Experimental Analysis of Vapour Compression Refrigeration System with Diffuser at Condenser Inlet

R. T. Saudagar, U. S. Wankhede

Abstract- This paper discusses design and testing of diffuser at condenser inlet in vapour compression refrigeration system. Four diffusers with divergence angle 10^0, 15^0, 20^0 and 30^0 were designed for same inlet and outlet diameters. The diffusers used were with inlet diameter equal to discharge tube diameter of compressor and outlet diameter equal to condenser inlet diameter. The system was analysed using the first and second laws of thermodynamics to determine the refrigerating effect, the compressor work input, coefficient of performance (COP) and the rate of heat rejected from the system. During the test, the COPs of the system without diffuser and with optimized diffuser at condenser inlet were found out. With diffuser at condenser inlet, amount of heat rejected from condenser is also increased. To remove the same amount of heat, less heat transfer area required. This concept reduces size of condenser to achieve the same system efficiency.

Keywords: Condenser, Diffuser, Experimental analysis, Vapour compression refrigeration system.

I. INTRODUCTION

The most frequently used refrigeration cycle is the vapour compression refrigeration cycle. Ideal vapour compression refrigeration cycle results, by eliminating impracticalities associated with reversed Carnot cycle such as vaporizing the refrigerant completely before compression, replacing turbine with throttling device (expansion valve or capillary tube). Generally, domestic and industrial refrigerator, air conditioning system, heat pump and water cooler designed base on vapour compression refrigeration cycle.

Yari et al. [1] developed a new configuration of the ejector-vapour compression refrigeration cycle, which used an internal heat exchanger and intercooler to enhance the performance of the cycle. On the basis of first and second laws of thermodynamics theoretical analysis on the performance characteristics was carried out. The effects of the evapourative and condenser temperatures on the coefficient of performance, second law efficiency, exergy destruction rate and entrainment ratio were investigated. Results obtained showed that there were increase of 8.6% and 8.15% in coefficient of performance and second law efficiency values respectively of the new ejector-vapour compression refrigeration cycle compare to the conventional ejector-vapour compression refrigeration cycle with R125. It was also found that there was increase of 21% in the coefficient of performance of the new ejector-vapour compression cycle compare to the conventional vapour compression system.

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Selvaraju et al. [2] analyzed an ejector with environment friendly refrigerants. Vapour ejector refrigeration is a heat-operated system utilizing low-grade energy such as solar energy, waste heat from industrial processes, etc., and it could satisfactorily be operated at generator temperature as low as 650C. Investigations were carried out for analyzing the performance of the system and its components with few selected organic and inorganic refrigerants. Results obtained were showed that among the working fluids selected, R134a given a better performance and higher critical entrainment ratio in comparison with other refrigerants.

Bergander [3] investigated new regenerative cycle for vapour compression refrigeration which described a novel approach to the Rankine vapour compression cycle for cooling and refrigeration. Generally expansion valve, capillary tube and other throttling valves are used in vapour compression refrigeration system to lower the pressure of liquid refrigerant and low pressure refrigerant delivers to the evaporator. Specific innovation was two phase ejector applied as second step of compression, which results reduction in mechanical work required for compressor for the process of compression of gas at the expense of available kinetic energy of gas in the ejector. Injected liquid phase into accelerated flow of the vapour phase and separated working medium to high and low density phases achieved gain in efficiency. In this, compression ratio was lowered by decreasing discharge pressure from the compressor, not by increasing suction pressure. Results obtained were showed that pressure on the ejector increased by 15-16% and prototype achieved energy saving of 16%.

Akintunde [4] obtained the validation of a design model for vapour compression refrigeration system developed by Akintunde [5]. This model was used to design a vapour compression refrigeration system. The experimental set-up was made up of a compressor- reciprocating type, 0.746 kW capacity, using R134a as working fluid, with cylinder stroke volume of 32.7 cm2, evaporator and condenser, bare coil tube-in-tube serpentine copper coil. The analysis showed that the model results were comparable to the actual system from both quantitative and qualitative points of view. Under the same operational conditions, maximum absolute deviations of the variable parameters – mass flow rate, coefficient of performance and circulating water temperature were within the range of 16%.

