

Economic Water Rate and Optimum Performance of Two Stage Azeotropic Refrigerating System

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Abstract—With a view to conserve energy, the use of azeotropes in a multistage refrigerating system is quite timely. Depending upon the requirement, such a system incorporates conventionally either a water cooled or air cooled condenser. The total operating cost of a refrigerating system with a water cooled condenser comprises the cost of water and the cost of electricity needed to drive the compressor(s). There is enough potential for research in finding out the ways to achieve maximum coefficient of performance and the least operating cost simultaneously for multi-stage azeotropic system. However, to avoid overloading of sewage facilities and to comply with municipal codes for the use of water, the water flow rate required in refrigerating system should be minimized. In the present investigation, economic water rates for two stage refrigerating systems, operating on most commonly used azeotropes R-500 and R-502, have been searched out over a wide range of operating limits. Such economic rates, if followed, would produce maximum COP and consume minimum power. The effects of controlling variables, e.g. approach, cost ratio etc have also been studied on the heat transfer to condenser, optimum condensing temperature and economic water rate. The results have been presented in tabular form only.

Index Terms— Azeotropes, multistage refrigerating, compressors, water cooled condenser.

1. INTRODUCTION

THE total cost of a multistage refrigerating system with water cooled condenser comprises the cost of power required to run the compressors and the cost of water needed in the condensing unit to make the system work in a closed and continuous cycle. To minimize the total operating cost, the power consumed by compressors may be reduced by properly selecting their compression ratio, while cost on water can be diminished by consuming cooling water as low as possible. The compression ratio of a compressor gets affected, if either the condensing pressure or inter-stage pressure is changed. The quality of cooling water gets controlled by the quantity of heat to be transferred to the condenser. It is evident that larger is the water flow rate through a condenser, the lower will be the condensing temperature resulting in lower compression ratio for high pressure (HP) compressor and hence lower cost for compressor power, but more cost on water. To the contrary, if the lower water is used, condensing temperature would be higher and thereby expenditure on water decreases while that on compressor power increases. Hence it calls for a compromise between condensing temperature and cooling water rate to achieve minimum total operating cost. The cooling water rate that minimizes the total operating cost is usually termed as Economical Cooling Water Rate.

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2. REVIEW OF PREVIOUS WORK

To cope with the existing energy shortages and the need to conserve the expended energy to the maximum possible extent, attempts have been made by Macharnen and Chapman (4) and Downing (5) on various refrigerants and their mixtures. Among the mixtures of refrigerants, R-500 and R-502 have become very common. These are known as azeotropes. An azeotrope, by definition, is the mixture of refrigerants that does not separate in to their original components with pressure/temperature changes. It has fixed thermodynamic properties unlike those of their components.

Azeotrope R-500 consists of 73.8% R-12 and 26.2% R-152. Its normal boiling point is about 3.5°C lower than that of R-12. It produces refrigerating effect per unit of swept volume about 18% more than that of R-12. A Freon-12 system designed for 60 cycle current can be shifted to 50 cycle current by using azeotrope R-500. It would result in approximately the same refrigerating capacity and evaporator and condenser conditions.

Azeotrope R-502 is a mixture of 48.8% refrigerant R22 and 51.2% refrigerant R-115. It boils at a temperature of about 4.8°C lower than that of R-22. Significantly lower discharge temperatures and lower winding temperatures are realized because of the higher capacities and lower values of compression ratio associated with R-502. Further R-502 decreases the swelling or softening effect on the common electrical insulating materials caused by

the presence of R-115. The inter-stage pressure for the two stage refrigerating system is conventionally selected as the geometric mean of operating pressure limits to minimize the total compression work. But it has been

established in (6) that if power input to the system is to be minimized, the inter-stage pressure should be optimized with coefficient of performance (COP) as the objective function.

