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APPLICATION OF CHEMICAL DEHUMIDIFICATION SYSTEM

TO

A ROOF FAN HOUSE AT MICHOUD ASSEMBLY FACILITY

AT

NEW ORLEANS, LOUISIANA

NASA CONTRACT NO. NASW-2920

JULY 15, 1976

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EXECUTIVE SUMMARY

A. INTRODUCTION

Gershon Meckler Associates, P.C., has prepared this report to assess the feasibility of incorporating a "chemical dehumidification system" to reduce the energy consumption associated with dehumidification of the chilled air supplied for the environmental control of Michoud Assembly Facility (MAF) New Orleans, Louisiana. This report incorporates the proposed system suggested in our preliminary engineering report dated May 1, 1976 as well as a response to a assessment report prepared by the Grumann Aerospace Corporation Energy Programs Group, under NASA Contract No. NASI - 14387, Task order No. 8 Part B. This report is intended to incorporate Grumann comments into a comparative energy consumption and cost analysis of the chemical dehumidification and existing systems and document the savings offered by the proposed chemical dehumidification system over the existing air washer-reheat system.

B. CRITERIA

Analysis of the proposed chemical dehumidification system is based upon:

- 40% duty cycle
- Investment cost and pay back periods, calculated according to NASA's "Calculations of "Pay Back" for Direct Energy Projects," directive dated July 7, 1976.
- Cost of gas $1 per million Btu as per the actual billing structure.

C. RESULT OF ANALYSIS

Analysis of the proposed chemical dehumidification system indicates that it offers a decrease in steam, electricity and water consumptions as compared with the existing air washer-reheat system. The reduction in steam, electricity and water consumption are given below:

Steam	- 3,637,037 lb/year
Electricity	- 1,547,366 KWH/year
Water	- 43,710,397 gallons/year

The following table shows simple payback periods of the proposed chemical dehumidification system.

Total Investment	Cost of Energy		Cost of Water	Total Savings	Simple Payback Period (without escalation)	Simple Payback Period (with escalation)
	per 10^6 Btu	per KWH	per 1000 gal			
$	$	$	$	$	Years	Years
5.088×10^6	1.0	0.03	1.5	340,549	14.9	10
5.088×10^6	3.6	0.03	1.5	935,172	5.44	4.5

D. RECOMMENDATIONS

Our preliminary report dated May 1, 1976 indicated 2.5 years simple payback period which was based on 100% duty cycle and steam cost of $3.60 per million Btu. Subsequent to our preliminary report we have been advised to use a 40% duty cycle and fuel cost of $1.00 per million Btu. With this input simple payback periods with and without escalation have been recalculated and found as shown in the above table.

Based on an anticipated rise in energy costs, and the potential increase in the usage factor of this facility, it is recommended that an initial pilot installation be made in one, representative, roof fan house on Building #103, to be selected by NASA. This will confirm projected energy consumption and cost savings to be achieved for the overall facility.

Furthermore, instrumentation and data acquisition is recommended to achieve direct comparative verification of the actual energy consumption and cost savings associated with chemical dehumidification on the basis of actual performance. This prototype installation will then serve as a basis for the overall plantwide modification to achieve the projected energy consumption and cost savings.

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A. INTRODUCTION

This report presents results of an analysis to assess the feasibility of incorporating a chemical dehumidification system to reduce the energy consumption associated with dehumidification of the chilled air supply for environmental control at the Michoud Assembly Facility (MAF), New Orleans, Louisiana. This report contains an analysis of the proposed system suggested in our preliminary engineering report dated May 1, 1976, as well as a response to an assessment report prepared by the Grumman Aerospace Corporation Energy Programs Group, under NASA Cotnract No. NASI - 14387, Task order No. 8 Part B. This report will incorporate Grumman's comments into a corporative energy consumption and cost analysis of the chemical dehumidification and existing systems and document the savings offered by the proposed chemical dehumidification system over the existing washer-reheat system.

B. DESIGN BASIS AND CRITERIA FOR EVALUATION

Both the existing and proposed systems have been analyzed on the same performance basis as follows:

1.* The required space condition is 75°F, 50% R.H., as called for by the control setpoints on the 1974 fan house modification drawings.

2.* The sensible heat ratio of the internal space load is 0.62, in accordance with the equipment design as shown by the 1974 fan house modification drawings. This was calculated on the basis that the internal sensible load is 18 Btu/hr.ft.^{2} (10 Btu/hr ft.^{2} roof load + 8 Btu/hr.ft.^{2} for the people, lights, and equipment), and that the latent load is such as to require the present design to provide 54,000 CFM at 50°F saturated air.

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*Reconformation of these design conditions was made with the operating personnel at MAF. They state that both design conditions are valid assumptions. However, design conditions specified in Item2 should be reconfirmed by accurate on-site measurements.

3. Analysis of local weather data indicates that outside air can be utilized for space cooling and dehumidification for approximately 20% of the daily cycle.

4. The present average is approximately 25 fan houses operating at any given time for a period of 10 hours per day on single shift operations.

5. 5 fan houses operate during unoccupied periods, including weekends.

6. The impact of the shuttle Enternal Tank (ET) fabrication will probably require a second shift operation. This will increase weekday operation to 20 hours/day - 5 days/week.

   Therefore, a duty cycle for the fan houses can be established as:

$$\text{*Duty Cycle} = \frac{(20 \text{ units} \times 100 \text{ hrs./wk}) + (5 \text{ units} \times 168 \text{ hrs./wk})}{43 \text{ units (total)} \times 168 \text{ hrs./wk}} = 0.4$$

7. The investment cost and pay back periods are calculated according to NASA "Calculations of Pay Back for Direct Energy Projects."

8. Cost of gas is $1 per million Btu, as per the actual billing structure.

C. DESCRIPTION OF EXISTING SYSTEM

The existing HVAC system for the MAF facility consists of a central plant's utility building housing high pressure steam boilers generating 210 PSIG high pressure steam and providing approximately 14,000 tons of refrigeration. The steam boiler plant provides 210 PSIG steam to approximately 14,000 tons of refrigeration. The refrigeration plant consists of steam turbine driven centrifugal machines providing 42°F chilled water for air conditioning throughout the facility. 43°F chilled and 50 lbs steam are distributed to 43 roof mounted fan HVAC chilled air systems varying in size from 81,000 CFM to 59,800 CFM, together providing a total of 2,960,000 CFM for Building #103.

Each chilled air supply system is housed in individual roof mounted fan houses arranged to include fresh air, return air, inlets, mixing

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*This criterion has been established by assessment report prepared by the Grumann Aerospace Corporation Energy Programs Group, under NASA Contract No. NASI - 14387, Task order No. 8 Part B.

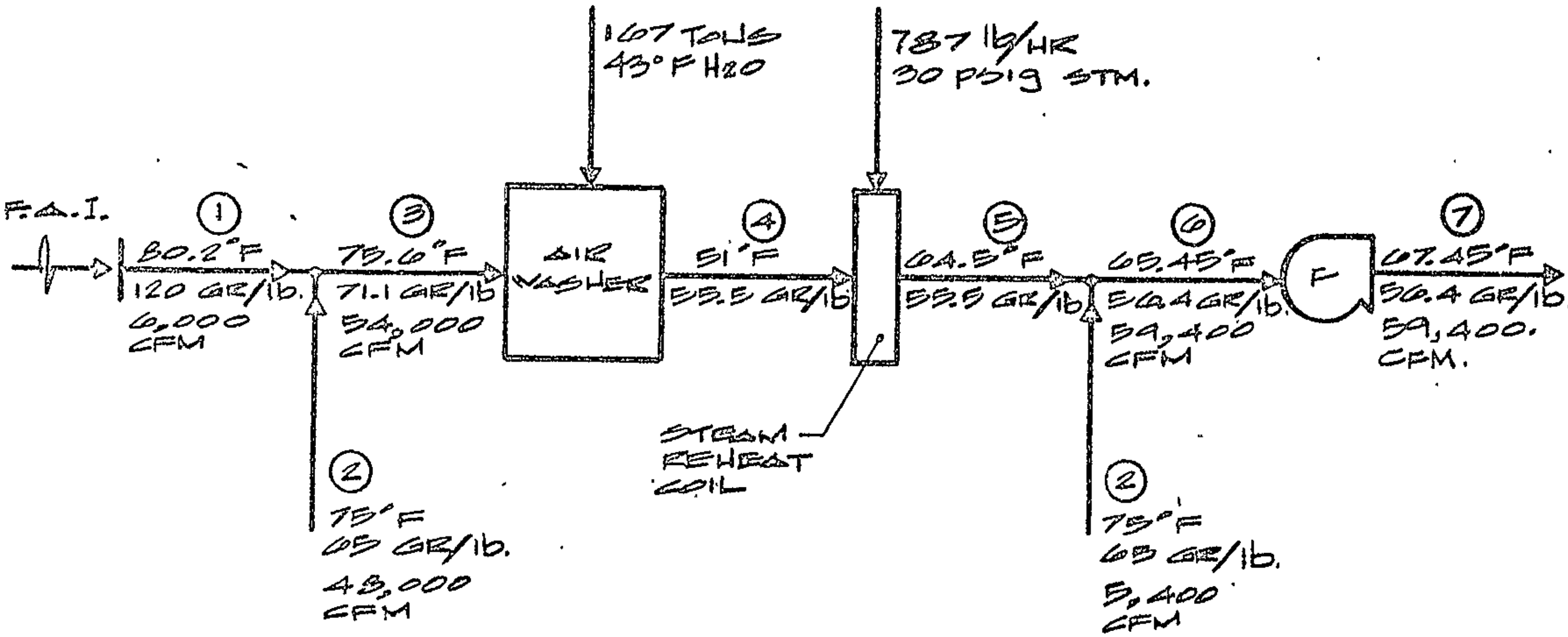
plenum, filter section, chilled water spray washer, reheat coil and bypass dampers. Each fan house system is designed to provide 50°F dew point air for cooling and dehumidification.