In this work, diffuser is installed at condenser inlet. In vapour compression refrigeration system, condenser is used to remove heat from high pressure vapour refrigerant and converts it into high pressure liquid refrigerant. The refrigerant flows inside the coils of condenser and cooling fluid flows over the condenser coils. Condenser used in domestic vapour compression refrigeration system is air cooled condenser, which may be naturally or forced air cooled. Heat transfer occurs from the refrigerant to the cooling fluid. High pressure liquid refrigerant flows through an expansion device to obtain low pressure refrigerant. Low pressure refrigerant flows through the evaporator. Liquid refrigerant in the evaporator absorbs latent heat and get converted into vapour refrigerant which returns to compressor. Compressor raises the pressure and temperature of the vapour refrigerant and discharges it into the condenser to complete the cycle [6, 7]. In the present cycle, the vapour refrigerant leaves the compressor with comparatively high velocity. This high velocity refrigerant directly impinges on the tubing of condenser which may cause damage to it by vibration, pitting or erosion. It results undesirable splashing of refrigerant in the condenser coil. It also results a phenomenon called as “liquid hump”. Liquid hump refers to a rise in the level of the condensed refrigerant liquid in the central portion of the condenser as compared to the level at the ends of the condenser. It reduces the effective heat transfer surface area which can reduce condenser efficiency.

Diffuser is the static device. It raises the pressure of flowing fluid by converting its kinetic energy. In vapour compression refrigeration system, to avoid the problems of high velocity refrigerant, one of the ways is to use diffuser at condenser inlet. It smoothly decelerates the incoming refrigerant flow achieving minimum stagnation pressure losses and maximizes static pressure recovery [8]. Due to pressure recovery, at same refrigerating effect, compressor has to do less work. Hence, power consumption of the compressor will be reduced which results improvement in system efficiency. As the refrigerant flow passes through the diffuser, pressure as well as temperature will be increased. In air cooled condenser, for constant air temperature, temperature difference between hot and cold fluid will be increased. Amount of heat rejected from condenser will be rise. To remove the same amount of heat, less heat transfer area will be required. Using the diffuser at condenser inlet will provide an opportunity to use a smaller condenser to achieve the same system efficiency. Use of diffuser will also provide an advantage of reducing the effect of starvation in vapour compression refrigeration systems [9].

The cross-sectional area of diffuser should reduce in the flow direction for supersonic flows and should increase for subsonic flows [8]. The velocity of refrigerant leaving the compressor is sub-sonic. Hence, cross-sectional area of diffuser should be increasing.

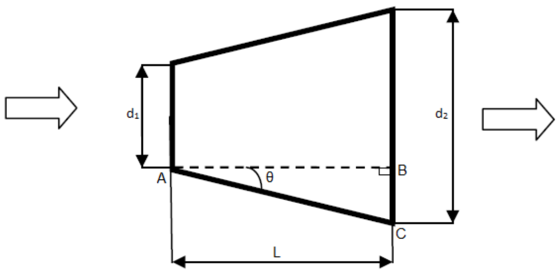

Figure1: Geometry of diffuser

Diffuser’s inlet and outlet diameters were designed. To design length of diffuser equation 1 is developed from Figure1.

$$L = AB = \frac{(d_2 - d_1)/2}{\tan\theta} \quad \dots\dots\dots (1)$$

Relation between length and divergence angle of diffuser plotted as shown in Figure 2. With increase in divergence angle of diffuser, its length is reduced for same inlet and outlet diameters.

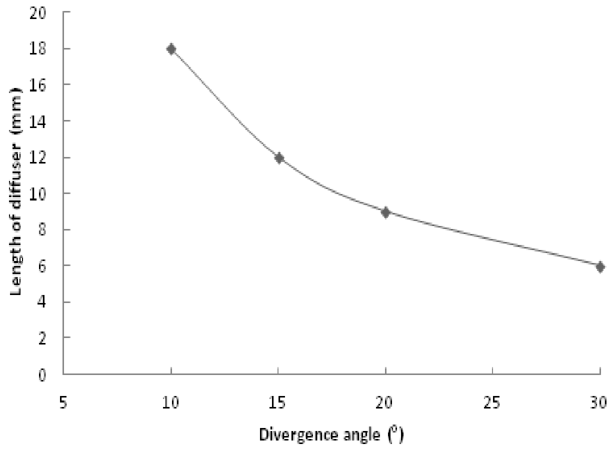

Figure 2 : Relation between length and divergence angle of diffuser

II. EXPERIMENTAL SET-UP

The schematic diagram of the vapour compression refrigeration system with diffuser at condenser inlet shown in Figure 3. Pressure gauges at the inlet and outlet of the compressor and after diffuser at condenser inlet were used to record the pressures. A calibrated refrigerant flow meter (Rotameter) was used to indicate refrigerant mass flow rate. A vapour compression refrigeration system instrumented with four temperature sensors to record temperatures. Flow control valves were used to control refrigerant flow through different diffusers selected during experimentation work. Variable speed drives to compressor and condenser fan were used to maintain the constant refrigerating effect throughout experimentation.

