac{rev}{min}$ rotational speed of compressor motor

$$\eta_n = 1-\frac{Vc}{Vs}\{(P_{ratio})^{1/n} - 1)\} \text{ volumetric efficiency}$$

Volumetric efficiencies of at least 0.8 are desirable and are achievable. Based on this, the pressure ratio was chosen. Furthermore, the choice of pressure ratio will determine the pressure of the evaporator

$$P_{ratio} = 3.55 = P_{evaporator} = \frac{\text{P condenser}}{\text{P ratio}}$$

$$P_{evaporator} = 2.254\times10^5 \text{ Pa}$$

The volumetric efficiency for our chosen pressure ratio is η_v =0.883.

Mass flow rate of refrigerant

Based on the values for pressure and volumetric efficiency, the mass flow rate for both the refrigerant and the water can be calculated. Using the conservation of mass, it can be shown that the mass flow rate for the refrigerant is:

$m_{ref} = \frac{Va}{V1} \times \omega_{motor}$

The actual volume of the refrigerant V_a is defined as:

$V_a = \eta_n . V_s$

$V_a = 13.243\ cm^3$

Therefore, the mass flow rate of refrigerant can be determined by calculating the specific volume at the given state. In order to do this, an analysis of the system at state 1 must be performed:

T_{sat} = 263.06 K Sat. temperature of R-134 a at 2 bar

$P_1 = P_{evaprator}$ Pressure at state 1

$P_1 = 2.254 \times 10^5$ Pa

Since two independent properties are known, any other property of the state can be determined. From data handbook, the following values can be interpolated:

$v_1 = 0.15508 \frac{m3}{kg}$ Specific volume at state 1

$h_1 = 262.321 \frac{kJ}{kg}$ Specific enthalpy at state 1

$S_1 = 1.012 \frac{kJ}{kgk}$ Specific entropy at state 1

This leads to a refrigerant mass flow rate of:

$m_{ref} = 5.37 \times 10^{-3} \frac{kg}{s}$

Power requirements

The work done by the compressor must be calculated. The work done can be calculated using the properties of the system. It can be shown that the work is defined as:

$$W_{compressor} = \frac{Pcondensor.(V2) - Pevaporator.(V1)}{1-n}$$

The only unknown in this equation is the specific volume at state 2, however, that can be calculated using the other known properties:

$V_2 = \left(\frac{P_{evaporator}}{P_{condensor}} \times v_1{}^n\right)^{\frac{1}{n}}$ Specific volume at state 2

$V_2 = 0.034 \frac{m^3}{kg}$

This leads to a work of

$W_{compressor} = -32.55 \frac{kJ}{kg}$ Work done by the compressor

Given that the mechanical efficiency of the system is 0.85, the power required to operate the system can be determined.

Isentropic efficiency

$\eta_{compressor} = 0.85$, $W_{cycle} = \frac{W_{cycle}}{\eta_{compressor}}$ power required

$W_{cycle} = -37.947 \frac{kJ}{kg}$

Isentropic efficiencies can be used to relate the actual performance of a device under certain inlet and outlet conditions to the performance under ideal circumstances with the same environment conditions. Isentropic efficiency for a compressor can be written as:

$$\eta_{compressor} = \frac{T_2 - T_{evaporator}}{T_{2s} - T_{evaporator}}$$

The temperature of the evaporator is already known, and the remaining unknowns, can be calculated as follows: from data book for 8 bars

$T_2 = T_{condensor} = T_{condensor} = 31.33K + 273.15K$

With an evaporator pressure of 2 bar, the saturation temperature for the refrigerant at the inlet to the compressor can be determined from the tables as:

$T_2 = 263.06K$

For proper operation, this inlet temperature at the compressor is 3 degrees above the saturation temperature for the refrigerant at the evaporator pressure. Therefore, the temperature at the inlet of the compressor should be:

$T_{evap} = T_{sat\,2} + 3\ K = T_{evap} = 266.06\ K$

Since the process is isentropic, the following is known about the entropy at each state:

$$S_{2s} = s_1 = S_{2s} = 1.012\ \frac{kJ}{kgk}$$

Knowing this allows the determination of T_{2s} from Property table

$$T_{2s} = 289.15\ K$$

From this temperature and the temperature at the inlet of the compressor, the isentropic efficiency can be determined:

$\eta_{comp\ ise} = 0.601$

Rate Of Heat Loss From The Condenser

The rate of heat loss is calculated using the first law. H_{2s} is interpolated from data book.

$$h_{2s} = 263.890\ \frac{kJ}{kg}$$

Heat loss is then determined to be

$Q_{comp} = w_{cycle} m_{ref} + m_{ref}(h_1 - h_2) Q_{comp} = -212.217\ w$

RESULTS AND DISCUSSION

The system was tested using the measuring instruments and the performance of the system was analyzed using the values measured.

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Table-2. Temperature readings of refrigeration rig with WHR chamber.

