Numerical Computation of Economic Cooling Water rate for Two-stage Azeotropic Refrigerating systems

Prof. D.V. Mahindru[1] and Priyanka Mahendru[2]

[1] Shri Ramswaroop Memorial Group of Professional Colleges, Tewari Ganj, Lucknow

Received: 23 April 2011 Accepted: 13 May 2011 Published: 27 May 2011

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 refrigerating 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, Multi â??" Stage, Refrigeration, Condenser, Thermodynamic Concept.

contrary, if the lower quantity of water is used , condensing temperature would be higher and thereby expenditure on water decreases while that oncompressor 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.

1 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? 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? 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+COP)

1.1

But Q ?h per unit of cooling is expressed by :Q ?h/Q ?c=P(1+COP)/Q ?c=(1=1/COP) 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 :

As COP o > COP, We get Q ?ho <Q ?h from equations 1.2 and 1.31.3

"It means that heat rejection to condenser would be minimum and hence minimum quantity of cooling water would be required for given condenser when 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.

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. 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 a) System Employing Water Cooled Condenser

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 : i. 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. ii.

The pressure drop in evaporator, compressor valves, condenser piping etc are neglected. iii.

Entire condensation of HP gas inside the condenser takes place at a fixed temperature (T h).

Referring to fig-1(b), one may write refrigerant mass flow through LP compressor on per ton-hour basis as : M1=12,600/(h 1 -h 5)

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(h2 -h1) / (h1 -h5) (h -h5) 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) 5.2b

and, total compression work shall be :W T =(h 2 -h 1) + M 3 ((h 4 -h 3 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) 5.3b

P ower consumption of the system :

P= W/(3600) 5.4 V.

IV.

)Q ?h/ Q ?c= (1+1/COP)

?P is the increase in power 'P' per degree rise in condensing temperature. Optimum condensing temperature is expressed as : 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 at 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 be expressed as: P=P(T i,, T h, T e, T wi, AP,Q ?c) 5.12 ?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) 5.13

Q ?h=Q ?h(T h ,T i, T e, T wi, AP,Q ?c) 5.14

? we =? we (T h ,T i, T e, T wi, AP,Q ?c, c,c w) 5.15 COP=(T h ,T i, T e, T wi, AP,Q ?c)

5 b) 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 (?we).from equations 5.15 and 5.16, it is clear that ? we and COP depend upon interstage and condensing temperatures (Ti and Th) if other parameters are kept fixed. It leads to a two dimensional maximization problem with the two decision variables (Ti and Th) subject to the constraints: $Te<Ti < Th$ 5.17 And $Th>Twi +AP$

6 c) Solution Technique

To find Tho, where total operating cost is minimum together with the optimum system performance, initially some convenient $Th > Ta$ was assumed. With the help of this Th and given values of evaporator temperature(T e) subroutine maximises the COP and transfers required optimum quantities (?3 and P) to the main program. Now Th 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, DTo is estimated from equation 5.11 to determine Tho from equation 5.10. With this new value of Tho, the above computations are repeated till two successive values of Tho differ by + 0.1% . Condensing temperature , thus predicted , is the required optimum condensing temperature (Tho) because it produces minimum operating cost for maximum COP.

7 RESULTS AND DISCUSSIONS

8 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 approach (AP) has been kept at 3?. For a fixed set of Rc, AP, ta and te 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 Rc (except equal to 10) ,AP, ta and te refer Tables 6.1 to 6.3, the detailed graphical presentation is available in reference-11 (page 27 to 44). COP's of R-500 system is observed to be higher than that of R -502 systems (Ref.

9 August

) h 2 =h 3 + T i (s 1 -s 3) 5.5 h 4 =h 9 + T h (s 3 -s 9) 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 5.7 Economic water rate expression as developed in ref **??**7, **??**) per unit ton of refrigeration, when total operating costs are minimized, is given by : ? 5.8 When Q ?h is heat rejection to condenser per ton per hour and is given as : Q ?h=? (h 4 -h 5) 5.9

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: M ?we = 15.45 (Q h .?P . / W 0.5

10 REFERENCES RÉFÉRENCES REFERENCIAS

1. For a preliminary design of two stage azeotropic refrigerating system, the Tables 6-1 to 6-3 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. [1] [2]

Figure 1:

