Experimental Analysis of Refrigeration system using Microchannel condenser & Round Tube condenser

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ABSTRACT -Micro channel condenser now days can be effectively used due to its compact size in automobile sector. For its performance, refrigeration set up designed to detect experimental performance of microchannel condenser. In this paper performance analysis of microchannel condenser compared with round tube and coil tube. In analysis of microchannel condensers it can be found more effective at various loads and operating conditions. For review same size of microchannel and round tube condenser are considered. From the previous experiments the micro-channel condenser was made to have nearly an identical face area, depth and fin density as the round-tube condenser which was the baseline. Also varying the refrigerants, C.O.P & Efficiency of micro channel the various reviews of reviewer micro channel condenser can be efficient and also refrigerator system requires less power.

Key words: Microchannel condenser, Round tube condenser, COP, Efficiency, refrigeration.

I.INTRODUCTION

The purpose of this review is to experimentally estimate improved condenser and evaporator capacity and system COP by replacing a round-tube condenser with a micro channel condenser having almost identical frontal area and depth, without considering heat exchanger cost. The baseline (round tube) condenser along with all other elements of the system was part of a carefully sized air-conditioning system.With the advent of refrigeration, interest in condensation has increased since a condenser is one of the main components of the basic vapor-compression refrigeration. Fig.(1) shows a schematic of a full microchannelcondenser. The tubes are brazed to the headers with louvered fins in between. The refrigerant is circuited using baffles inside the headers, involving more than one tube in each pass. This reduces the total pressure drop of the condenser due to fewer passes.These heat exchangers provide the same amount of heat transfer with a much lower refrigerant charge (about 2/3 that of a serpentine condenser) than condensers now commonly used in the industry. This attribute is very beneficial to car manufacturers as any reduction in the space.Also, a smaller charge requirement translates into a lower cost for installation and less of a threat to the environment should the system ever leak.

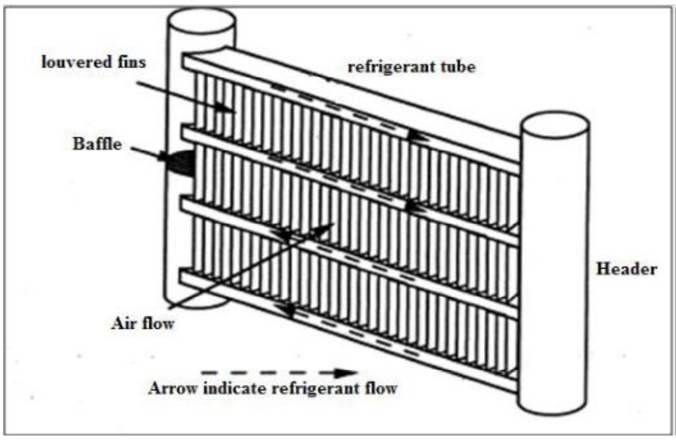

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Fig.1. Microchannel condenser

Now, a days while considering the global environment it is necessary to overcome ozone depletion, greenhouse effect, and other problems associated with the current state of technology, research in refrigeration has grown even more. Now, research focuses on determining the properties and effects of using alternative, non chlorofluorocarbon (CFC) refrigerants such as 1, 1,1,2-tetrafluoroethane (R-134a) as replacements for the commonly used CFC's. Time is being wisely spent on new refrigeration component technology in order to protect our finite environment. This project will follow suit by using ozone-safe R-134a and R-190a as its working fluid. The future use of the microchannel condenser technology is predicted to increase substantially over the next decade. The new government rules relate to refrigeration system can be easily followed if micro channel condenser with R134a and R190a can be used as a refrigerant.

I.EXPERIMENTAL SETUP

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In this project three types of condenser are used
1.micro channel condenser
2.Round tube condenser
3.Coil- tube condenser
In this experiment using refrigerant R134a,cop of refrigerant is calculated . Compressor passes refrigerant to two condenser one after another cop of both the system is thus calculated.

