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Energy Recovery from the Heat of Rejection

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ENERGY RECOVERY FROM THE HEAT OF REJECTION

by

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ABSTRACT

The paper shows a method in which the heat rejected by the refrigeration system can be recovered. The application discussed here pertains to an air conditioning facility situated in humid climate and the one that requires relative humidities lower than 55%. Reheat is required in such applications for controlling the inside design conditions. The new method utilizes the heat of rejection for reheating instead of using energy from an external source. This method not only reduces the energy consumption (kWh) of the system but it also lowers the peak demand (kW) and provides for additional subcooling of the refrigerant liquid. The owning costs of the new system are compared with the annual savings in energy, on the basis of life cycle costing.

INTRODUCTION

This paper deals with energy recovery from the heat of rejection of the refrigeration plant of a facility manufacturing pharmaceutical products.

During the energy audit of the facility, energy consumption patterns of various areas were studied. Air conditioning systems were some of the highly energy intensive activities and even among these, the air conditioned areas which required relative humidities lower than 55% were the most energy intensive. High ambient dew point temperatures during the summer and the monsoon months associated with inside relative humidities lower than 55%,required the use of reheat.

ANALYSIS OF THE OLD SYSTEM

Tables - 1A and 1B show the details of cooling load analysis of the air conditioning systems for summer and monsoon. Figure - 1 explains the psychrometric process.

It is interesting to see that reheat constitutes from 7% to 80% of the total cooling load of the low relative humidity (lower than 55%) areas during the major part of the year in humid climates.

TABLE - 1A

THE ANALYSIS OF COOLING LOADS IN SUMMER (R5)

PARTICULARS OF LOADS		HARD CAPSULE PLANTS I, II, III Summer	SOFT CAPSULE PLANTS I, II, III Summer	FIRST DRYER Summer	SECOND DRYER Summer
OUTDOOR DESIGN	db (C)	43.34	43.34	43.34	43.34
	(F)	110	110	110	110
	wb (C)	25.56	25.56	25.56	25.56
	(F)	78	78	78	78
INSIDE DESIGN	db (C)	21.11	18.34	26.67	23.89
	(F)	70	65	80	75
	rh (%)	53	55	40	21
ROOM SENSIBLE HEAT	(kW)	145	23.5	4.47	5.77
	(Btu/h)	494773	80173	15247	19704
ROOM LATENT HEAT	(kW)	22.86	15.75	5.16	3.10
	(Btu/h)	78005	53746	17608	10598
ROOM TOTAL HEAT	(kW)	167.86	39.25	9.63	8.87
	(Btu/h)	572778	133919	32855	30302
RETURN AIR LOADS	(kW)	39.24	9.11	2.99	3.16
	(Btu/h)	133936	31094	10178	10792
GRAND TOTAL HEAT	(kW)	207.1	48.36	12.62	12.03
	(Btu/h)	706714	165013	43033	41094
SENSIBLE HEAT FACTOR		0.864	0.599	0.464	0.650
INDICATED ADP	(C)	9.45	NIL	NIL	NIL
	(F)	49	NIL	NIL	NIL
SELECTED ADP	(C)	9.45	4.45	4.45	-1.11
	(F)	49	40	40	30
SELECTED SENSIBLE HEAT FACTOR		0.864	0.72	0.73	0.94
REHEAT REQUIRED	(kW)	NIL	17	9.48	42.87
	(Btu/h)	NIL	58031	32360	146330
AIR QUANTITY	(l/s)	11706	2747	614	1834
	(cfm)	24800	5820	1300	3885
REFRIGERATION LOAD	(kW)	207.10	65.36	22.10	54.90
	(TR)	59.00	18.60	6.30	15.60
HEAT REJECTED	(kW)	271.40	85.56	28.98	71.76
	(TR)	77.10	24.31	8.23	20.39
POWER OF COMPRESSION	(kW)	61.95	19.50	6.615	16.38

Notes for Table - 1A

dry bulb temperature = db, wet bulb temperature = wb, relative humidity = rh, Apparatus dew point = ADP
(R5) : Based on reference 5.