The spray washer section requires 43°F chilled water to maintain 50°F dew point. The existing chilled air system cools and dehumidifies approximately 90% of the total air stream to 50°F saturated.

The outdoor make-up air (approximately 10% of the total) mixes with 10% return air and is supplied to the space at approximately 65°F and 55 grains/lb. NASA has had many studies in the past that have highlighted the inefficiencies of this system, particularly with respect to the use of a spray washer to provide cooling and dehumidification. However, because of the low cost of fuel in the past and the scarcity of modifications, these changes were not implemented.

In 1974, the fan houses were rehabilitated and modified, the primary purpose being to replace worn out equipment. In addition, a new 4,000 ton chiller was installed in the central utility plant, primarily to provide for more efficient loading of the refrigeration plant. Fundamentally the existing system provides dehumidification and cooling requirements by steam generated refrigeration and reheating. This process is characterized by a psychrometric path established by the chilled spray washer followed by a reheat coil to maintain proper conditions. Figures 1 through 6 illustrate the system and the psychometric analysis of the existing systems at different seasonal conditions.

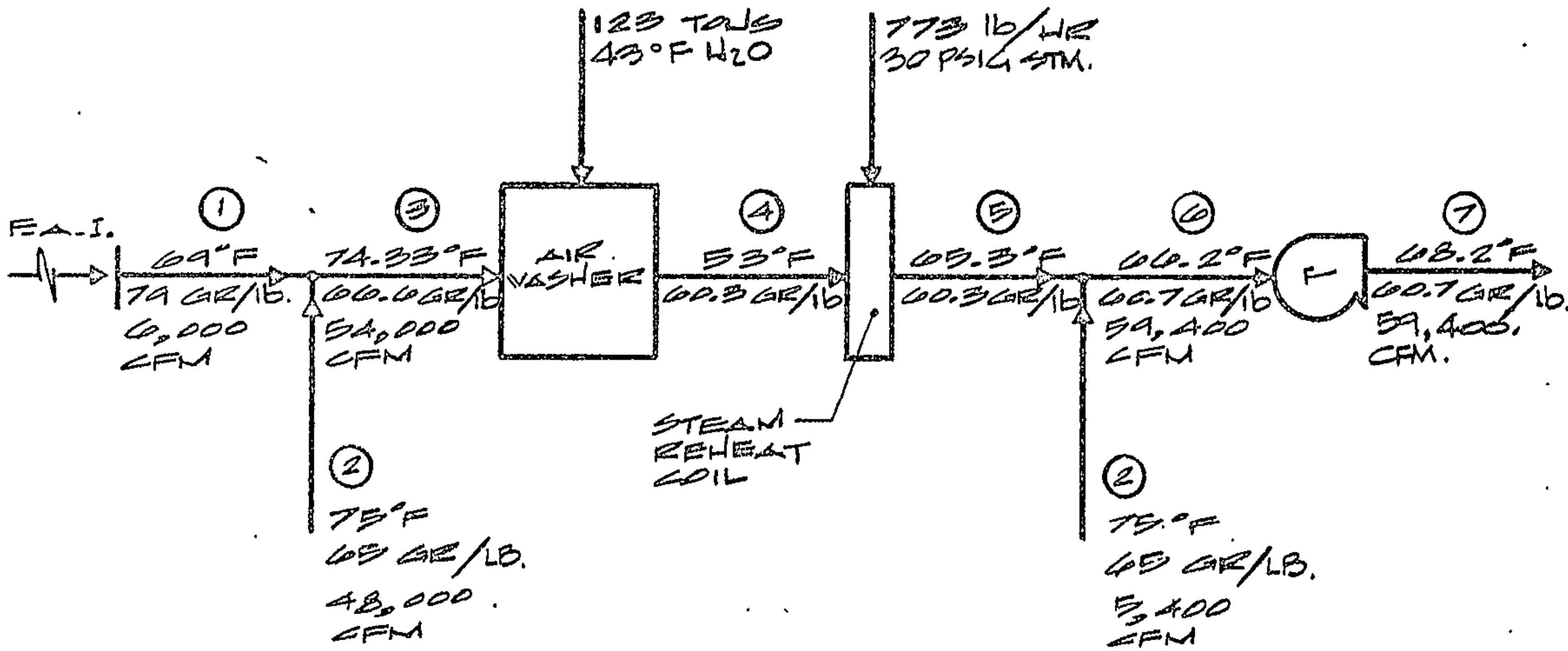
The two key characteristics of the existing system are 1) a minimum of 43°F chilled water is required and 2) reheating must be used continuously to maintain proper balance between dehumidification and sensible cooling. This type of system wastes energy since it requires low temperature refrigeration (43°F chilled water) to achieve the proper dew point for dehumidification control and then must be reheated by steam.