2. THERMO DYNAMIC CONCEPT

In general, one may write the heat rejected to condenser for a refrigerating system as :

$$Q_h = P(1 + \text{COP}) \quad \text{-----1.1}$$

But Q_h per unit of cooling is expressed by :

$$Q_h / Q_c = P(1 + \text{COP}) / Q_c = (1 + 1/\text{COP}) \quad \text{-----1.2}$$

Further, for a two stage refrigerating system, COP becomes maximum if inter-stage pressure is optimized for minimum power input. Equation 1.2 may be written as :

$$Q_h / Q_c = (1 + 1/\text{COP}_o)$$

As $\text{COP}_o > \text{COP}$, We get

$$Q_h / Q_c = (1 + 1/\text{COP}) \quad Q_{h_o} < Q_h \quad \text{from equations 1.2 and 1.3} \quad \text{-----1.3}$$

It means that heat rejection to condenser would be minimum and hence minimum quantity of cooling water would be required for given condenser with a two stage system operates with optimum inter-stage pressure/temperature as decided on the basis of minimum power input.

Thus the problem of finding out economical cooling water rate for a two stage refrigerating system is coupled optimization problem, that is, first the system needs to be optimized for its minimum power consumption, and then optimum condensing temperature is to be searched out to minimize the total operating cost. On the other hand, if the system employs air cooled condenser, the total operating cost of the system would be that of power cost only. It would turn out to be minimum, if the COP of the system is maximum.

3. PRESENT WORK

In the present investigation, azeotropes R-500 and R-502 have been selected as the working fluids for two stage refrigerating system. Economic water rates that minimize the total operating cost and maximum COP are searched out over a wide range of operating temperature limits. Systems are also optimized for maximum COP, if they incorporate air cooled condensers in place of water cooled condensers. Optimum design quantities of interest are presented in the form of tables. Effects of operating variables on the design quantities are also displayed through tables.

4. SYSTEM ANALYSIS

5.1 System Employing Water Cooled Condenser (case-1)

Figure -1(a) shows the schematic of idealized two stage refrigeration system. The various heat and work quantities and pressure levels are indicated in the figure. The following simplifying assumptions are made for this system analysis:

- The thermodynamic cycle of the system is a standard one comprising isentropic compression, isentropic expansion and absence of superheating of the suction vapour and sub cooling of the high pressure (HP) condensate.
- The pressure drop in evaporator, compressor valves, condenser piping etc are neglected.
- Entire condensation of HP gas inside the condenser takes place at a fixed temperature (T_h).