TABLE - 1B

THE ANALYSIS OF COOLING LOADS IN MONSOON

PARTICULARS OF LOADS		HARD CAPSULE PLANTS I, II, III	SOFT CAPSULE PLANTS I, II, III	FIRST DRYER	SECOND DRYER
		Monsoon	Monsoon	Monsoon	Monsoon
OUTDOOR DESIGN	db [C]	31.11	31.11	31.11	31.11
	[F]	88	88	88	88
	wb [C]	26.11	26.11	26.11	26.11
	[F]	79	79	79	79
INSIDE DESIGN	db [C]	21.11	18.34	26.67	23.89
	[F]	70	65	80	75
	rh [%]	53	55	40	21
ROOM SENSIBLE HEAT	[kW]	133	21.13	3.06	3.76
	[Btu/h]	453861	72103	10450	12840
ROOM LATENT HEAT	[kW]	23.50	15.85	5.24	3.20
	[Btu/h]	80179	54096	17872	10910
ROOM TOTAL HEAT	[kW]	156.50	36.98	8.30	6.96
	[Btu/h]	534040	126199	28322	23750
RETURN AIR LOADS	[kW]	38.66	8.97	2.92	2.98
	[Btu/h]	131932	30608	9978	10161
GRAND TOTAL HEAT	[kW]	195.16	45.95	11.22	9.94
	[Btu/h]	665972	156807	38300	33911
SENSIBLE HEAT FACTOR		0.849	0.571	0.369	0.541
INDICATED ADP	[C]	9.17	NIL	NIL	NIL
	[F]	48.5	NIL	NIL	NIL
SELECTED ADP	[C]	9.45	4.45	4.45	-1.11
	[F]	49	40	40	30
SELECTED SENSIBLE HEAT FACTOR		0.864	0.72	0.73	0.94
REHEAT REQUIRED	[kW]	16.26	19.63	11.10	46.32
	[Btu/h]	55511	67000	37870	158085
AIR QUANTITY	[l/s]	12045	2764	614	1888
	[cfm]	25520	5855	1300	4000
REFRIGERATION LOAD	[kW]	211.42	65.58	22.32	56.26
	[TR]	60.00	18.70	6.40	16.00
HEAT REJECTED	[kW]	276.00	86.02	29.44	73.60
	[TR]	78.41	24.44	8.36	20.91
POWER OF COMPRESSION	[kW]	63.00	19.635	6.72	16.80

Notes for Table - 1B : dry bulb temperature = db, wet bulb temperature = wb, relative humidity = rh, Apparatus dew point = ADP

The old air conditioning facility consisted of two stage vapour compression system with reciprocating compressors using R717 (Ammonia), shell and tube flooded chillers using Propylene Glycol brine and shell and tube water cooled condensers. The system had electrically operated reheat coils installed in the supply air ducts downstream of the cooling coils/sprays.

Figure - 2 shows the system components, while Figure - 3 explains the two stage vapour compression cycle on the Pressure - Enthalpy diagram. The state points are identified by ASHRAE subscripts, which apply to both the figures.

Table - 2 presents the system parameters at the respective state points.

FEATURES OF THE NEW SYSTEM

It was observed in the facility that the demand for reheat as well as the availability of the rejected heat occur simultaneously. Hence a heat recovery system using the rejected heat was designed, incorporating the following :

1. Finned-tube reheat coils suitable for using the hot gas from the high pressure compressor.

2. Three-way diverting valves to regulate the quantity of hot gas circulated through the reheat coils.

3. Dry-bulb temperature controllers to control the operation of the diverting valves.

4. Refrigerant piping with isolating and non-return valves to circulate the hot gas from the compressor through the reheat coils and the condenser.

All other system components such as the compressors, desuperheater, brine chiller as well as the condenser were kept unaltered.

Table - 3 presents annual savings in energy.