SYSTEM ANALYSIS OF EXISTING SYSTEM

SUMMER AVERAGE CONDITION

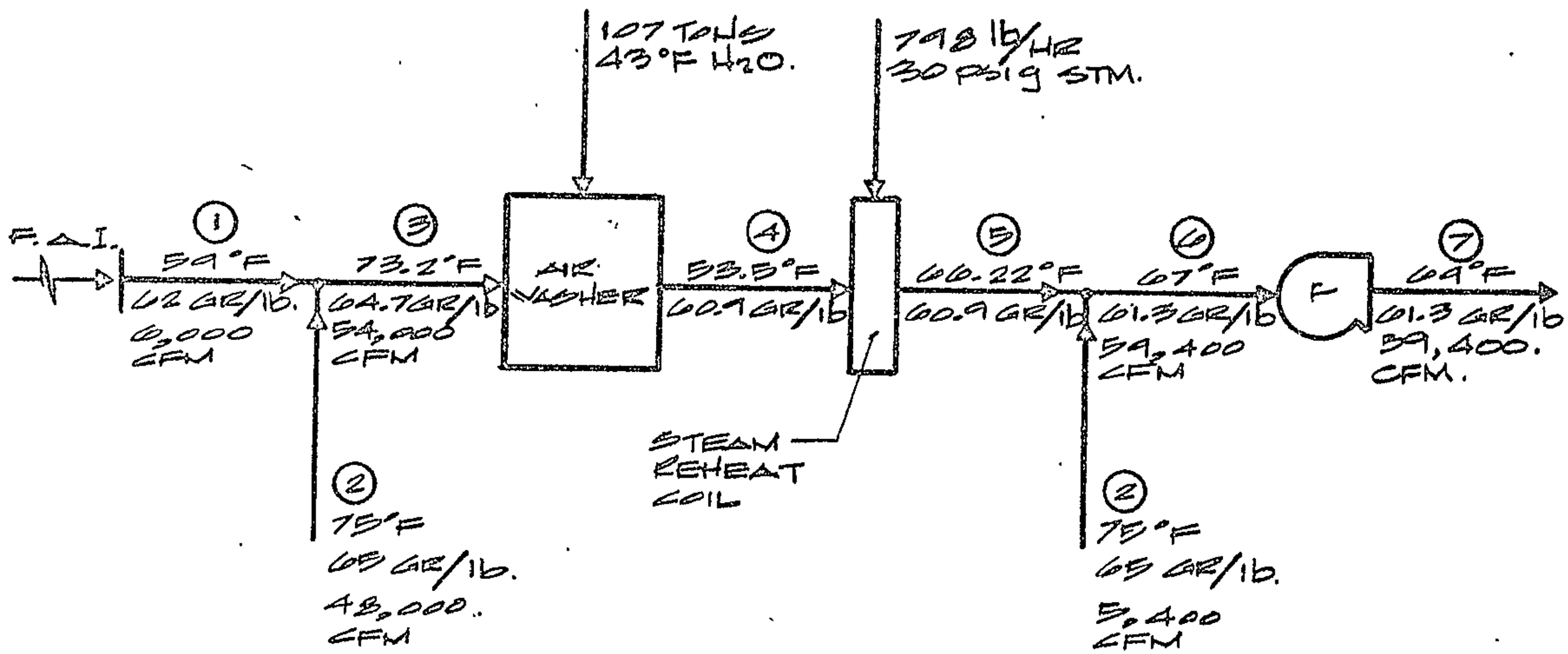
FIGURE 1



SYSTEM ANALYSIS OF EXISTING SYSTEM

FALL/SPRING AVERAGE CONDITION

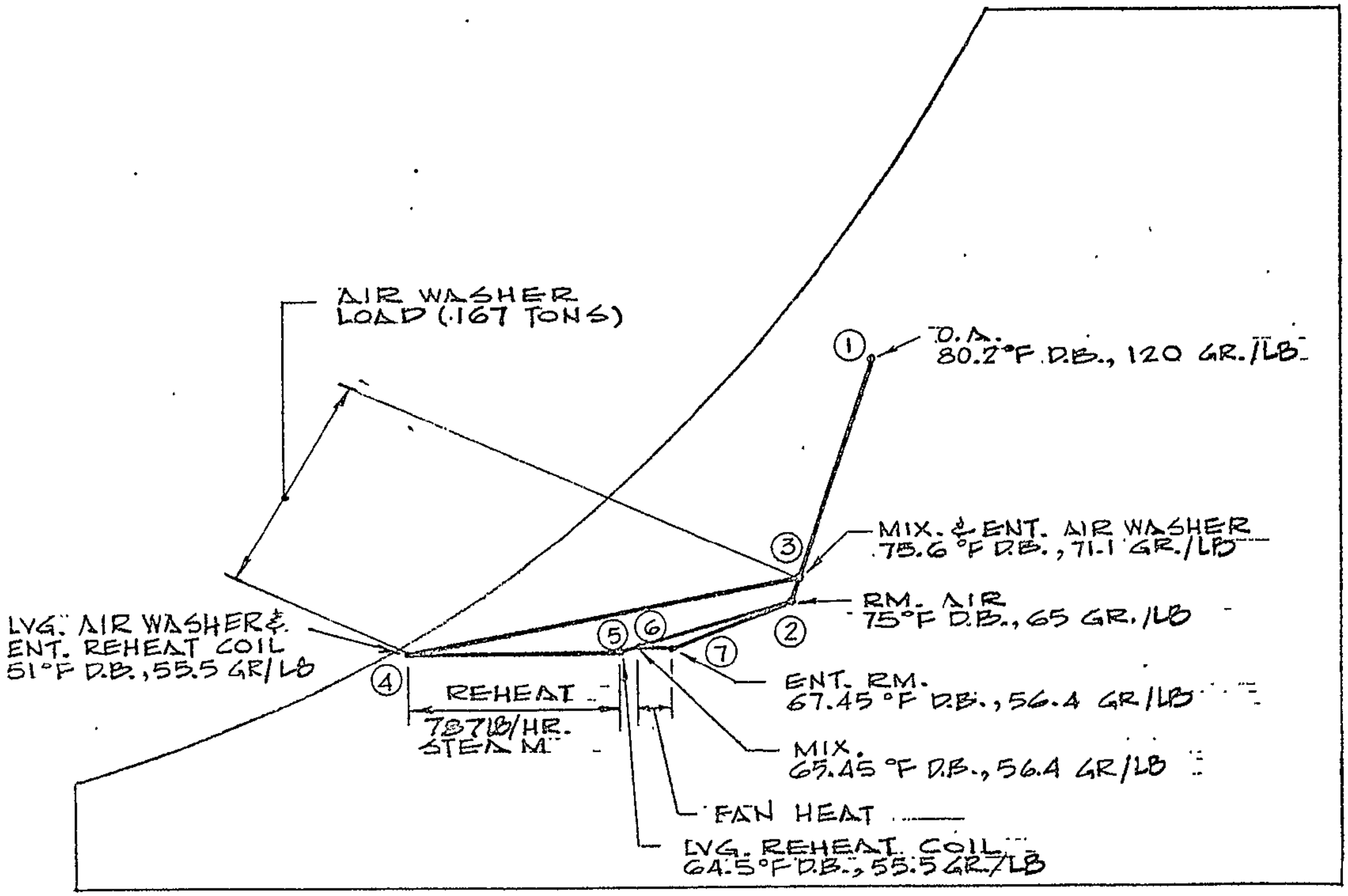
FIGURE 2



SYSTEM ANALYSIS OF EXISTING SYSTEM

WINTER AVERAGE CONDITION

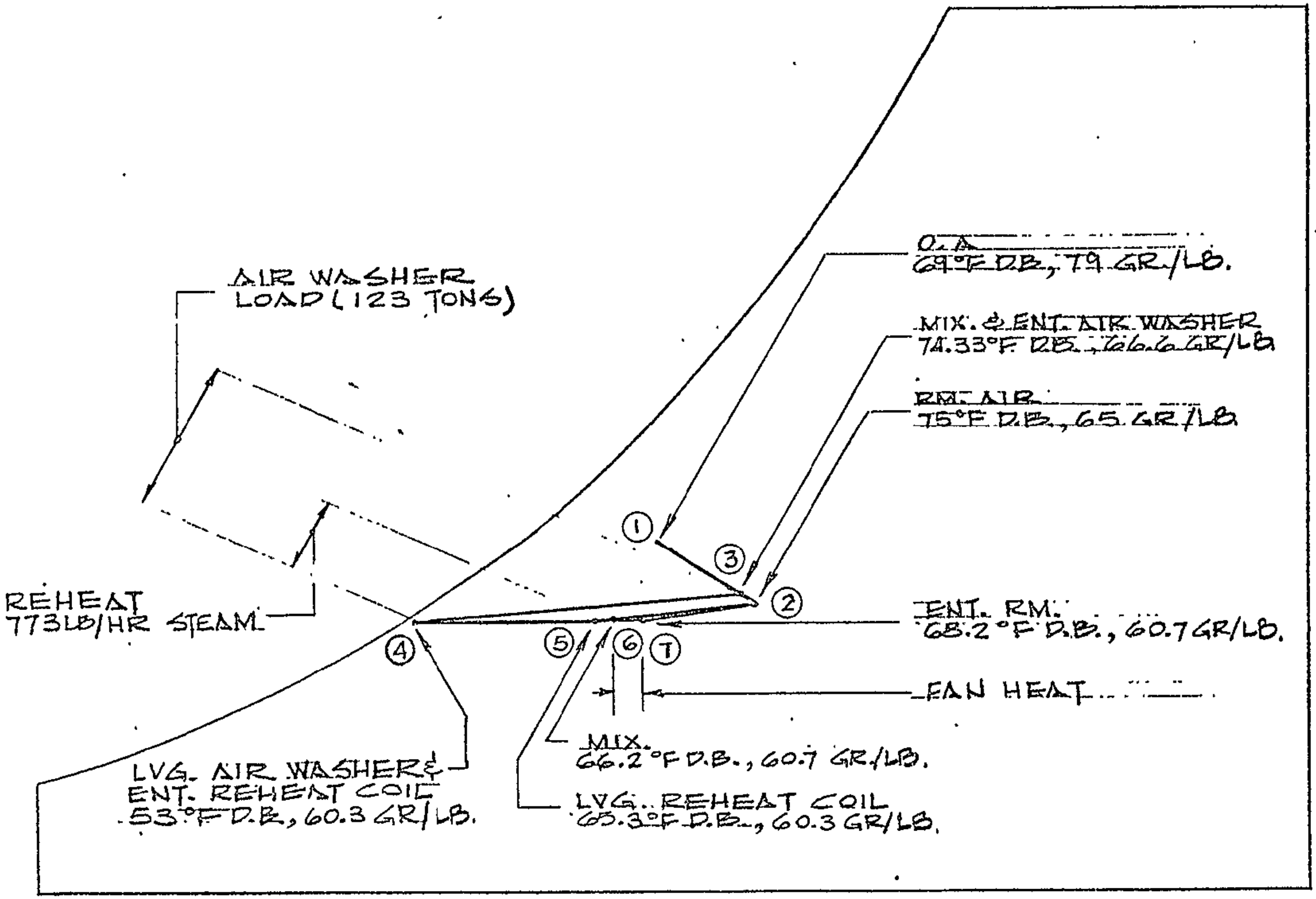
FIGURE 3.