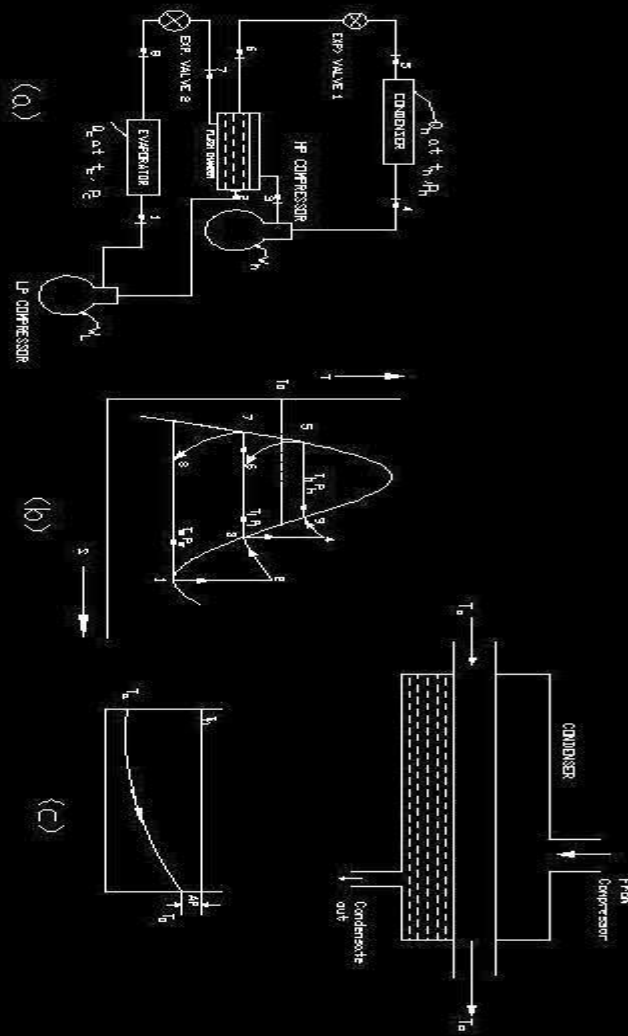


Fig. (a) Two stage Azeotropic Refrigerator System
 (b) Temperature Entropy ($T-s$) diagram
 (c) Condensation in the Condenser

Referring to fig-1(b), one may write refrigerant mass flow through LP compressor on per ton-hour basis as :

$$M_1 = 12,600 / (h_1 - h_5) \quad \text{-----5.1}$$

By energy balance on the flash chamber, refrigerant mass flow through HP compressor turns out to be :

$$M_3 = M_1 (h_2 - h_1) / (h_3 - h_6) \\ = 12,600 (h_2 - h_1) / (h_1 - h_5) (h_3 - h_6) \quad \text{-----5.2a}$$

If we consider that refrigerant mass flow through LP compressor is unit kg, then mass flow through HP compressor based on similar lines, would be :

$$M_3 = (h_2 - h_1) / (h_3 - h_6) \quad \text{-----5.2b}$$

and, total compression work shall be :

$$W_T = (h_2 - h_1) + M_3 (h_4 - h_3) \quad \text{-----5.3a}$$

However, on the basis of per ton hour, the total compression work may be written as :

$$W = W_1 + W_2 \\ = M_1 (h_2 - h_1) + M_1 (h_4 - h_3) \quad \text{-----5.3b}$$

Power consumption of the system :

$$P = W / (3600) \quad \text{-----5.4}$$

For preliminary design purposes, the enthalpies per unit mass of superheated vapour at points 2 and 4 can be approximately related to the enthalpies per unit mass of the saturated vapours at points 3 and 4, respectively as:

$$h_2 = h_3 + T_i (s_1 - s_3) \quad \text{-----5.5}$$

$$h_4 = h_9 + T_h (s_3 - s_9) \quad \text{-----5.6}$$

saturated properties of both the azeotropes are estimated from the correlations available in reference (10)

The coefficient of performance of system shall be :

$$COP = Q_c / W = 12,600 / W \quad \text{-----5.7}$$

Economic water rate expression as developed in ref (7,8) per unit ton of refrigeration, when total operating costs are minimized, is given by :

$$M_{we} = 15.45 (Q_h \Delta P C / C_w)^{0.5} \quad \text{-----5.8}$$

When Q_h is heat rejection to condenser per ton per hour and is given as :

$$Q_h = \dot{M}_3 (h_4 - h_5) \quad \text{-----5.9}$$

ΔP is the increase in power 'P' per degree rise in condensing temperature.

Optimum condensing temperature is expressed as :

$$T_{ho} = T_{wi} + DT_o \quad \text{-----5.10}$$

$$\text{Where } DT_o = 0.1545 (Q_h / P \cdot C_w C)^{0.5} + AP \quad \text{-----5.11}$$

With AP as the approach, Fig 1(c) representing the end temperature difference.