TABLE - 2

THE SYSTEM PARAMETERS AT THE RESPECTIVE STATE POINTS. (R1)
REFRIGERATION LOAD AT 266.5 K (20F)
REFRIGERANT R717 (AMMONIA)

STATE POINT	TEMPERATURE K (F)	PRESSURE MPa (psia)	DENSITY kg/m^3 (lb/ft^3)	ENTHALPY kJ/kg (Btu/lb)	ENTROPY kJ/kg.K Btu/lb.F	QUALITY (%)
1	260 (7.9)	0.2517 (36.50)	2.20 (0.1375)	480 (206)	10.50 (2.508)	-
1.1	262 (12.0)	0.242 (35.10)	1.80 (0.112)	490 (210.30)	10.60 (2.532)	-
2.1	337 (147)	0.643 (93.22)	4.10 (0.256)	648 (278.11)	10.62 (2.537)	-
2.2	287.04 (57)	0.639 (92.7)	4.90 (0.306)	520 (223.17)	10.20 (2.436)	-
2	360 (188.33)	1.707 (247.6)	11.5 (0.718)	660 (283.30)	10.30 (2.461)	-
3	316 (109)	1.6803 (243.7)	577.0 (36.00)	-565 (-242.49)	6.4 (1.528)	-
3.1	310.40 (99.1)	1.6803 (243.7)	580 (36.25)	-580 (-248.92)	6.30 (1.505)	-
3.2	339.8 (152)	0.643 (93.22)	4.00 (0.25)	650 (279.00)	10.65 (2.54)	-
4	260.93 (10)	0.2655 (38.51)	12.50 (0.781)	-580 (-248.92)	6.375 (1.523)	18
4.1	285 (53.33)	0.648 (94)	39 (2.437)	-565 (-242.49)	6.40 (1.528)	13
4.2	285 (53.33)	0.648 (94)	39 (2.437)	-565 (-242.49)	6.40 (1.528)	13
3.3	305.37 (90)	1.676 (243.1)	584 (36.5)	-612 (-262.66)	6.21 (1.483)	-
4.4	260.93 (10)	0.2655 (38.51)	14 (0.875)	-612 (-262.66)	6.409 (1.531)	16

(R1) Based on ASHRAE method in reference 1.

TABLE - 3

ANALYSIS OF THE ENERGY CONSUMED BY AN AIR CONDITIONING SYSTEM FOR A PHARMACEUTICAL INDUSTRY REQUIRING CONTROLLED CONDITIONS IN HUMID CLIMATE

PARTICULARS OF ENERGY CONSUMPTION		PARTICULARS OF SYSTEM	
		OLD SYSTEM USING REHEAT FROM EXTERNAL SOURCE OF ELECTRICITY	NEW SYSTEM USING REHEAT FROM THE HEAT OF REJECTION
MAXIMUM VALUE OF INSTANTANEOUS POWER			
REFRIGERATION	(kW)	156.11	151.56[b]
REHEAT	(kW)	93.31	-
TOTAL	(kW)	249.42	151.56
ANNUAL PEAK[c] DEMAND	(kW)	2,370	1,440
ANNUAL ENERGY[d] CONSUMPTION	(kWh)	1,516,166	921,414
DEMAND PENALTY	($)	4,976	3,024
ENERGY CHARGE	($)	101,078	61,428
TOTAL ENERGY CHARGE	($)	106,054	64,452

(a) Refrigeration Load : 355.9kW (101.1 TR), Heat Rejected : 465 kW (132 TR).

(b) Assumptions for Additional Subcooling of Refrigerant in the New Reheat Coils shown in the State Points (3.1 - 3.3) of Table - 2 and Figure - 3 are, ambient Wet Bulb Temp. = 26.11°C (79F),Cold Water from the Cooling Tower at Temp. = 30°C (86F), The Lowest Temp. at which the Refrigerant Liquid may be Cooled =32.3°C(90F)

(c) Annual Peak Demand is the Sum of Monthly Peak Demands in a Year.

(d) Annual Energy Consumption is the Sum of Monthly Energy Consumptions in a Year.

(e) Energy Tariffs are for Locations in India and are based on the present rate of US$1.00 = Rs.30.00

(f) The cold water temperature from cooling tower is based on the method in reference 4.