PSYCHROMETRIC ANALYSIS OF EXISTING SYSTEM

SUMMER AVERAGE CONDITION

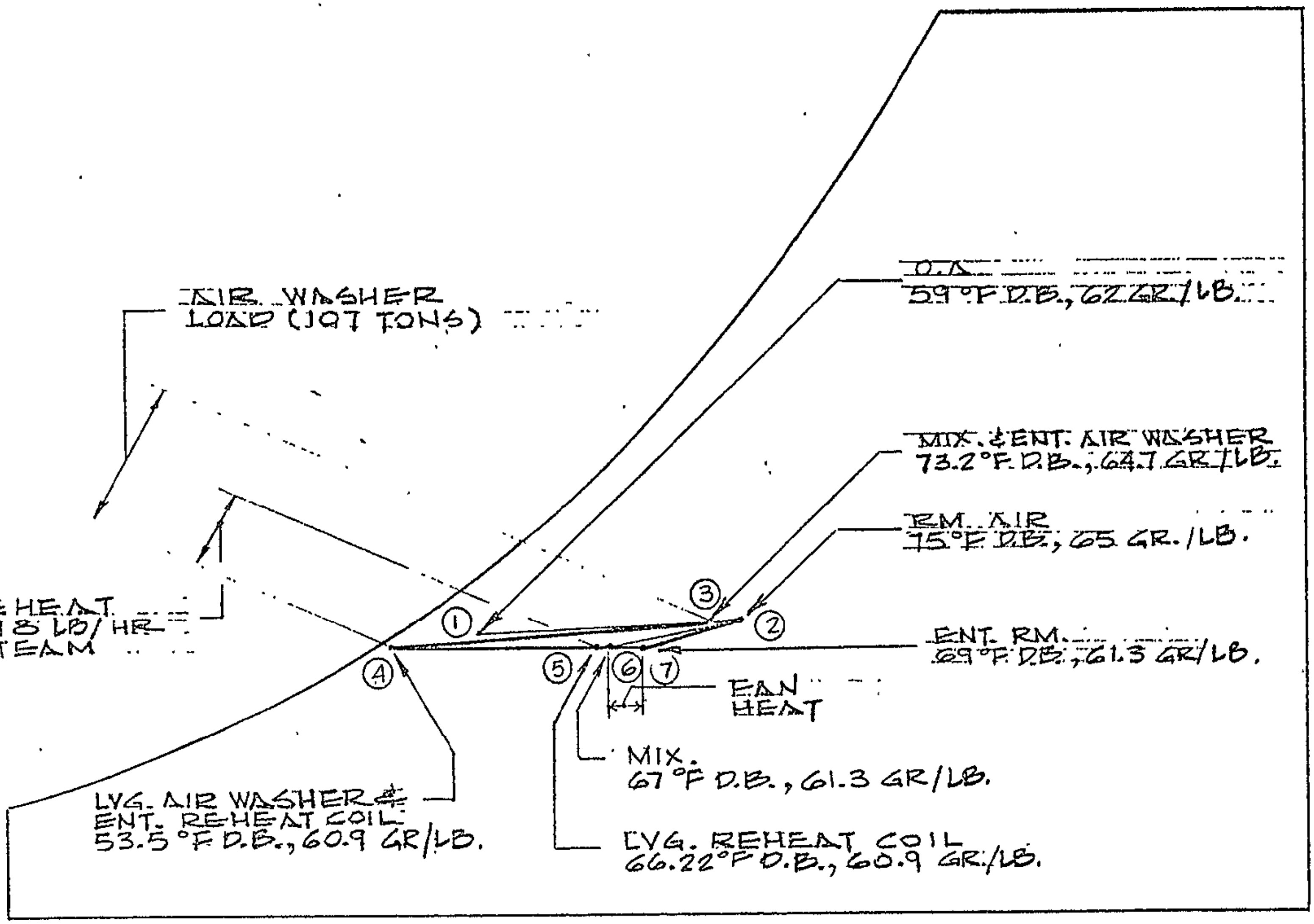
FIGURE 4



PSYCHROMETRIC ANALYSIS OF EXISTING SYSTEM

FALL/SPRING AVERAGE CONDITION

FIGURE 5



PSYCHROMETRIC ANALYSIS OF EXISTING SYSTEM

WINTER AVERAGE CONDITION

FIGURE 6

Under partial load conditions the set point of the discharge air temperature can only be increased to a maximum of 3.5°F from its original setting and still maintain the required relative humidity in the conditioned space. In order to maintain temperature and humidity control reheat is manditory with the existing system.

D. PROPOSED CHEMICAL DEHUMIDIFICATION SYSTEM MODIFICATION

Process Description

Moist air required for air conditioning and ventilation may be dehumidified by direct contact with a liquid absorbent solution. Such a system has two basic sections, each with its own pumping unit, namely, a dehumidifying section and a regeneration section. Cooling coils in the dehumidifier maintain the solution and conditioned air at a constant temperature. A portion of the moisture laden solution returning from the dehumidifying section is pumped to the regeneration section for concentration. Regeneration is accomplished by passing the solution and an outside air stream over steam heated coils, where the water is driven from the solution into scavenger air. The steam supply to this coil is modulated by a solution level controller. A constant level controller assures a constant solution concentration, which, coupled with a controlled dehumidifier discharge temperature, maintains a constant supply air dew point.

The absolute degree of dehumidification is a function of the liquid absorbent solutions' temperature and concentration, internal vapor pressure, and the efficiency of the contractor coil in the dehumidifer.

In this process, a reduced space latent load is sensed by the space humidistat, which positions automatic air dampers so as to reduce the amount of dehumidified air supplied to the space. To maintain a constant air flow through the dehumidifier, additional automatic dampers allow for recycling of the air through the dehumidifier. This process separates the dehumidification and cooling functions, allowing for maximum system operating efficiency.

Figures 7 through 12 illustrate the system and psychrometric analysis of chemical dehumidification at different seasonal conditions. Figure 13 shows a modification plan to Fan House #22 incorporating the proposed chemical dehumidification system.

E. ANALYSIS

Comparative energy consumption analysis of the existing system and the proposed modification which includes the chemical dehumidification system has been based on the following:

- The existing Fan House #22 system supplies approximately 59,000 CFM out of a total of a total of 2,986,300 CFM for Building #103 This represents 2% of the chilled air volume serving Building #103. The energy savings and cost of modification associates with Fan House #22 were scaled to establish the over-all pay back period for the entire facility.

- The average hourly steam consumption generating chilled water and reheat has been established using 24 hour average weather data for MAF, New Orleans, La. The weather data used is as follows:

Average D.B.	June-Sept.	80.2°F
Average D.P.	June-Sept.	72.4°F
Average D.B.	Dec.-March	57.1°F
Average D.P.	Dec.-March	51.3°F
Average D.B.	Oct.-Nov.-Apr.-May	69.0°F
Average D.P.	Oct.-Nov.-Apr.-May	60.3°F

- The utility costs are as follows:

Steam:	$3.60/10^6$Btu
Natural Gas:	$1/MCF
Electric:	$0.03/KWH

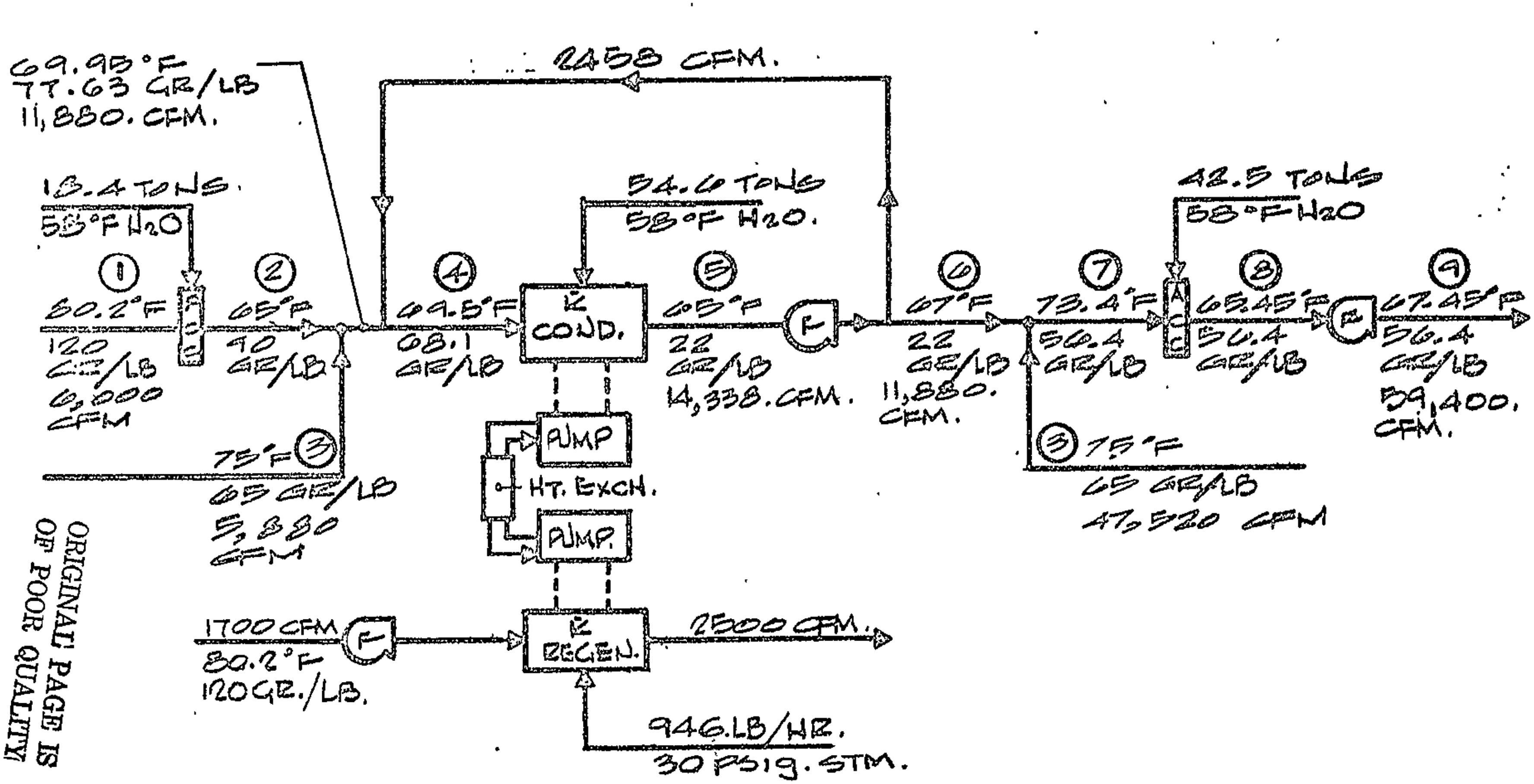
1. STEAM CONSUMPTION OF THE EXISTING SYSTEM

a) Summer Season

- Refrigeration

$$(167\text{ tons})\ (16\text{ lb }\frac{\text{steam}}{\text{ton}})\ (\frac{24\text{ hours}}{\text{day}})\ (122\text{ days/seasons})$$