It is evident that the expressions given in equation 5.4 and 5.7 to 5.9 can be expressed in temperature alone. An explicit expression has not been attempted as it becomes extremely involved. Moreover, it serves no useful purpose because we can directly feed the above expressions in computer program to evaluate the objective function. The governing performance quantities in terms of operating parameters/variables can be expressed as:

$$P = P (T_i, T_h, T_e, T_{wi}, AP, Q_c) \quad \text{-----5.12}$$

$$\Delta P = P (T_h + 1, T_i, T_e, T_{wi}, AP, Q_c) - P (T_h, T_i, T_e, T_{wi}, AP, Q_c) \quad \text{-----5.13}$$

$$Q_h = Q_h (T_h, T_i, T_e, T_{wi}, AP, Q_c) \quad \text{-----5.14}$$

$$\dot{M}_{we} = \dot{M}_{we} (T_h, T_i, T_e, T_{wi}, AP, Q_c, C, C_w) \quad \text{-----5.15}$$

$$COP = (T_h, T_i, T_e, T_{wi}, AP, Q_c) \quad \text{-----5.16}$$

5.2 System Employing Air Cooled Condenser (Case-2)

All the assumptions as mentioned above are also valid for this case. Since System is not incorporating water cooled condenser, only power consumed by compressors needs to be minimized to minimize the total operating cost. It indirectly suggests that COP should be maximized. In equation 5.7, compression work (W) has been substituted from equation 5.3a. It has been established in ref.6 that COP is strongly dependent upon the inter-stage pressure/temperature besides other factors. Thus in order to achieve maximum COP, inter-stage pressure/temperature should be optimized. Equations 5.2 to 5.7 have been used for this.

5.3 Objective Function and Optimization

For case 1, the objective function is the total operating cost together with the COP as given by equations (5.15) and (5.16) above. The total operating costs are to be minimized producing maximum COP as well. Since total operating costs have been minimized while deriving expressions for economic water rate (\dot{m}_{we}) from equations 5.15 and 5.16, it is clear that \dot{m}_{we} and COP depend upon inter-stage and condensing temperatures (T_i and T_h) if other parameters are kept fixed. It leads to a two dimensional maximization problem with the two decision variables (T_i and T_h) subject to the constraints:

$$T_e < T_i < T_h \quad \text{-----5.17}$$

And

$$T_h > T_{wi} + \Delta P \quad \text{-----5.18}$$

For case -2 the objective function is the coefficient of performance as given by (5.7) but in this equation (W) has been calculated from equation (5.3a). There is only one decision variable (Inter-stage pressure/temperature) and hence would be a one dimensional maximization problem subjected to the constraint as given by equation (5.17) only.

5.4 Solution Technique

To find T_{ho} , where total operating cost is minimum together with the optimum system performance, initially some convenient $T_h > T_a$ was assumed. With the help of this T_h and given values of evaporator temperature (T_e) subroutine maximises the COP and transfers required optimum quantities (\dot{m}_3 and P) to the main program. Now T_h is increased by unit degree and the above process is repeated. P is determined. Q_h is found from equation 5.9 in the main program. Thereafter, DT_o is estimated from equation 5.11 to determine T_{ho} from equation 5.10. With this new value of T_{ho} , the above computations are repeated till two successive values of T_{ho} differ by + 0.1%. Condensing temperature, thus predicted, is the required optimum condensing temperature (T_{ho}) because it produces minimum operating cost for maximum COP. Finally, at T_{ho} , \dot{m}_{we} is determined from equation 5.8, Fig 1.2(a) and 1.2(b).

For case-2, only subroutine OPTIMUM is used. Different values of condensing and evaporating temperatures are fed to the sub-routine and inter-stage temperatures are optimized for maximum COP's. Later at these optimal points, main design quantities are computed and are termed as the optimum design quantities as they yield maximum COP of the system or minimum operating cost. Method of successive iterations has been used to optimize the systems for both the cases. During computation, the necessary properties of the liquid and gaseous azeotropes have been found from the co-relations given in ref (9).