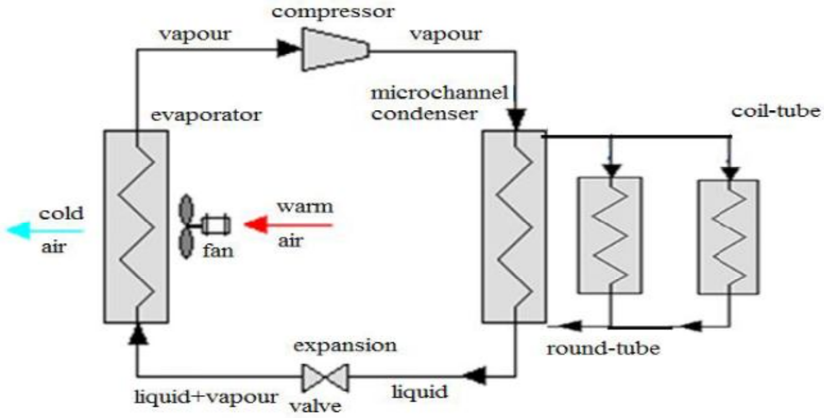


Fig 2. Schematic experimental setup

Fig 3.Actual setup

Fig 4.Actual setup

III.PROCEDURE

1. Connect the two plugs to main.
2. Before ON the supply, conform that all the switches on panel are off position.
3. See the dimmerstat is at zero position.
4. Then put ON the heater switch & give power to heater.
5. This will heat the water in evaporator & this can be seen dial thermometer.
6. Adjust the heater voltage such that the Temperature dial thermometer reading reaches 25 - 300 C.Now ON the D.P. switches.
7. Put ON the condenser fan switch & wait for 2 - 3 minutes.
8. Now switch ON the solenoid valve switch & the compressor switch.
9. The refrigeration flow will start. This can be confirmed on the sight glass.
10. Now the ammeter, voltmeter will show the current & voltage for compressor.
11. Note down the time for 10 revolutions of energy for compression. After some time we will see that the Temperature of water in the evaporator slowly goes down & reaches steady state.(Adjust this temp. at 28 to 300 C).

Basic paremeters –

1. HP Condenser pressure in Kg/cm2. = Kg/cm2
2. LP Evaporator Pressure in Kg/Cm2 = Kg/cm2
3. Rotameter in Reading LPH = LPH
4. Condenser Inlet Temperature in 0C = T1
5. Condenser Outlet Temperature 0C = T
6. Evaporator Inlet Temperature in 0C = Tci
7. Evaporator Outlet Temperature 0C = Teo
8. Time for 10 Pulses of heater energymeter = in sec
9. Time for 10 Pulses of comp energymeter = in sec.
10. Ammeter reading = in Amp
11. Voltmeter reading = in V
12. Evaporator Bath Temp in 0C = 0C

IV.SPECIFICATIONS

1. COMPRESSOR : Hermetically sealed type
   Kirloskar Make 1/3 T. Capacity.
2. AIR COOLED CONDENSOR : Copper Coils with fins & cooling fan.
3. EXPANSION DEVICE : i) Thermostatic expansion valve.
   ii) Capillary Tube
4. EVAPORATOR : Copper Coil immersed in water. The evaporator is installed from outside to prevent heat loss.
5. ENERGY METER : One each for power supply to the Measurement of Compressor & Evaporator Heater.
6. DIMMERSTAT : To control power supply to heater.
7. PRESSURE GAUGE : One each for the measurement of high & low pressure.
8. ELECTRIC HEATER : Immersion type 1.5 Kw
9. DIGITAL TEMPERATURE INDICATOR : To Measure the temperatures at various Points i. e. Evaporator Inlet & Outlet Condenser Inlet & Outlet temperatures of Refrigerant & Evaporator bath Temperature.
10. SOLINOID VALVE : Castle, Italy, Mate.
11. HP & LP CUTOUT : Safety device suitable for the low & high pressure of compressor.
12. SERVICE VALVE : Needle type for changing gas.
13. AMMETER : 0 - 15 Amps for compressor
14. VOLTMETER : 230 V, A.C. for compressor
15. FILTER DRYER : 1 No.
16. THERMOSTAT : Safety Device.
17. SWITCHES : For various controls

The refrigerant circuit is mounted on a board. The unit is supported on frame. Do not operate the changing valve. See the sufficient water is filled in the evaporator container.

V.OBSERVATIONS

1. Compressor Energymeter Constant (E M C) com = 3200 Pulses/kWatt hr
2. Heater Energymeter Constant (E M C) h = 3200 Pulses/kwatt hr

HP : Condenser Pressure in PSI
LP : Evaporator Pressure in PSI
Q : Refrigerant flow Rate in LPH

T1 : Refrigerant inlet Temp. To Condenser in 0C
T2 : Refrigerant Outlet Temp. from microchannel condenser in 0C
T3:; Refrigerant outlet from round tube condenser
T4 : Refrigerant outlet from coil-tube condenser
T5 : Inlet temperature to evaporator
T6 : Outlet from evaporator
T7 : Inlet temperature of ice plant
T8 : Outlet temperature of ice plant
T9 : Bath temperature evaporator1
T10: Atmospheric temperature
T11: Air outlet temperature of michrochannel condenser
T12: Outlet tc : Time for 10 Pulses of Compressor energy meter in sec.