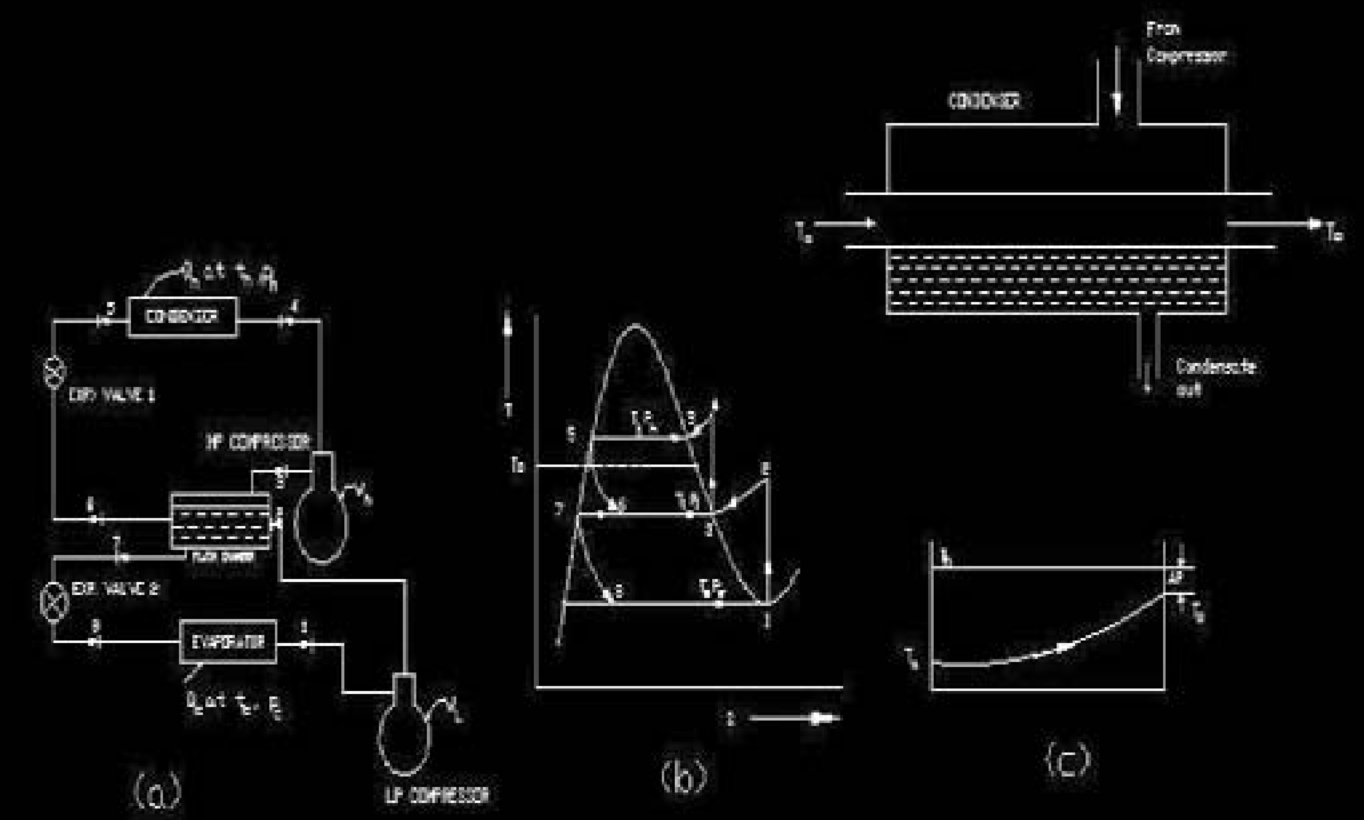

Fig.1 (a) Two stage Azeotropic Refrigerator System
(b) Temperature Entropy (T-S) diagram
(c) Condensation in the Condenser

6

o 2.29 2.82 3.49 4.40 1.75 2.08 2.49 3.01
COP
a .1 : systems Des Azeotrope- (?) - - 20 - - - 40 -30 -20 13.06 17.56 50.12 50.33 606.87 558.34 181
Ef- in- ign R- (?) 50 40 - 20 50 40
fect cor- Pa- 502 t e - - 30 6.55 4.12 8.55
of po- ram- R- (?) 7.66 3.01 1.73 30.93 49.72 49.92
am- rat- e- 500 t io 30.34 30.53 30.73 469.5 725.96 662.23
bi- ing ters (?) 595.05 547.65 506.17 1560 2094 35.95 193.4
ent wa- : tho 18,289 17280 16394.8 1.20 1.61 1.91
tem- ter R (?) 2.21 2.69 3.32 6.27 2.91 7.43
per- cooled ? - - 1.46 31.27 49.94 50.8
a- con- = we 12.25 3.38 31.11 446.45 680.87 628.80
ture denser 3.0 (Kg/ton- 31.13 30.95 477.37 17546 19.38 08.18 5663.4
on AP h) 527.79 512.50 16215.0
the = Q 18096.7 17067.8
op- 3? ?ho
ti- (KJ/ton-
mum h)
de- COP
sign o tio
quan- (?)
ti- t ho
ties (?)
for ?
two- we
stage (Kg/ton-
azeotropic h)
re- Q
frig- ?ho
er- (Kg/ton-
at- h)
ing

Figure 3: Table 6

6

				o COP		1.75	2.07	2.47	2.98	2.02	2.44	2.97	3.67	2.11	2.53
.3 : Effect of Cost Ratio on the optimum design quantities for two-stage azeotropic	refrigerating system incorporating water cooled condenser	Design Parameters : AP= 3.0(?) t a = 30?	Azeotrope R-502 R-500	R t e (?) t io (?) t ho (?) ? we Q ?ho	(Kg/ton-h)	-50 -	0.5	-30	-20	-50 -	5.0	-30	-20	-50	-40
						3.92	-40	13.2	17.82	2.78	-40	6.42	11.00	-3.75	0.82
					(KJ/ton-h)	49.48	8.57	50.41	50.89	38.51	1.73	38.83	38.99	36.92	37.03
						295.59	49.94	248.71	229.41	828.24	38.67	698.75	645.65	1153.26	1057.
					(Kg/ton-h)	20,407.2	270.49	18138.0	17,196.7	19,121.7	759.07	17054.8	16,196.9	18,953.4	17,86
						1.61	19198.4	2.28	2.74	1.93	18020.7	2.83	3.50	1.98	2.39
					(Kg/ton-h)	2.90	1.91	12.43	17.29	-	2.32	6.42	11.2	-4.83	0.11
						50.013	7.54	50.67	51.01	2.96	1.62	40.02	40.17	36.49	36.82
				COP o t io (?) t ho (?) ? we Q ?ho		277.90	50.34	239.05	223.10	39.62	39.87	572.77	534.11	1271.04	1098.
						19,806.0	257.40	17,698.2	16,832.8	679.22	617.21	16,843.3	16032.8	18,564.3	17,57
							18,677.5			18,832.1	17,765.7				

Figure 4: Table 6

10 REFERENCES RÉFÉRENCES REFERENCIAS