TIME	T_1 (^0C)	T_2 (^0C)	T_3 (^0C)	HOT CHAMBER TEMPERATURE T_H(^0C)
5	8.7	8.8	12.7	33
10	4.7	6.3	8.3	33.8
15	3.5	3.6	3.6	41
20	3.4	3.4	3.4	43
25	3.2	3.1	3.2	44
30	3.1	3.0	2.8	44
35	2.8	2.6	2.4	45
40	2.7	2.5	2.0	47
45	2.3	2.0	1.6	48
50	1.4	1.0	1.0	51

Table-1 depicts the temperature readings at three various points on the evaporator coil as T_1, T_2, T_3 where T_1 is the temperature measured at the walls of the evaporator near air inlet to the evaporator tube bank and T_2, is the intermediate region temperature and T_3 is the one taken at the air outlet region of the evaporator tube bank while T_H temperature reading inside the waste heat recovery chamber as hot chamber temperature.

Coefficient of performance

The coefficient of performance for the designed unit is calculated using the Carnot efficiency formula.
Initial temperature of water (1 liter) =303 K
Final temperature of water = 279 K

COP conventional = $\frac{Q_{evaporator}}{W_{compressor}}$

$Q_{evaporator} = mc_{p\ water}\ \Delta T$

= 1 × 4.187 × (303-279)
= 100.488 kJ

$W_{compressor}$ = 32.55 kJ

COP = $\frac{100.488\ kJ}{32.55\ kJ}$
= 3.086

$COP_{W.H.R} = \frac{Q_{evaporator}+Q_{W.H.R}}{W_{compressor}}$

$Q_{W.H.R} = mc_p\Delta T$
= 1 × 4.187 × (318-288)
= 125.61 kJ

$COP_{W.H.R} = \frac{100.48 + 125.61}{32.55}$

$= \frac{226.09}{32.55}$
= 6.946

% Increase in COP = $\frac{COP_{W.H.R} - COP_{conventional}}{COP_{conventional}}$

$= \frac{6.945-3.086}{3.086} \times 100$
=125 %

Air flow rate using anemometer is 3m/s. The heat exchanger is recuperative type. Considering the dry air at 15 °C.

Reynolds number;

$Re = \frac{\rho vl}{\mu}$

$= \frac{1.247+1.205}{2}$

= 1.226 Kg/m^3
V= 3 m/s

$\mu = \frac{(17.65\times10^{-6})+(18.14\times10^{-6})}{2}$

$= 1.7895\times 10^{-5}\ \frac{Ns}{m^2}$

$Re = \frac{1.226\times3\times.1972}{1.7895\times10^{-5}}$
= 40530.96
= 4× 10^4.

Nusselt Number ;

$Nu = [0.43+0.50Re^{.5}]\ Pr^{0.38}$

$= .43+ (0.50\times(4\times10^4)^{0.5}).704^{0.38}$
= 87.944

Prandtl number of dry air at 15°C = .704

$Nu = \frac{h_\circ L}{K}$

$87.944 = \frac{h_\circ\times.1972}{K_{air}}$

$h_\circ = \frac{87.944\times0.025525}{.1972}$

$h_\circ$ = 11.38 w/m^{2}k
Internal flow
Considering refrigerant R134a,

$Re = \frac{\rho vl}{\mu}$
= 4.515× 2.4 × 1972
= 9712.99

Length of one coil = 0.1972 m
For Flow through tubes if Reynolds number is over 2500; the flow is turbulent.
Re = 9712.99 > 2500, hence turbulent.
For Fully developed flow, from Reynolds analogy and Heat and mass transfer data book
For Flow through a tube

$Nu = 0.023\ Re^{0.8}\ Pr^{0.4}$

$Pr = \frac{\mu c_p}{k}$ at 1bar, 6°C ;

$Pr = \frac{220 \times .82356}{85 \times 10^3}$
= .002

$Nu = 0.023 \times 9712^{0.8} \times .002^{0.4}$
= 2.96

Heat transfer coefficients have been calculated using the following closure relationships.

$Nu = \frac{h_i L}{K} = h_i = \frac{Nu \times K}{L}$
$= \frac{2.96 \times 85}{0.1972}$

$h_i = 1277.90$ w/m^{2}k

Overall heat transfer coefficient,

$U = \frac{1}{\frac{1}{h_i} + \frac{L}{k_{cu}} + \frac{1}{h_\circ}}$
$= \frac{1}{\frac{1}{11.38} + \frac{1}{386} + \frac{1}{1277.90}}$
= 11.052 w/m^{2}k
Q = UA (LMTD)

$= U \times n \times 2\Pi rl \times \frac{(\theta_1 - \theta_2)}{\ln\left(\frac{\theta_1}{\theta_2}\right)} = 15.89\ w$

Effectiveness, $\mathcal{E} = \frac{Q}{Q_{max}}$

$= \frac{15.89}{100.48} = 0.158$
$\mathcal{E} = 15.8$ %

CONCLUSIONS

This paper presented experimental data on an alternative concept for a household refrigeration evaporator. In this novel concept, the air-side cross-sectional area decreases with distance from the inlet, accelerates the flow and promotes an enhancement of the local heat transfer coefficient in a built experimental facility. A model has been proposed for calculating the heat transfer capacity in the AFE taking into account the flow rate through the side gaps. The model, which possesses no fitting parameters, predicted the experimental heat transfer ±10% with an efficient condenser waste heat recovery arrangement. The study showed that for low heat transfer capacities, the performances of the baseline and accelerated flow evaporators are quite similar as a small pumping power is required for a specified capacity Thus, under such conditions, the AFE concept may be more advantageous than the straight evaporators because of its lower material costs. With improved COP as the unwanted heat is recovered which indeed improved and extended the applicability of an conventional Vapor compression refrigeration par its boundary.

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