OBSERVATION TABLE

Sr. No.	Temperature	Microchannel condenser			Round tube condenser			Coil tune condenser		
		I	II	III	I	II	III	I	II	III
1	T1	56.90	60	61.1	33.9	35.6	36.7	33.7	34.9	35.9
2	T2	32.9	33	33.81	32.83	33.9	34.2	32.8	33.5	34.6
3	T3	33.7	32.7	32.8	32.6	34.4	34.4	32.8	35.1	34.2
4	T4	33.2	33.4	32.7	31.5	37.9	31.8	32.01	36.8	32.2
5	T5	-5	-4.9	-1.8	12.21	15.2	18.6	11.8	14.8	18.3
6	T6	-	-	-	-	-	-	-	-	-
7	T7	33.4	33.6	34.2	31.8	32.2	32.7	32.1	33.1	31.2
8	T8	31.8	31.9	32.4	29.8	30.5	31.1	30.1	30.7	31.3
9	T9	28.2	22.7	20.2	16.5	14.1	12	19.6	17	13
10	T10	-	-	-	-	-	-	-	-	-
11	T11	33.2	33.6	34.1	33.4	34.2	34.6	33.2	34.1	34.8
12	T12	30.9	31.2	31.2	32.5	32.8	33.4	31.6	32.2	32.9

Microchannel condenser-

Sr.No,	Evaporator				Compressor
	Heater Input		Temperature Difference After 15 min.		Time required for 10 number of pulse
	V(Volt)	I(Amp)	T9i (°C)	T9f (°C)	
1	230	0.901	28.2	24.6	124.47
2	231	0.869	22.7	20.4	190.8
3	215	0.823	20.2	18.4	232
4	229	0.965	31.1	28.7	187
5	230	0.984	30.0	27.4	128

Round tube condenser-

Sr.No,	Evaporator				Compressor
	Heater Input		Temperature Difference After 15 min.		Time required for 10 number of pulse
	V(Volt)	I(Amp)	T9i (°C)	T9f (°C)	
1	229	0.965	16.5	14.2	138
2	230	0.984	14.1	12.2	155
3	226	0.938	12.0	9.3	97
4	225	0.935	20.2	18.1	159
5	223	0.965	18.1	16.8	257

Coil tube condenser –

Sr.No,	Evaporator				Compressor
	Heater Input		Temperature Difference After 15 min.		Time required for 10 number of pulse
	V(Volt)	I(Amp)	T9i (°C)	T9f (°C)	

1	221	0.920	19.6	17.3	138
2	223	1.001	17	14.1	97
3	224	0.818	13	10.6	143
4	229	0.965	22.3	20.1	130
5	230	0.984	20	18.3	187

VI.CALCULATION

Temperature of bath T9 = 19.4
V = 216volt
I = 0.793amp
Time for 10 pulses (tc) = 2.39 min
= 143.4 sec
Compressor power –

$$= \frac{Nc*3600}{Tc*3200}$$

$$= \frac{10*3600}{124.47*3200}$$

=90.38 watt

$$RE = \frac{mCp\ \Delta T}{Time} = \frac{7*4186*(28.2-24.6)}{15*60}$$

= 117.208 watt

$$COP= \frac{RE}{Compressor\ power}$$

$$= \frac{117.208}{90.38}$$

Cop=1.298

Carnot COP

i) Suction LP=31PSI=(31*0.0731)+1.013
SuctionLP=3.279kg per cm2
TL=2 °C
TL=275K

ii) Delivery HP=141PSI =(141*0.0731)+1.013
=10.927 kg per sq.cm
TH= 45 °C
TH=318K

$$Carnot\ COP= \frac{TL}{TH - TL}$$

$$= \frac{275}{318- 275}$$

= 6.39

$$Efficiency = \frac{Actual\ COP}{Carnot\ COP} = \frac{1.298 *100}{6.39}$$

= 20.31 %

VII.RESULT TABLE

Microchannel condenser –

Sr. no.	Evaporator	Compressor	Coefficient of Performance	Carnot COP	Efficiency
	Heat supplied to Evaporator	Compressor Work			
	Qevap (RE) Load	Wc	COP		
1	117.208	90.38	1.298	6.39	20.31
2	74.88	59.2	1.27	6.41	19.81
3	58.60	48.49	1.208	6.50	18.58
4	78.13	60.1	1.3	6.35	20.47
5	84.60	87.21	0.97	6.43	15.08