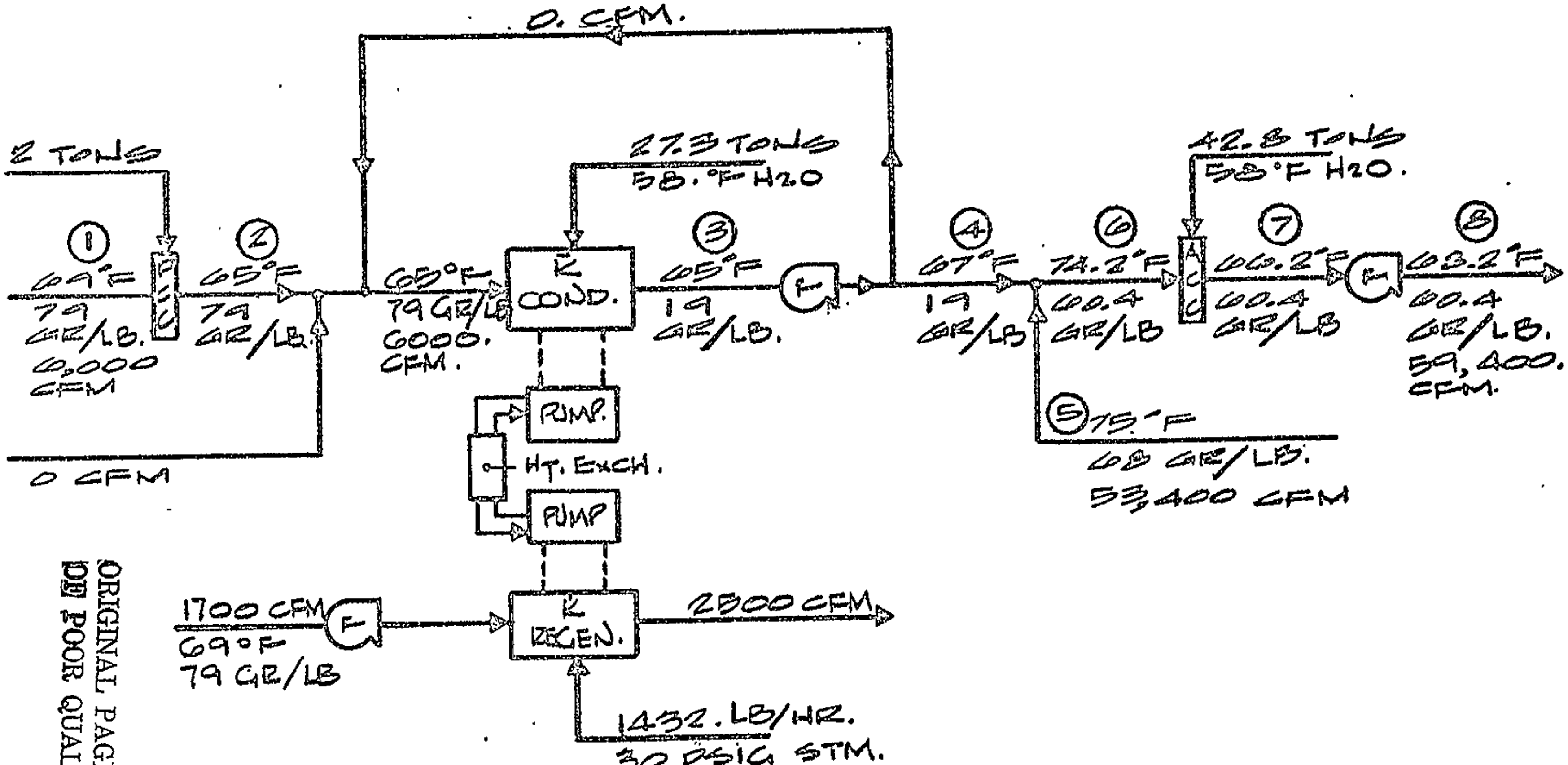
$$(0.4\text{ duty cycle factor}) = 3{,}129{,}446\text{ lb steam}$$



SYSTEM ANALYSIS OF PROPOSED SYSTEM

SUMMER AVERAGE CONDITION

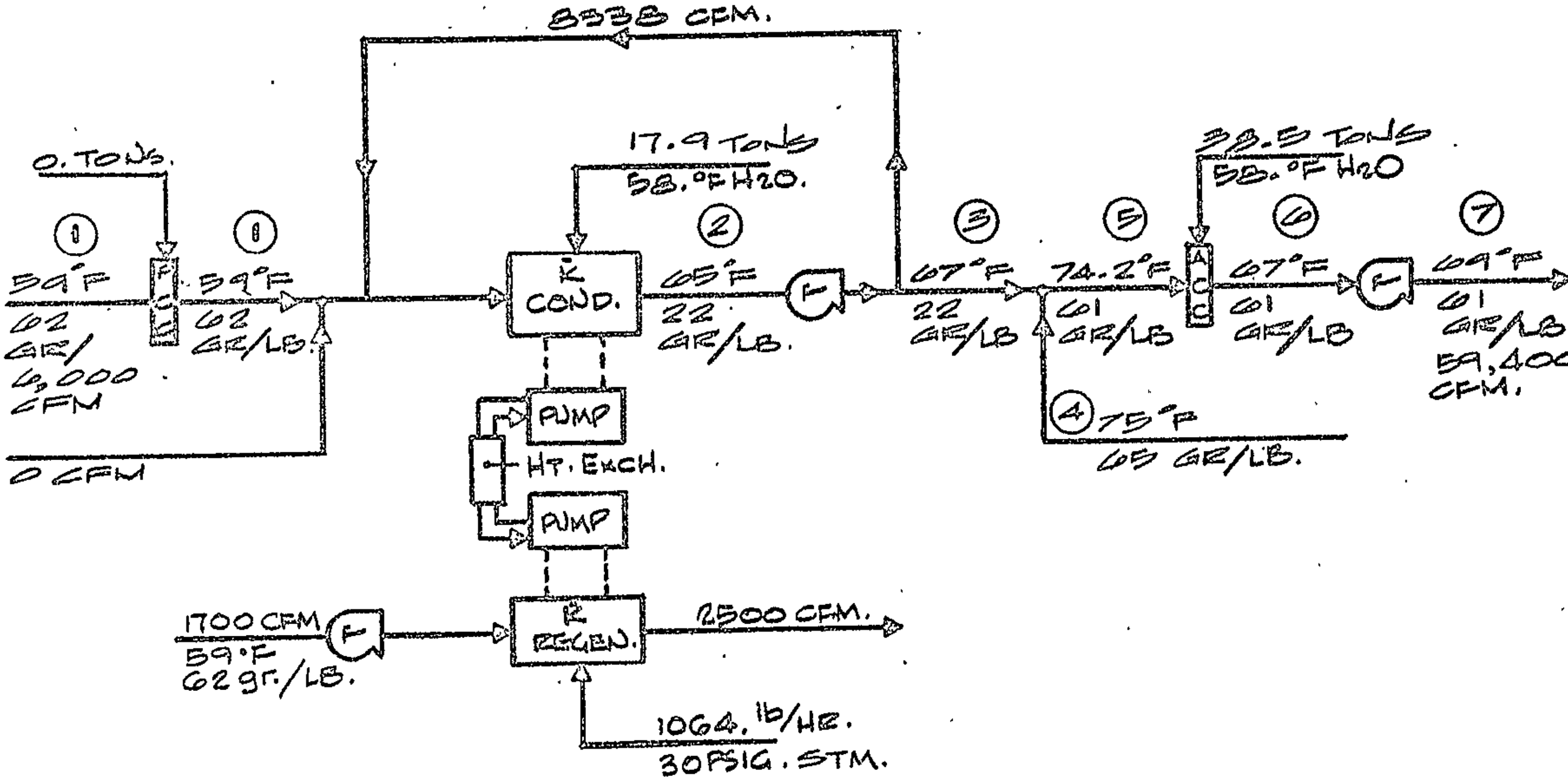
FIGURE 7



SYSTEM ANALYSIS OF PROPOSED SYSTEM:

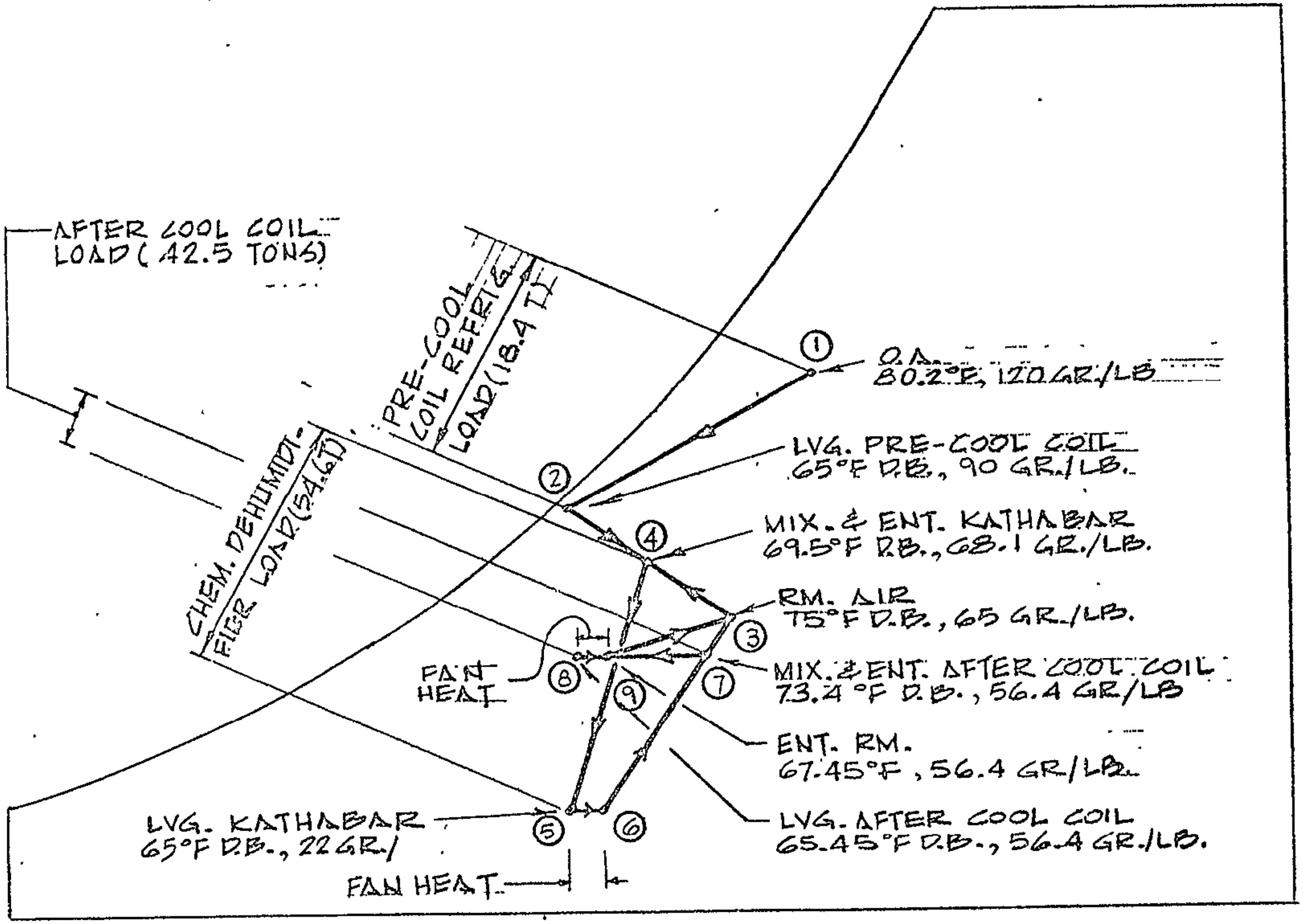
FALL / SPRING AVERAGE CONDITION

FIGURE 8



SYSTEM ANALYSIS OF PROPOSED SYSTEM:
WINTER AVERAGE CONDITION

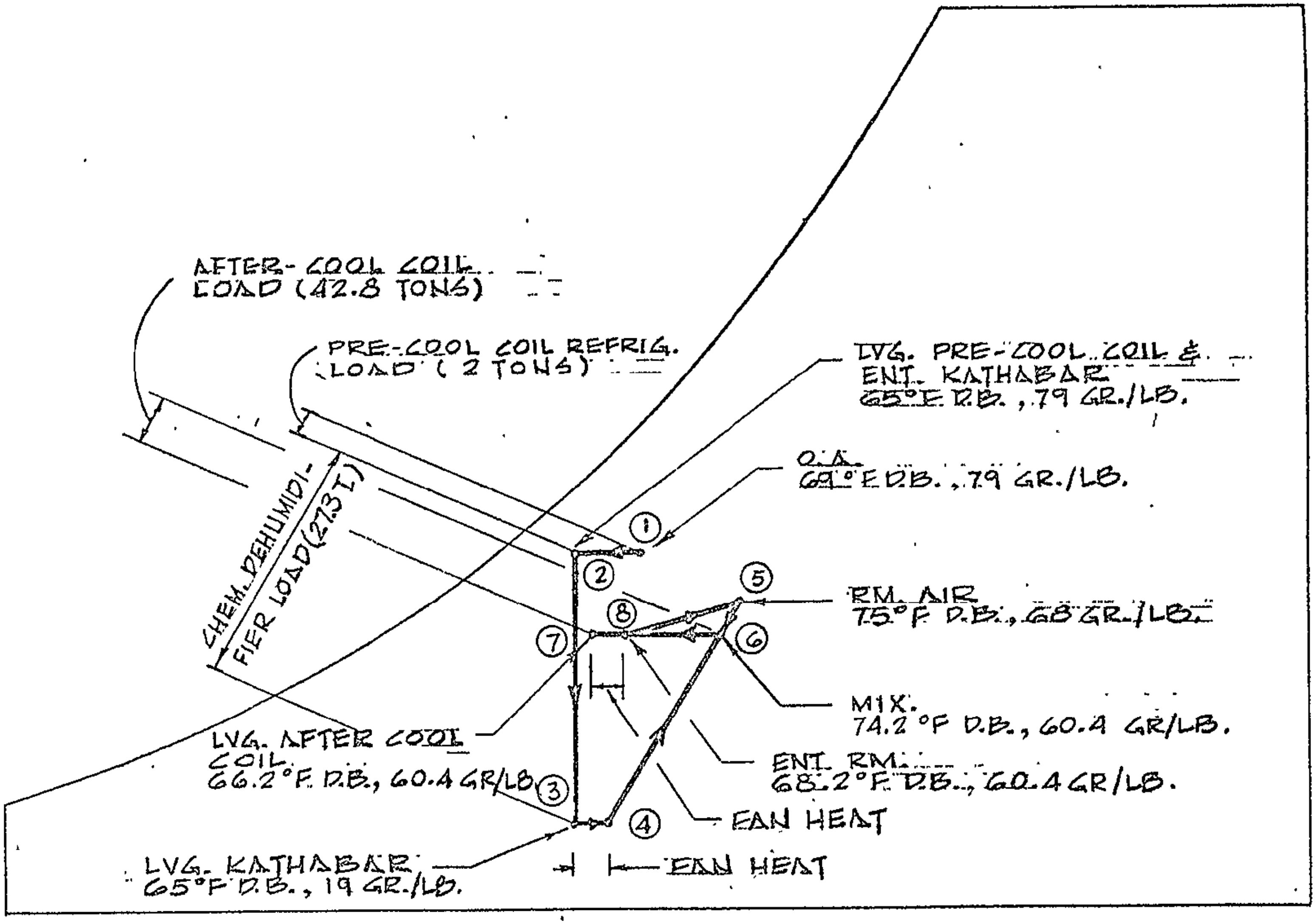
FIGURE 9.