Different ranges of various operating parameters considered for the azeotropic systems are based upon practical considerations and their commonly adopted values. They are as follows:

Evaporator temperature	: 5 to -50°C
Ambient/Cooling Water Temperature	: 15 to 75°C
Approach Temperature	: 2 to 5°C
Cost ratio	: 0.5 to 10

6. RESULTS AND DISCUSSIONS

6.1 SYSTEM WITH WATER COOLED CONDENSER:

Besides the direct use of Tables 6-1 to 6-3 for preliminary optimum design of the systems, they also exhibit the quantitative effects of operating variables on the design quantities for a specified set of operating parameters. Not only this, the feasible operating conditions can also be achieved with the help of the figures achieved. The graphical presentation of thermodynamic performance, display of main optimum design quantities for R-500 and R-502 respectively when the systems have been optimized for COP's only over a quite wide range of operating temperature limits is under the scope of future work..

The approach (AP) has been kept at 3%. For a fixed set of R_c , AP, t_a and t_e values, tho for R-500 is found to be slightly higher than that of R-502. On the other hand, economic water rate and heat rejection to condenser Q_{ho} are seen to be higher in case of R-502 for given R_c (except equal to 10) ,AP, t_a and t_e refer Tables 6.1 to 6.3. the detailed graphical presentation is under the scope of future research.

6.2 SYSTEM WITH AIR COOLED CONDENSER

Tables 6-4 to 6-5 can be directly used for preliminary design for preliminary optimum design of the systems operating on R-500 and R-502 using air Cooled Condenser. They also exhibit the quantitative effects of operating variables on the design quantities for a specified set of operating parameters. Not only this, the feasible operating conditions can also be achieved with the help of the figures achieved. The graphical presentation of thermodynamic performance, display of main optimum design quantities for R-500 and R-502 respectively when the systems have been optimized for COP's only over a quite wide range of operating temperature limits, is under the scope of future research.

COP's of R-500 system is observed to be higher than that of R-502 systems (Ref. Tables 6.4 and 6.5)

TABLE 6-1: THE OPTIMUM DESIGN QUANTITIES FOR TWO-STAGE AZEOTROPIC REFRIGERATING SYSTEM INCORPORATING WATER COOLED CONDENSER

Design Parameters : $R_c = 3.0$ $AP = 3^\circ\text{C}$

Azeotrope	R-502					R-500				
	t_e (°C)	t_{io} (°C)	t_{ho} (°C)	\dot{M}_{we} (Kg/ton-h)	\dot{Q}_{ho} (KJ/ton-h)	COP_o	t_{io} (°C)	t_{ho} (°C)	\dot{M}_{we} (Kg/ton-h)	\dot{Q}_{ho} (Kg/ton-h)
-50	-7.66	30.34	595.01	18,289.2	2.21	-12.25	31.13	527.79	18096.5	2.29
-40	-3.01	30.53	547.61	17,280.1	2.69	-3.38	30.95	512.50	17067.8	2.82
-30	1.73	30.73	506.17	16,392.8	3.32	1.46	31.11	477.37	16215.0	3.49
-20	6.55	30.93	469.5	15,602.9	4.20	6.27	31.27	446.45	15462.3	4.40
-50	4.12	49.72	725.90	20,435.5	1.61	2.91	49.94	680.87	19808.5	1.75
-40	8.55	49.92	662.23	19,193.4	1.91	7.43	50.8	628.80	18663.4	2.08
-30	13.06	50.12	606.87	18,106.1	2.29	12.28	50.22	583.64	17663.9	2.49
-20	17.56	50.33	558.34	17,143.3	2.77	16.95	50.36	544.00	16784.4	3.01
-50	16.17	69.13	911.86	23,414.2	1.17	15.10	69.32	831.87	22028.1	1.34
-40	20.27	69.32	824.51	21,817.0	1.37	19.69	69.47	761.94	20659.7	1.56
-30	24.45	69.51	749.28	20,429.9	1.614	24.27	69.63	700.83	19458.9	1.84
-20	28.71	69.71	683.87	19,210.3	1.91	28.81	69.79	647.39	18404.9	2.17

TABLE 6-2: THE OPTIMUM DESIGN QUANTITIES FOR TWO-STAGE AZEOTROPIC REFRIGERATING SYSTEM INCORPORATING WATER COOLED CONDENSER