Round tube condenser-

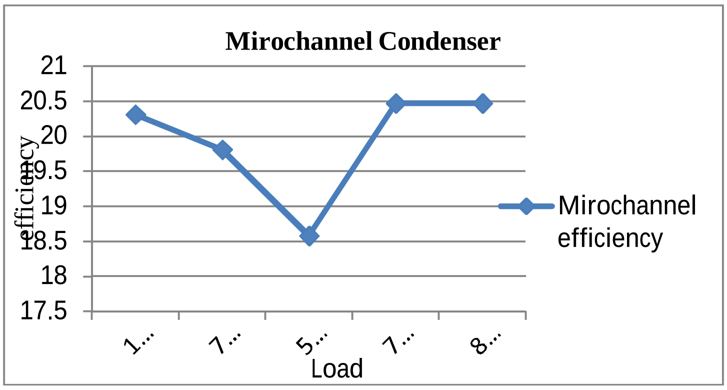
Sr. no.	Evaporator	Compressor	Coefficient of Performance	Carnot COP	Efficiency
	Heat supplied to Evaporator	Compressor Work			
	Qevap (RE) Load	Wc	COP		
1	74.88	81.5	0.92	6.571	14.15
2	58.604	72.58	0.81	6.0	13.5
3	84.906	115.97	0.76	6.39	11.89
4	68.37	70.48	0.95	6.01	15.80
5	42.32	43.62	0.97	5.70	17.01

Coil-tube condenser –

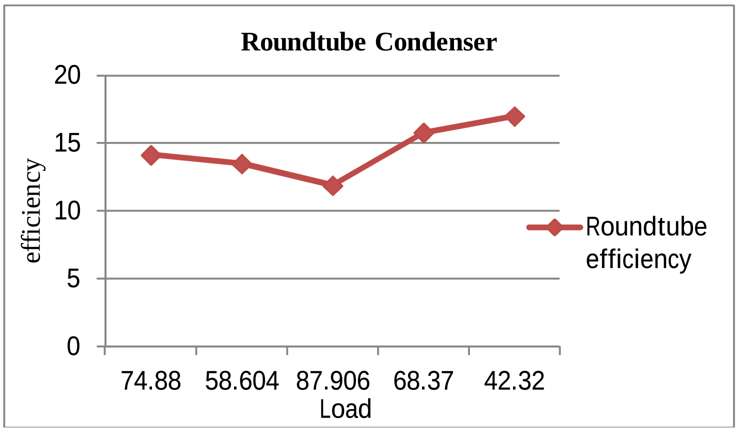
Sr. no.	Evaporator	Compressor	Coefficient of Performance	Carnot COP	Efficiency
	Heat supplied to Evaporator	Compressor Work			
	Qevap (RE) Load	Wc	COP		
1	74.88	81.52	0.92	6.5	14.15
2	94.41	115.97	0.814	6	13.5
3	68.37	78.67	0.869	6.39	11.89
4	71.62	86.28	0.83	6.3	13.17
5	55.34	60.15	0.92	5.8	45.86