PSYCHROMETRIC ANALYSIS OF PROPOSED SYSTEM

SUMMER AVERAGE CONDITION

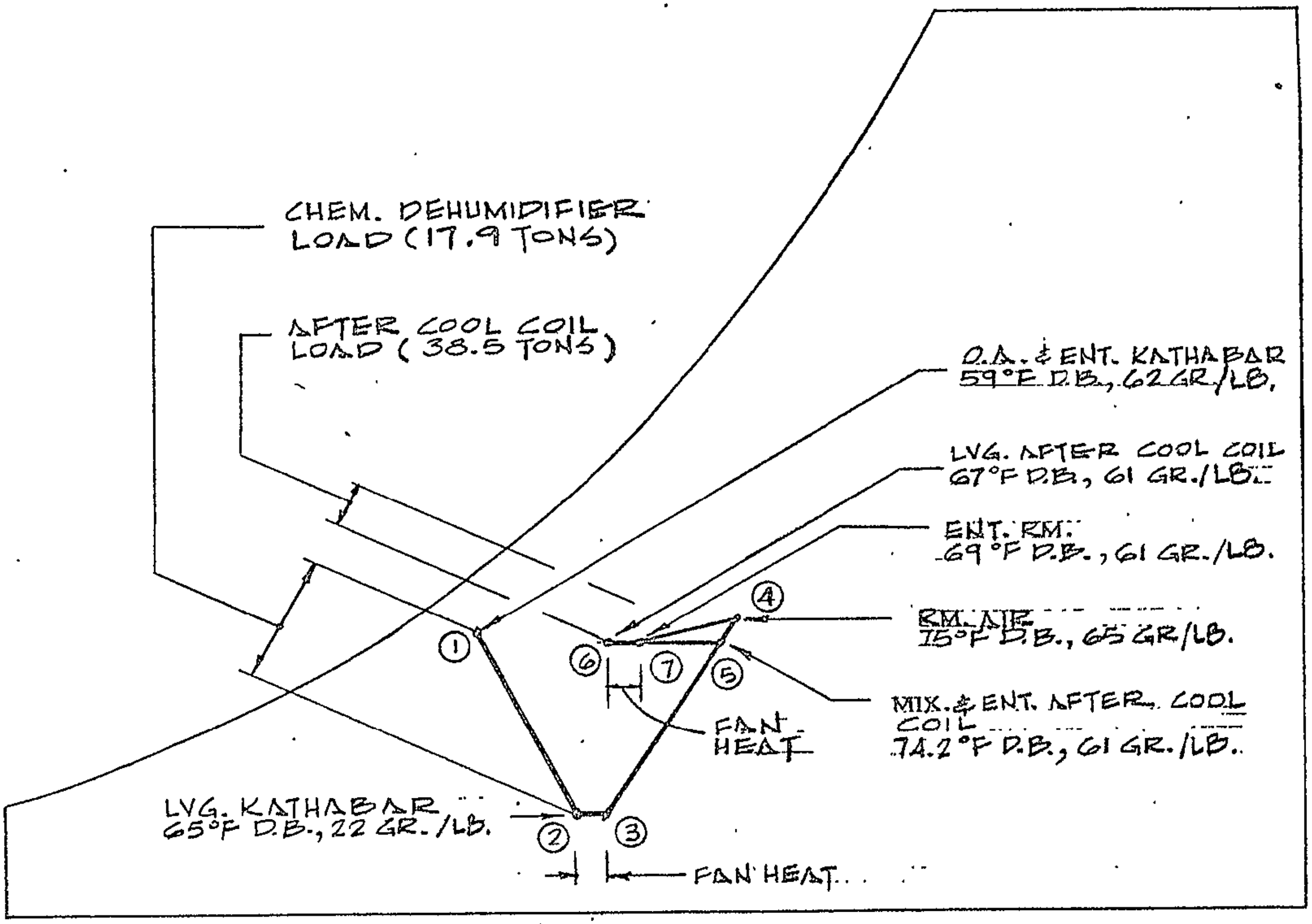
FIGURE 10



PSYCHROMETRIC ANALYSIS OF PROPOSED SYSTEM

FALL / SPRING AVERAGE CONDITION

FIGURE 11



PSYCHROMETRIC ANALYSIS OF PROPOSED SYSTEM

WINTER AVERAGE CONDITION

FIGURE 12

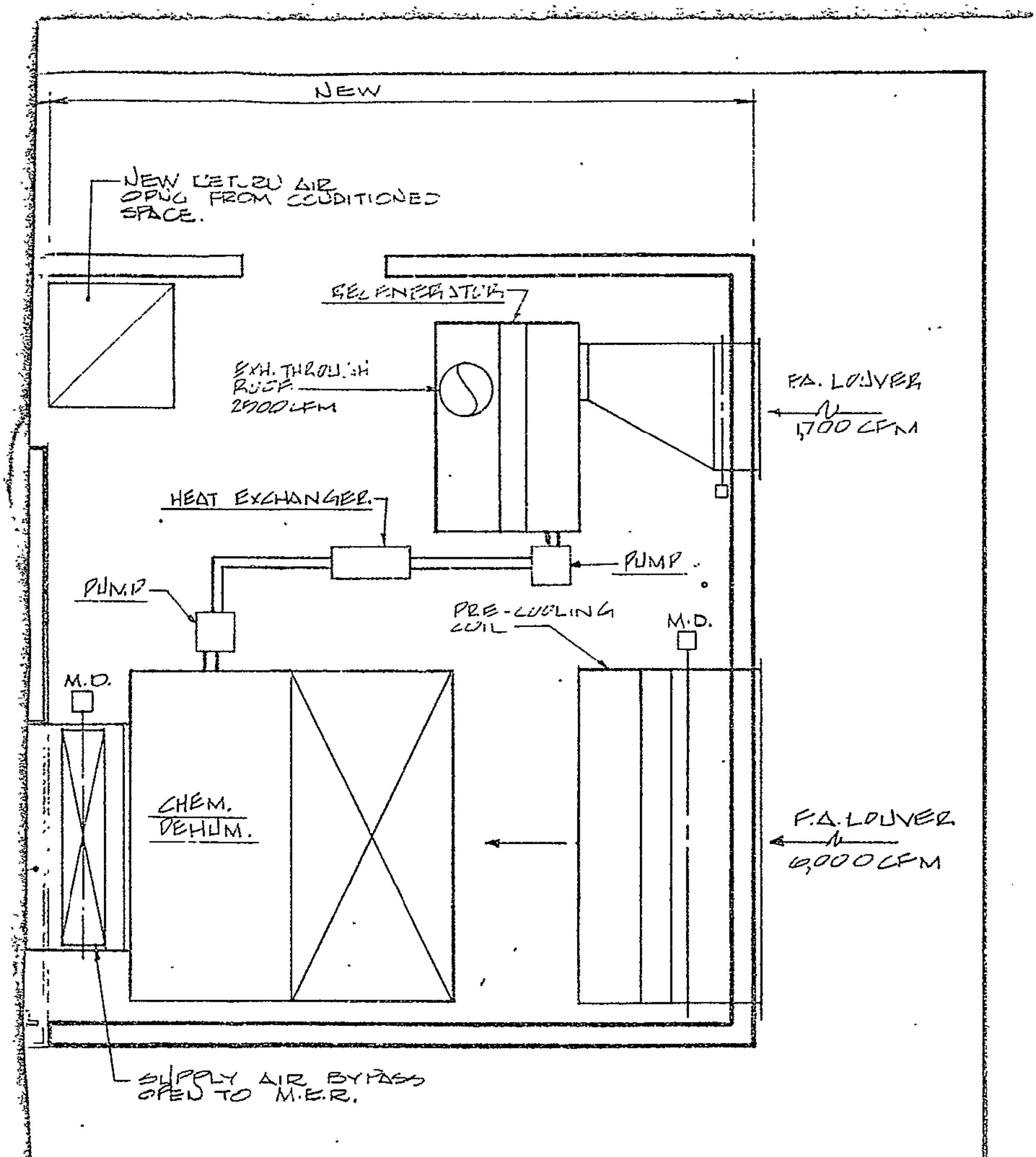
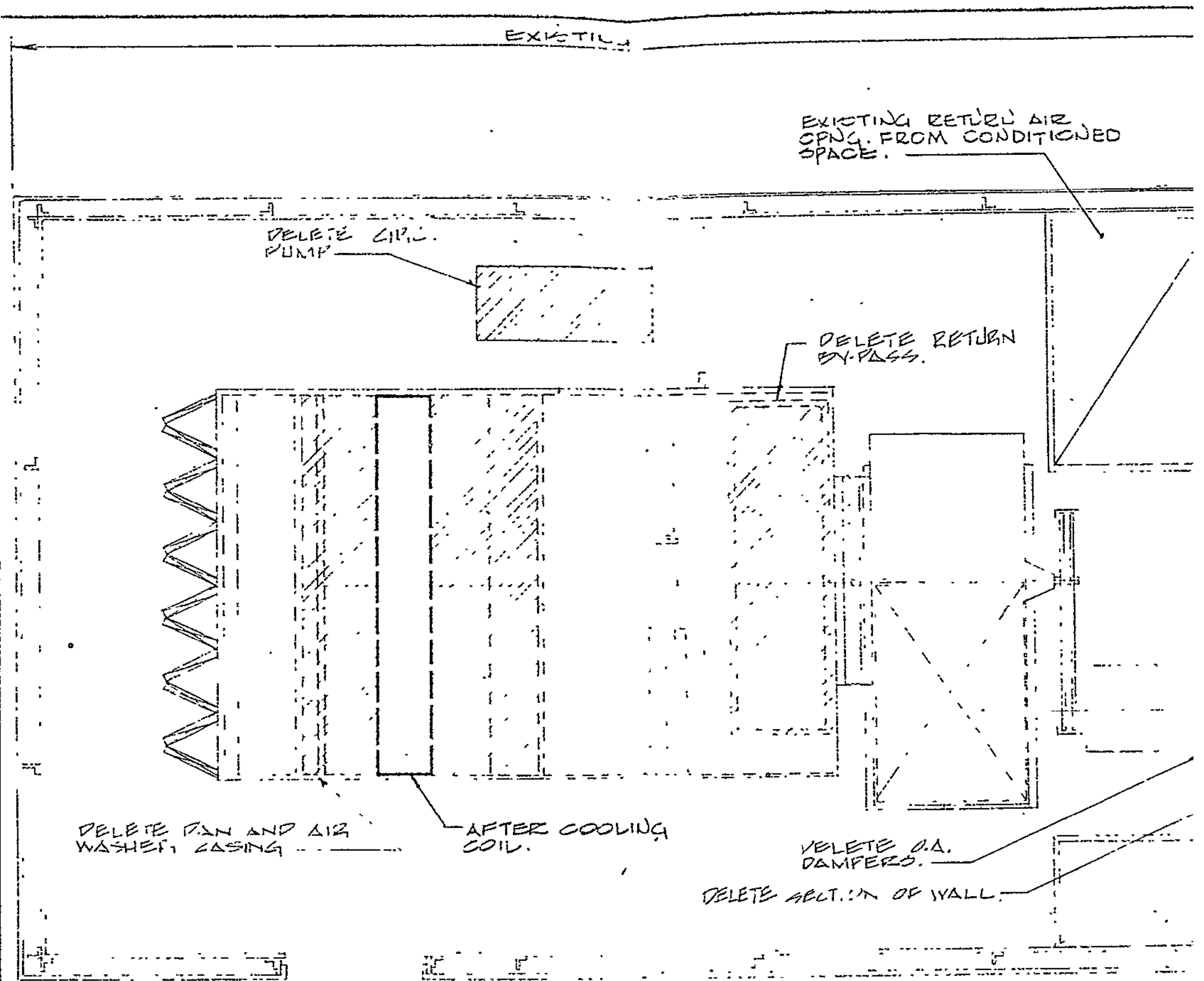

NEW
NEW RETURN AIR OPNG FROM CONDITIONED SPACE.
REGENERATOR
EXH. THROUGH ROOF 2900 CFM
F.A. LOUVER
1,700 CFM
HEAT EXCHANGER
PUMP
PUMP
PRE-COOLING COIL
M.D.
M.D.
CHEM. DEHUM.
F.A. LOUVER
6,000 CFM
SUPPLY AIR BYPASS OPEN TO M.E.R.