LIFE CYCLE COSTING

An analysis of the owning and the operating costs of the old system versus the new one was done as follows (2) :

Owning Costs			US $ (1)
Engineering design	...	...	2,000
Heating Coils	...	...	7,500
Diverting Valves	...	...	10,000
Isolating Valves	...	...	2,000
Non-return Valves	...	...	1,500
Piping and Fittings	...	...	4,000
Temperature Controllers		...	10,000
Installation	...	...	2,000
Total of owning costs (Present Worth of Investment), PW =			$39,000

Annual Operating Costs			
Spares Replacement	...	...	250
Maintenance	...	...	1,000
Insurance Premium (Accident, Fire)			200
Total of Annual Operating Costs		...	$ 1,450

Analysis of Costs

First Year's gross savings in energy = $106,054 - $64,452 = $ 41,602

Net Annual savings in energy = $ 41,602 - $ 1,450 = $ 40,152

Rate of interest on borrowings, i = 20 %

Rate of escalation, j = 13 %

$$PW = \text{Net annual savings} \times \frac{\left[\dfrac{1 + j}{1 + i}\right]^n - 1}{\left[1 - \dfrac{1 + i}{1 + j}\right]}$$

Hence, the number of years to pay-back, n works out to one year.

NOTES :

(1) These costs apply to Indian conditions and are based on the present rate of;
US $ 1.00 = Rs.30.00

(2) The life cycle costing is based on ASHRAE method in reference 2.

Figure - 4 shows the schematic layout of the system with the reheat coils using hot gas from the high stage compressor. The condenser was sized for the maximum heat of rejection. This ensured that adequate subcooling (state point 3.1 in Table - 2 and Figure - 3) occurred in case the reheat coils used minimum of hot gas. Backup arrangement for the reheat coils was not necessary since the requirement of reheat and the availability of the rejected heat are simultaneous occurrences.

CONCLUSIONS

The life cycle costing indicates that the pay-back period of the new system modifications works out to one year. Hence this method of heat recovery is very cost effective. Especially in large industrial applications, where refrigeration and air conditioning systems constitute some of the most energy intensive activities, a substantial conservation of energy can be achieved.

The heat of rejection can also be used to pre-heat water in the applications such as distillation plants and boiler feed water.

REFERENCES

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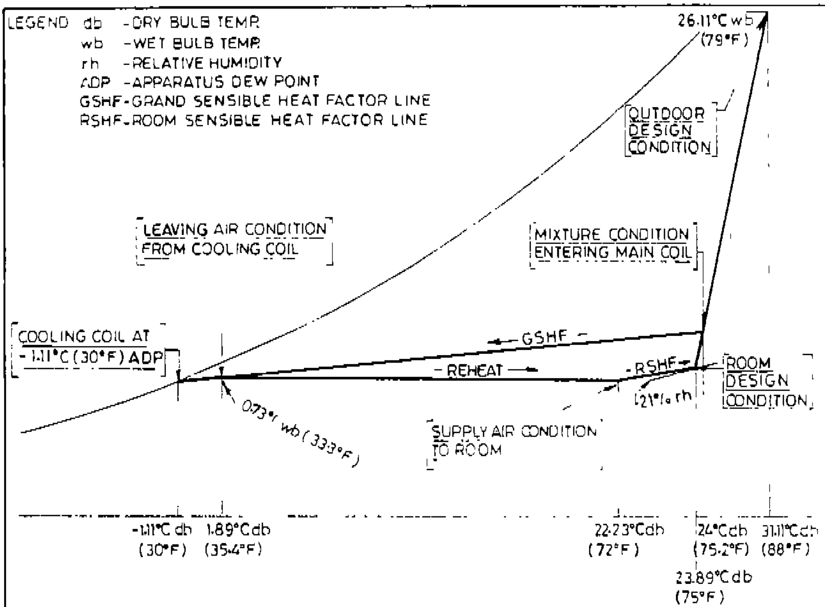