VIII. Graph

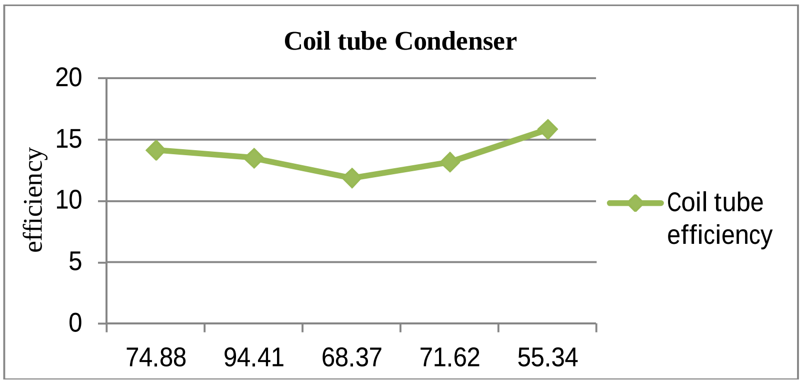
Efficiency: Micro channel Condenser-



Roundtube Condenser-



Coil tube condenser-



IX.CONCLUSION

In this study, air-conditioning systems with a micro-channel condenser and a round-tube condenser and coil tube were experimentally examined. These three condensers hadalmost identical frontal area and depth because the purpose of this study was to measure performance improvement using the micro-channel condenser which had an almost identical package volume as the round-tube condenser. An economic issue such as heat exchanger cost was not considered. Significant performance improvement was presented for the system with the micro-channel condenser. The COP of the system with the microchannel condenser is around 7-8% higher than that with the round condenser in ARI A condition. Also, using a micro-channel condenser resulted in a 2.5 ^{o}C lower condensing temperature and decreased the refrigerant pressure drop from 166 kPa in the round-tube condenser to 57 kPa in the micro-channel condenser. The refrigerant charge amount for the system with the micro-channel condenser was 9.2% smaller than that with the roundtube condenser. Even though the micro-channel condenser showed better heat transfer performance of capacity and system COP than the round-tube condenser, the round tube condenser has a cost advantage at this point in time. The micro-channel condenser was also investigated using a numerical model. The model could estimate the condenser capacity accurately. From the simulation results, it could be calculated that the consideration of non-uniform distribution for the air and refrigerant did not make a significant difference for predicting the condenser capacity. However, this small influence of non-uniform air and refrigerant on capacity is valid only for the microchannel condenser examined in this study. The air and refrigerant mal-distribution effect on heat exchanger capacity is different according to the types of heat exchanger (microchannel or round tube), the usage of heat exchanger (evaporators or condensers), and the circuit geometry of heat exchangers.

X.REFERENCES

[1] ANSI/ASHRAE Standard 37-2005, 2005. Method of Testing for Rating Electrically Driven Unitary Air- Conditioning and Heat Pump Equipment. American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA, USA.
[2] ARI Standard 210/240, 2003. Standard for Unitary Air Conditioning and Air Source Heat Pump Equipment. Air Conditioning and Refrigeration Institute, Arlington, VA, USA.
[3]. Aganda, A.A., Coney, J.E.R., Sheppard, C.G.W., 2000. Airflow maldistribution and the performance of a packaged air conditioning unit evaporator. Appl. Therm. Eng. 20, 515–528.
[4]. Beaver, A.C., Yin, J.M., Bullard, C.W., Hrnjak, P.S., 1999. An experimental Investigation of Transcritical Carbon dioxide Systems for Residential Air-Conditioning.
[5]. ACRC Report CR-18.University of Illinois at Urbana-Champaign, Urbana, IL, USA.
[6]. Brix, W., Jakobsen, A. Rasmussen, B.D., Carlsen, H., 2007.Analysis of air flow distribution in refrigeration system. In:International Congress of Refrigeration, ICR07-B2-581, Beijing.
[7]. Chang, Y.J., Wang, C.C., 1997. A generalized heat transfer correlation for louver fin geometry. Int. J. Heat Mass Transfer 40, 533–544.
[8]. Choi, J.M., Payne, W.V., Domanski, P.A., 2003. Effects of non-uniform refrigerant and air flow distributions on finned-tube evaporator performance. In:International Congress of Refrigeration, ICR0040, Washington DC.
[9]. Domanski, P.A., 1991. Simulation of an evaporator with non-uniform onedimensional air distribution. ASHRAE Trans.97, 793–802.
[10]. Elbel, S., Hrnjak, P.S., 2004. Flash gas bypass for improving the performance of transcritical R744 systems that useMicro-channel evaporators. Int. J. Refrigeration 27, 724–735.
[11]. Friedel, L., 1979. Improved friction pressure correlation for horizontal and vertical two-phase pipe flow. In: The EuropeanTwo-Phase Flow Group Meeting, paper E2, Ispra, Italy.
[12]. Fagan, T.J., 1980. The effects of air flow maldistributions on air to refrigerant heat exchanger performance. ASHRAE Trans.86, 699–713.
[13]. Hrnjak, P.S., 2002. Micro-channel Heat Exchangers as a Design Option for Charge Reduction in NH3 and HC Systems. Zero Leakage-Minimum Charge. IIR/IIF, Stockholm, p. 91. Incropera, F.P., DeWitt, D.P., 2002.
[14]. Fundamentals of Heat and Mass Transfer, fifth ed. John Wiley & Sons, New York.
[15]. Kandlikar, S.G., 2002. Fundamental issues related to flow boiling in minichannel sand microchannels. Exp. Therm. Fluid Sci. 26,389–407.
[16]. Klein, S.A., 2005. Engineering Equation Solver. Academic professional V7.457- 3D. F-Chart Software,Madison, WI, USA.