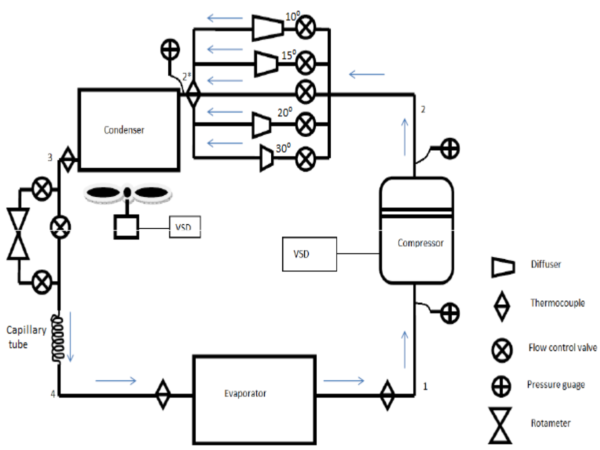

Figure 3: Schematic of vapour compression refrigeration system with diffuser at condenser inlet

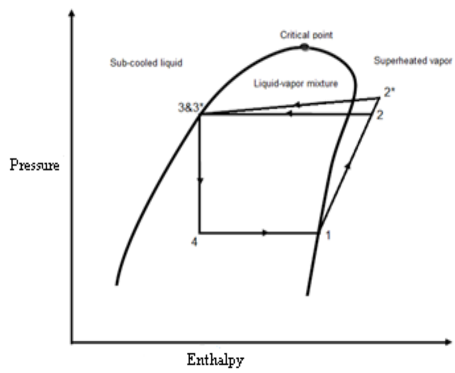

Figure 4: Pressure – Enthalpy chart

In Figure 3 and Figure 4, points 1, 2, 3, 4, 2*, and 3* represent states of refrigerant at compressor inlet, compressor outlet, condenser outlet, evaporator inlet, diffuser inlet and diffuser outlet respectively. Cycle 1-2-3-4-1 is for simple vapour compression refrigeration system and cycle 1-2*-3*-4-1 is for vapour compression refrigeration system with diffuser at condenser inlet.

III. RESULTS AND DISCUSSIONS

Table 1 summarizes temperature and pressure of refrigerant measured at various state points shown in Figures 3 and 4. Table 2 summarizes refrigerating effect, compressor power input, COP and heat rejection rate for without and with diffuser condition.

Figure 5 shows the variation of discharge pressure with time. As shown in the Figure 5, the maximum short-time discharge pressure within the first 20 min of starting the compressor runs up to 18.95 bar, after which the pressure reduced and stabilized. Figure 6 shows the relation between the mass flow rate of the refrigerant and the pressure of refrigerant at compressor outlet. As the graph shows, the pressure of refrigerant at compressor outlet increased so the mass flow rate of the refrigerant was also vary [10].

Figure 7 shows that at constant pressure, as the divergence angle increased gain in pressure increased till divergence angle 15°, because as the divergence angle increases separation of flow increases which significantly raise the pressure. After 15° of divergence angle it decreased, because there was decrease in length with increase in divergence angle. Due to short length, time for separation of flow reduces. This affects pressure gain in diffuser. Hence, maximum gain in pressure was obtained for diffuser with divergence angle 15^0 . Gain in pressure was also depends on discharge pressure. Because, as the discharge pressure increases mass flow rate increases which results increase in kinetic energy of the refrigerant. Due to this, maximum kinetic energy is available at diffuser inlet for the conversion to pressure energy. Hence, maximum gain in pressure was obtained at 18.95 bar for all diffusers.

As the refrigerant, flow through diffuser, kinetic energy is converted to pressure energy as well as temperature also increases. Because, a process in diffuser does not get enough time to dissipate heat to surrounding. From the Figure 8, it was observed that temperature gain was proportional to pressure gain. Maximum gain in temperature was obtained for diffuser with divergence angle 15° at 18.95 bar discharge pressure.

From Figure 9 and 10, it was observed that maximum percentage reduction in compressor work and maximum gain COP were also for diffuser with divergence angle 15° at discharge pressure 18.95 bar. Applying first law of thermodynamics to diffuser, it was observed that increase in enthalpy proportional to kinetic energy of the refrigerant. Rise in enthalpy is without consumption of power from universe. Hence, net compressor work was reduced. For constant refrigerating effect, COP of system was increased.