Design Parameters : $R_c = 3.0$ $t_a = 30^\circ\text{C}$

Azeotrope	R-502					R-500				
t_e ($^\circ\text{C}$)	t_{io} ($^\circ\text{C}$)	t_{ho} ($^\circ\text{C}$)	\dot{M}_{we} (Kg/ton-h)	\dot{Q}_{ho} (KJ/ton-h)	COP_o	t_{io} ($^\circ\text{C}$)	t_{ho} ($^\circ\text{C}$)	\dot{M}_{we} (Kg/ton-h)	\dot{Q}_{ho} Kg/ton-h)	COP_o
-50	-2.44	39.10	645.29	19,184.6	1.91	2.57	40.29	527.79	18096.5	2.01
-40	2.10	39.30	591.51	18,081.3	2.30	2.02	40.57	512.50	17067.8	2.42
-30	6.72	39.50	544.63	17,113.0	2.79	6.83	40.74	477.37	16215.0	2.94
-20	11.43	39.71	503.26	16,252.9	3.45	11.63	40.91	446.45	15462.3	3.63
-50	-1.32	41.03	658.27	19,397.3	1.85	-1.82	41.90	680.87	19808.5	1.97
-40	3.34	41.23	602.83	18,271.2	2.22	2.86	42.09	628.80	18663.4	2.37
-30	7.87	41.44	554.58	17,283.0	2.69	2.69	42.23	583.64	17663.9	2.88
-20	12.50	41.65	512.05	16,405.7	3.31	3.31	42.39	544.00	16784.4	3.54
-50	-0.71	42.0	665.00	19,505.8	1.82	-1.41	42.72	831.87	22028.1	1.95
-40	3.86	42.20	608.76	18,367.9	2.18	3.30	42.89	761.94	20659.7	2.34
-30	8.50	42.41	559.74	17,359.5	2.64	8.14	43.03	700.83	19458.9	2.84
-20	13.03	42.62	516.64	16,483.5	3.24	12.85	42.7	647.39	18404.9	3.49

TABLE 6-3: THE OPTIMUM DESIGN QUANTITIES FOR TWO-STAGE AZEOTROPIC REFRIGERATING SYSTEM INCORPORATING WATER COOLED CONDENSER

Design Parameters : $AP= 3.0(^{\circ}C)$ $t_a = 30^{\circ}C$

Azeotrope	R-502					R-500				
	t_e ($^{\circ}C$)	t_{io} ($^{\circ}C$)	t_{ho} ($^{\circ}C$)	\dot{M}_{we} (Kg/ton-h)	\dot{Q}_{ho} (KJ/ton-h)	COP_o	t_{io} ($^{\circ}C$)	t_{ho} ($^{\circ}C$)	\dot{M}_{we} (Kg/ton-h)	\dot{Q}_{ho} (Kg/ton-h)
-50	3.92	49.48	295.59	20,407.2	1.61	2.90	50.013	277.90	19,806.0	1.75
-40	8.57	49.94	270.49	19,198.4	1.91	7.54	50.34	257.40	18,677.5	2.07
-30	13.2	50.41	248.71	18,138.0	2.28	12.43	50.67	239.05	17,698.2	2.47
-20	17.82	50.89	229.41	17,196.7	2.74	17.29	51.01	223.10	16,832.8	2.98
-50	-2.78	38.51	828.24	19,121.7	1.93	-2.96	39.62	679.22	18,832.1	2.02
-40	1.73	38.67	759.07	18,020.7	2.32	1.62	39.87	617.21	17,765.7	2.44
-30	6.42	38.83	698.75	17,054.8	2.83	6.42	40.02	572.77	16,843.3	2.97
-20	11.00	38.99	645.65	16,196.9	3.50	11.2	40.17	534.11	16032.8	3.67
-50	-3.75	36.92	1153.26	18,953.4	1.98	-4.83	36.49	1271.04	18,564.3	2.11
-40	0.82	37.03	1057.20	17,865.8	2.39	0.11	36.82	1098.81	17,575.4	2.53
-30	5.38	37.15	973.64	16,912.1	2.92	4.54	37.39	904.37	16,616.3	3.14
-20	10.02	37.26	899.93	16,065.0	3.64	9.41	36.79	998.34	15,847.3	3.88