FIGURE:1 _ PSYCHROMETRIC PROCESS FOR THE SECOND DRYER IN MONSOON

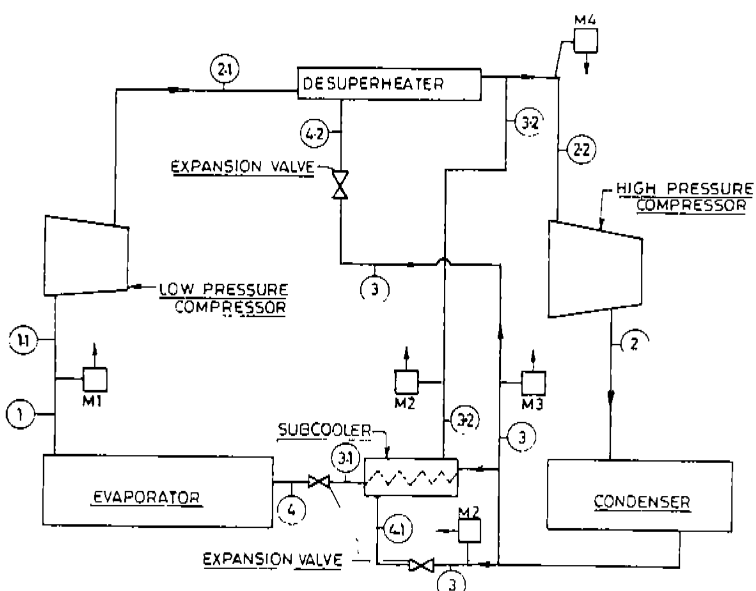

FIGURE-2-SYSTEM COMPONENTS OF TWO-STAGE COMPRESSION CYCLE

ASHRAE DIAGRAM

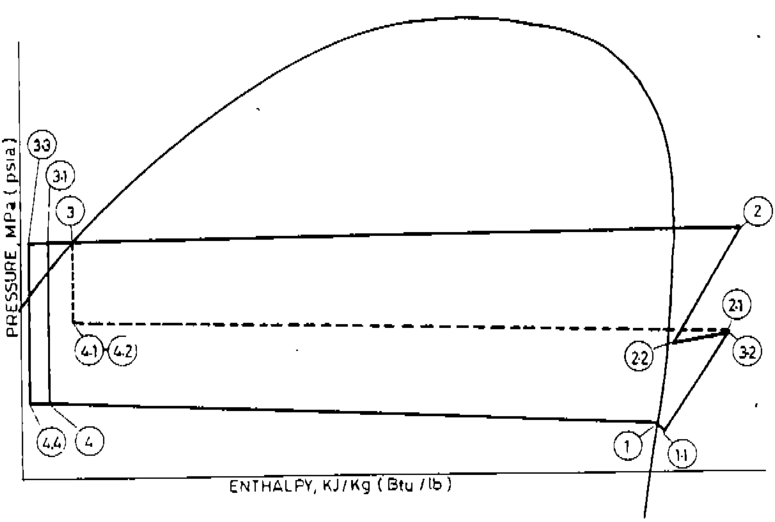

FIGURE: 3 - TWO STAGE COMPRESSION CYCLE

ASHRAE DIAGRAM

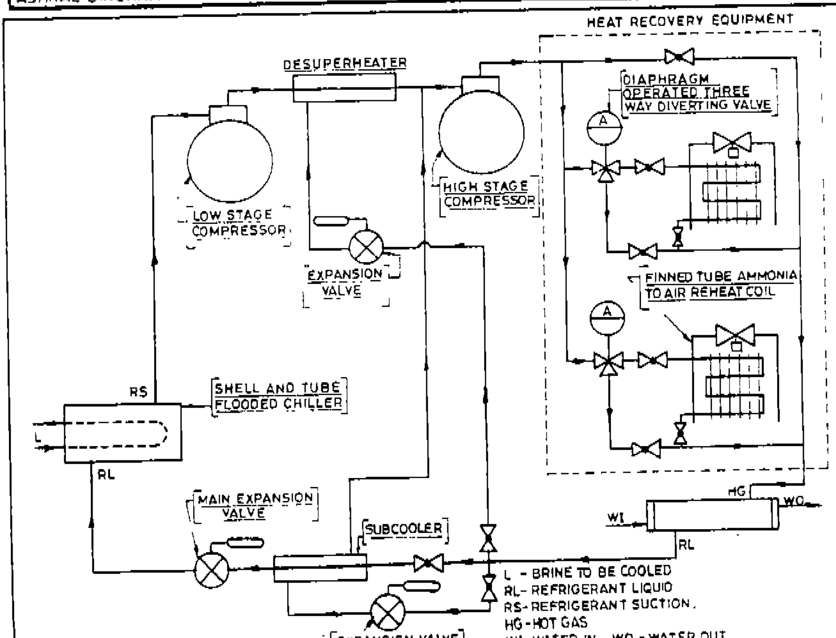

FIGURE:-4 SCHEMATIC OF HEAT RECOVERY FROM THE REJECTED HEAT