Table 1: Temperature and pressure of refrigerant at various state points

State points	Position	Temperature (°C)					Pressure (bar)				
		Without diffuser	With diffuser				Without diffuser	With diffuser			
			10^0	15^0	20^0	30^0		10^0	15^0	20^0	30^0
1	Compressor inlet	10.2	10.2	10.2	10.2	10.2	3.79	3.79	3.79	3.79	3.79
2	Condenser inlet	66.5	67.6	68.1	67.4	67.2	18.95	19.64	19.91	19.36	19.22
3	Condenser outlet	60.1	60.1	60.1	60.1	60.1	18.95	19.64	19.91	19.36	19.22
4	Evaporator inlet	8.2	8.2	8.2	8.2	8.2	3.79	3.79	3.79	3.79	3.79

Table 2: Refrigerating effect, compressor power input, COP and heat rejection rate for without and with diffuser condition

Parameters →		Refrigerating effect (W)	Rate of heat rejection from condenser (W)	COP	Compressor power input (W)
Without diffuser		320.468	442.416	2.62791	121.948
With diffuser at condenser inlet	10^0	320.468	453.76	2.8974359	121.948
	15^0	320.468	459.432	3.0540541	121.948
	20^0	320.468	448.088	2.7560976	121.948
	30^0	320.468	445.252	2.6904762	121.948

During a process through diffuser, temperature of refrigerant increases. Due to this, difference between temperature of refrigerant flows through condenser tubes and that of outside air flowing over condenser tubes increases. Hence, heat transferred rate from condenser increases. Figure 11 shows variation of rate of heat transferred through condenser with divergence angle of diffuser at varying discharge pressure. It was obtained maximum at discharge pressure 18.95 bar for diffuser with divergence angle 15^0.

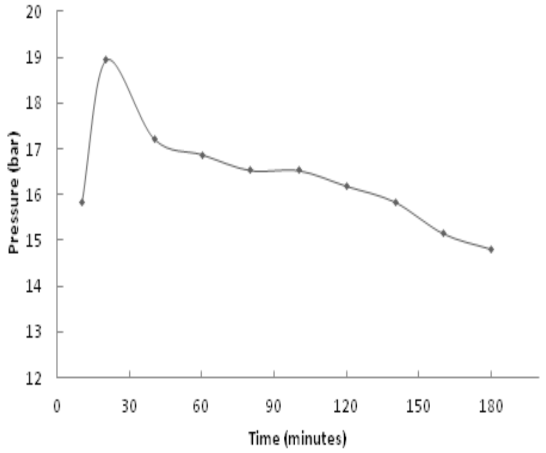

Figure 5: Variation of discharge pressure with time

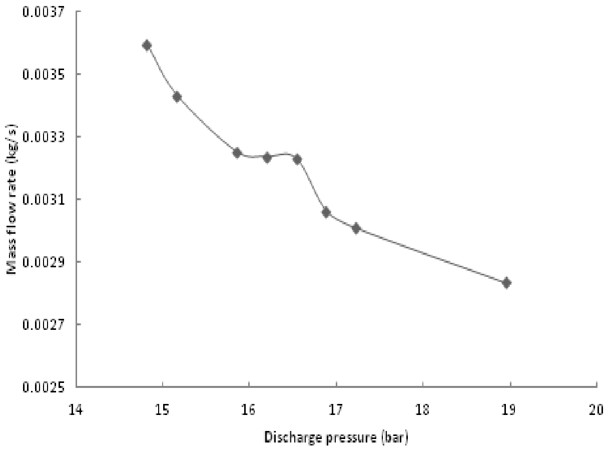

Figure 6: Variation of mass flow rate with discharge pressure

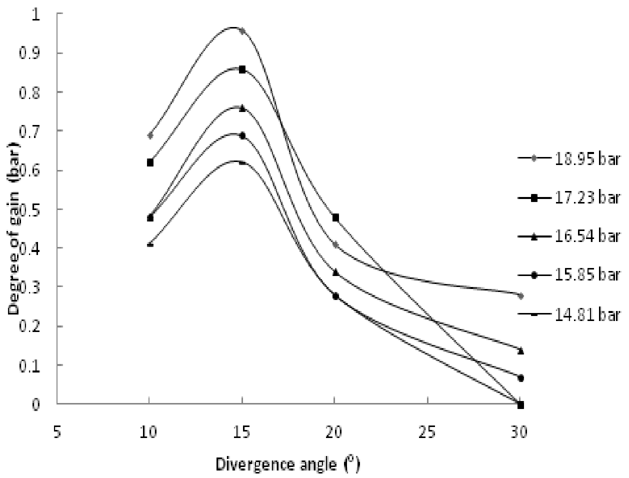

Figure 7: Variation of pressure gain with divergence angle at different discharge pressure