TABLE 6-4 OPTIMUM DESIGN QUANTITIES FOR TWO-STAGE R-500 AZEOTROPIC REFRIGERATING SYSTEM INCORPORATING AIR COOLED CONDENSER

t_h (°C)	t_e (°C)	t_{io} (°C)	r_1	r_2	M_3 (Kg/Kg)	V_1 (m ³ /kg)	V_3 (m ³ /kg)	W_T (KJ/Kg)	COP_o
15	-20	-2.22	1.902	1.718	1.14	0.1123	0.0695	26.14	6.71
	-30	-7.00	2.426	2.024	1.19	0.1645	0.846	35.56	4.93
	-40	-12.20	3.146	2.436	1.24	0.2484	0.1055	46.25	3.79
	-50	-16.82	4.320	2.893	1.30	0.3905	0.1297	58.23	3.01
30	-20	5.62	2.455	2.023	1.23	0.1123	0.0584	36.87	4.52
	-30	0.90	3.169	2.356	1.28	0.1645	0.0705	46.90	3.56
	-40	-3.84	4.22	2.760	1.34	0.2484	0.0858	58.23	2.86
	-50	-12.21	5.129	3.704	1.43	0.3905	0.1214	72.60	2.34
45	-20	14.00	3.172	2.292	1.33	0.1123	0.495	46.76	3.36
	-30	9.33	4.141	2.638	1.39	0.1645	0.0594	57.41	2.74
	-40	4.65	5.584	3.064	1.46	0.2484	0.0717	69.47	2.26
	-50	-0.05	7.818	3.557	1.54	0.3905	0.0873	83.07	1.89
60	-20	22.71	4.076	2.503	1.48	0.1123	0.0429	58.99	2.49
	-30	18.11	5.376	2.851	1.55	0.1645	0.0512	70.50	2.09
	-40	13.60	7.353	3.253	1.63	0.2484	0.0613	83.50	1.76
	-50	9.06	10.451	3.733	1.72	0.3905	0.0739	93.17	1.50

TABLE 6-5 OPTIMUM DESIGN QUANTITIES FOR TWO-STAGE R-502 AZEOTROPIC REFRIGERATING SYSTEM INCORPORATING AIR COOLED CONDENSER

t_h (°C)	t_e (°C)	t_{io} (°C)	r_1	r_2	M_3 (Kg/Kg)	V_1 (m ³ /kg)	V_3 (m ³ /kg)	W_T (KJ/Kg)	COP_o
15	-20	-2.02	1.84	1.65	1.169	0.0611	0.0396	21.97	6.46
	-30	-6.80	2.31	1.92	1.221	0.0877	0.0481	29.86	4.76
	-40	-11.60	3.00	2.25	1.279	0.1302	0.0588	38.69	3.67
	-50	-16.42	4.05	2.66	1.345	0.2004	0.0726	48.73	2.92
30	-20	6.02	2.34	1.84	1.278	0.0611	0.0340	30.91	4.29
	-30	1.40	2.99	2.12	1.339	0.0877	0.0408	39.31	3.37
	-40	-3.42	3.95	2.44	1.404	0.1302	0.0494	48.74	2.72
		-7.91	5.41	2.84	1.481	0.2004	0.0604	59.49	2.23
45	-20	14.40	2.98	2.04	1.431	0.0611	0.0299	39.68	3.09
	-30	10.03	3.86	2.30	1.499	0.0877	0.0355	48.64	2.52
	-40	5.55	5.17	2.62	1.577	0.1302	0.0425	58.74	2.08
	-50	1.15	7.21	2.99	1.667	0.2004	0.0513	70.24	1.74
60	-20	23.21	3.77	2.19	1.652	0.0611	0.0267	48.76	2.29
	-30	18.91	4.94	2.45	1.736	0.0877	0.0318	58.47	2.91
	-40	14.70	6.71	2.75	1.830	0.1302	0.0379	69.34	1.61
	-50	10.56	9.52	3.09	1.938	0.2004	0.0452	81.74	1.36

7. CONCLUSIONS:

1. For a preliminary design of two stage azeotropic refrigerating system, the Tables 6-1 to 6-5 presented can directly be used.
2. Economic water rate and heat transfer to condenser turns out to be relatively lower in case of R-500 for a given set of condenser, evaporator, ambient and approach temperatures and cost ratio.
3. R-500 system produces comparatively higher COP than R-502 system for specified operating conditions.
4. The effect of approach temperature is more pronounced on the economic water rate than the other quantities. It should be selected quite carefully.
5. Though, the initial investment in case of R-500 system turns out to be more than R-502 system, it would get compensated over a small span of time because of lower operating cost of the R-500 system.

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9. NOMENCLATURE:

- AP : Approach °C
C : Cost of Electricity (Rs/kw-h)
 c_w : Cost of Water (Rs/k-litre)
COP_o: Maximum co-efficient of performance.
 h_x : Enthalpy per unit mass at state condition 'x' (kJ/kg)
 \dot{M}_1 : Refrigerant mass flow through LP compressor (kg/ton-h).
 \dot{M}_3 : Refrigerant mass flow through HP compressor (kg/ton-h)
 \dot{M}_{we} : Economic water rate (kg/ton-h)
 M_3 : Refrigerant mass flow through HP compressor per unit mass flow through Lp compressor (kg/kg).
P : Power (kw/ton)
 P_o : Minimum power (kw/ton)
 ΔP : Power increase per ton per degree rise in condensing Temperature (kw/ton-°C)

- P_h : Condensing pressure (bar)
 P_e : Evaporating pressure (bar).
 P_{io} : Optimum inter-stage pressure (bar).
 \dot{Q}_h : Heat transfer to condenser (kJ/ton-h).

 \dot{Q}_{ho} : Heat transfer to condenser pertaining to minimum operating cost and maximum COP (kJ/ton-h)
 \dot{Q}_c : cooling effect per unit mass flow through LP compressor (kJ/kg).
 r_1 : Compression ratio (P_{io}/P_e) of LP compressor.
 r_2 : Compression ratio (P_h/P_{io}) of HP compressor.
 R_c : Cost ratio (c/c_w).

 s_x : Entropy per unit mass at state condition (kJ/kg-k)
 t_a, T_a : Ambient temperature ($^{\circ}\text{C}, \text{K}$)
 t_{w1}, T_{w1} : Inlet water temperature ($^{\circ}\text{C}, \text{K}$)

 t_h, T_h : condensing temperature ($^{\circ}\text{C}, \text{K}$)
 t_e, T_e : evaporating temperature ($^{\circ}\text{C}, \text{K}$).
 t_{ho}, T_{ho} : Optimum condensing temperature($^{\circ}\text{C}, \text{K}$).
 t_i, T_i : Interstage temperatures ($^{\circ}\text{C}, \text{K}$)
 t_o, T_o : Outlet water temperature ($^{\circ}\text{C}, \text{K}$)
 t_{io}, T_{io} : Optimum interstage temperature($^{\circ}\text{C}, \text{K}$).
 V_1 : Minimum volume flow through LP compressor per unit mass flow through LP compressor (m^3/kg).
 V_3 : Minimum volume through HP compressor per unit mass flow through LP compressor (m^3/kg).
 W_1 : compression work of LP compressor (kJ/ton-h).
 W_2 : compression work of HP compressor(kJ/ton-h).
 W : Total compression work (kJ/ton-h).
 W_T : Total compression work (kJ/ton-h).