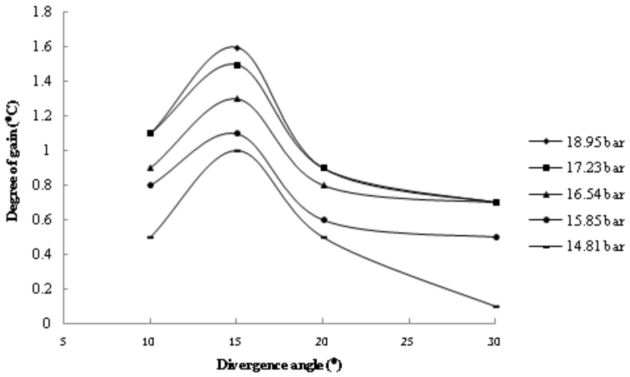

Figure 8: Variation of temperature gain with divergence angle at different discharge pressure

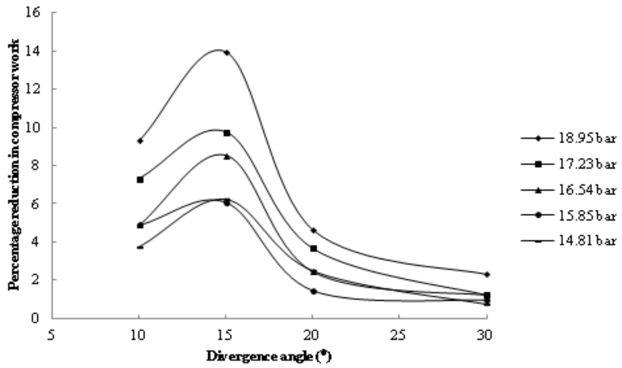

Figure 9: Variation of percentage reduction in compressor work with divergence angle at different discharge pressure

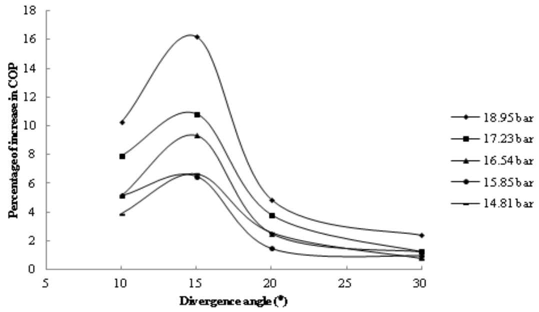

Figure 10: Variation of percentage increase in COP with divergence angle at different discharge pressure

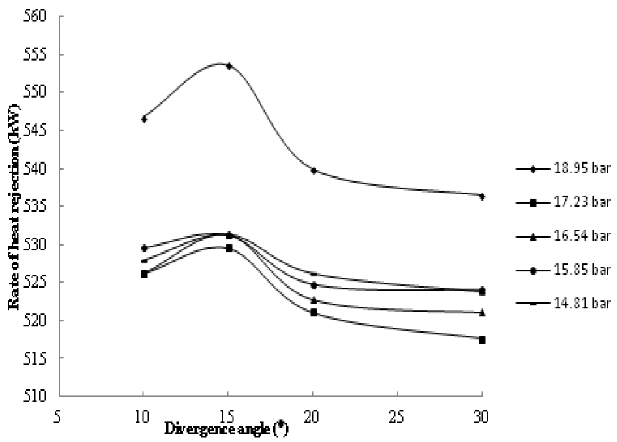

Figure 11: Variation of rate of heat rejection with divergence angle at different discharge pressure

IV. CONCLUSION

Experimental analysis has been carried out to study the effect of diffuser at condenser inlet on vapour compression refrigeration system. Four diffusers were tested with divergence angles of 10^0 , 15^0 , 20^0 and 30^0 . At particular discharge pressure, diffuser with divergence angle 15^0 gave the better results as compared to other diffusers at same discharge pressure for the same inlet and outlet diameters of diffuser. Diffuser at condenser inlet resulted gain in pressure. The discharge pressure increased by nearly 5%., compressor work is reduced by nearly 14%. Percentage of increase in COP was approximately 16% and rate of heat rejection increased by nearly 4%. By incorporating the diffuser at condenser inlet, size of condenser can be reduced.

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