FIGURE 13



FAN HOUSE No. 22 MODIFICATION PLAN

- Reheat

  $(847 \frac{\text{lb steam}}{\text{hour}})$ (24 hours/day) (122 days/season) (0.4 duty cycle factor)

  =992,006 lb steam

b) Fall/Spring Season

- Refrigeration

  (123 tons) (16) (24) (122) (.4) = 2,304,922 lb steam

- Reheat

  (773)(24) (122) (.4) = 905,388 lb steam

c) Winter Season

- Refrigeration

  (107) (16) (24) (28) (48) (.4) = 788,890 lb steam

- Reheat

  (798) (24) (48) (.4) = 367,718 lb steam

d) Total steam consumption = 8,488,320 lb steam per year.

2. STEAM CONSUMPTION OF THE PROPOSED SYSTEM

a) Summer Season

(938 lbs for regeneration)+ 115.5 tons x 12 $\frac{\text{lb}}{\text{ton}}$ (24) $\frac{\text{hrs}}{\text{day}}$

(122) $\frac{\text{days}}{\text{season}}$ (.4) duty cycle factor = 2,721,869 lb of steam

b) Fall and Spring Season

(550 + 75.2 x 12) (24) (122) (.4) = 1,647,878 lb of steam

c) Winter Season

(369 + 56.3 x 12) (24) (47) (.4) = 481,536 lb of steam

d) Total steam consumption = 3,851,283 lb steam per year.

3. SAVINGS

Annual steam savings

= 8,488,320 - 3,851,283 = 3,637,037 lbs steam per year

4. ELECTRIC ENERGY USAGE

a) In each Fan House, the electrical energy consumed by fans and air washers pumps in the existing system is equal to the electrical energy consumed by fans and desiccant pumps in the proposed system. Therefore, these kilowatts are not included in the calculations.

b) Central plant electrical energy savings

- Cooling tower pumps - 191 KW
- Cooling tower fans - 227 KW
- Chilled water pumps - 134 KW

  Total 552 KW
- Total KWH saved = 552 x 8760 x .4 x .8 = 1,547,366 KWH

5. COOLING TOWER MAKE UP WATER COST

- Average refrigeration tonnage saved per hour (for all seasons)

$$\begin{pmatrix}\text{Existing system tonnage} \\ \text{average for all seasons}\end{pmatrix} - \begin{pmatrix}\text{Proposed system tonnage} \\ \text{average for all seasons}\end{pmatrix} =$$

$$= \frac{(167 + 123 + 107)}{3} - \frac{(115.5 + 75.2 + 56.3)}{3} = 50 \text{ tons}$$

- Average refrigeration tonnage saved per year per Fan House

  50 tons x 292 days x 24 hrs x .4 duty factor = 140,160 ton-hr
- Water saved per year per Fan House

  = 140,160 x 6.2 (evaporation rate factor for steam driven compressors)

  = 868,992 gallons.
- Water saved per year for the entire facility

  868,992 x 50.3* = 43,710,297 gallons.

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*50.3 is a scale factor and it is calculated as:

$$\frac{\text{Total CFM for Building \#103}}{\text{CFM for Fan House \#22}} = \frac{2,986,300}{59,400} = 50.3$$

6. CAPITAL INVESTMENTS

Chemical dehumidifier	$38,000.00
Remove washer and add after cooling coil	15,000.00
Add precooling coil	4,000.00
Controls	10,000.00
Fan house modifications	10,000.00
Electrical work	10,000.00
Chiller modification	4,970.00
	91,970.00
Miscellaneous items (10%)	9,970.00
Capital investment per fan house	$101,167.00
For the entire facility	50.3 x 101,167 = $5.088 x 10^6

7. PAY BACK ANALYSIS

a) Simple pay back (without escalation)

- Steam (Generated by natural gas)

$$\frac{3{,}637{,}037\ \frac{\text{lb}}{\text{year}} \times 1000\frac{\text{Btu}}{\text{lb}} \times 1\frac{\$}{\text{MCF}}}{1 \times 10^6\ \frac{\text{Btu}}{\text{MCF}} \times 0.8\ (\text{Efficiency})} = \$4{,}546 \text{ per Fan House}$$

For entire facility

\$1/MCF : 50.3 x 4,546 = \$228,663.

\$3.60/10^6Btu: 3.60 x 50.3 x 4,546 = \$823,186.

- Electricity

$552\ \text{KW} \times 8760\frac{\text{hrs}}{\text{year}} \times .4\ (\text{Duty cycle factor}) \times .8 \times .03\ \$/\text{KWH}$

$= \$46{,}421$

- Tower water make up

$$43{,}710{,}290\frac{\text{gallons}}{\text{year}} \times 1.5\ \frac{\$}{\text{Gallons}} = \$65{,}565$$

- Pay Back

Total savings for entire facility

\$1/MCF: 228,663. + 46,421. + 65,565 = \$340,649

\$3.60/10^6Btu: 823,186 + 46,421 + 65,565 = \$935,172

The payback period for \$1/MCF: $\frac{5.088 \times 10^6}{340,649} = 14.9$ years

The payback period for \$3.60/10^6Btu = $\frac{5.088 \times 10^6}{935,172} = 5.4$ years

b) Simple payback (with escalation)

When the above calculations are repeared with a 10% increase in fuel costs, the following results are obtained:

Payback period for \$1/MCF: 10 years
Payback period for \$3.60.10^6Btu: 4.5 years

The results of this analysis are summarized in Tables 1 and 2.

A discussion on the major parameters that effect the applicability of Chemical Dehumidification System for comfort cooling is given in the Appendix.

TABLE 1

SIMPLE PAYBACK WITH ESCALATION

Gas	$3.60/$10^6$ Btu
Electricity	$.03 KWH
Water	$1.50/1000 gallons

TRANS PERIOD	ELECTRIC RATE* ($)	ENERGY SAVED (KW)	COST OF SAVED ENERGY ($)	WATER** RATE ($)	WATER SAVED (Gallons)	COST OF WATER SAVED ($)	GAS*** RATE ($)	GAS SAVED (10^6cu ft)	COST OF GAS SAVED ($)	TOTAL SAVINGS ($)	CUMULATIVE TOTAL SAVINGS ($)
FY 1977	.0300	1548431	$46452	$1.50	43710297	65565	3.60	228663	823186	935203	935203
FY 1978	.0330	1548431	51098	1.65	43710297	72121	3.96	228663	905504	1028723	1963926
FY 1979	.0363	1548431	56208	1.81	43710297	79333	4.35	228663	996055	1131596	3095522
FY 1980	.0399	1548431	61782	1.99	43710297	87267	4.79	228663	1095660	1244709	4340231
FY 1981	.0439	1548431	67970	2.19	43710297	95993	5.27	228663	1205226	1369189	5709420

The pay back period is 4 1/2 years

* per KW

** per 1000 Gallons

*** per 10^6Btu

TABLE 2

SIMPLE PAYBACK WITH ESCALATION

Gas	$1/MCF
Electricity	$.03 KWH
Water	$1.50/1000 gallon

TRANS PERIOD	ELECTRIC RATE* ($)	ENERGY SAVED (KW)	COST OF SAVED ENERGY ($)	WATER** RATE ($)	WATER SAVED (Gallons)	COST OF WATER SAVED ($)	GAS*** RATE ($)	GAS SAVED (10^6 cu ft)	COST OF GAS SAVED ($)	TOTAL SAVINGS ($)	CUMULATIVE TOTAL SAVINGS ($)
FY 1977	.03	1548431	46452	1.50	43710297	66556	1.0	228663	228663	340,680	340,680
FY 1978	.033	1548431	51098	1.65	43710297	72121	1.1	228663	251529	374,748	715,428
FY 1979	.0363	1548431	56208	1.81	43710297	79333	1.21	228663	276682	412,223	1,127,651
FY 1980	.0399	1548431	61782	1.99	43710297	87267	1.331	228663	304350	453,399	1,581,050
FY 1981	.0439	1548431	67976	2.19	43710297	95993	1.464	228663	334785	498,754	2,079,804
FY 1982	.0482	1548431	74634	2.40	43710297	105593	1.610	228663	368147	548,374	2,628,178
FY 1983	.0530	1548431	82066	2.64	43710297	116152	1.771	228663	404961	603,179	3,231,357
FY 1984	.0583	1548431	92073	2.90	43710297	127767	1.948	228663	445457	665,297	3,896,654
FY 1985	.0641	1548431	99300	3.19	43710297	140543	2.142	228663	490003	729,846	4,626,500
FY 1986	.07051	1548431	109230	3.51	43710297	153423	2.35	228663	539004	801,657	5,428,157
FY 1987	.0775	1548431	120153	3.86	43710297	168721	2.59	228663	592094	881,778	6,309,935

Pay Back period is 10 years

* per KW
** per 1000 Gallons
*** per 1000 cu. ft.

APPENDIX

APPLICABILITY OF CHEMICAL DEHUMIDIFICATION SYSTEM FOR COMFORT COOLING

In problems involving the cooling and dehumidification of air, it is desirable that the cooling process curve and sensible heat ratio line intersect as at point 1 of Figure A1. In such case, no reheating will be necessary and an air supply in the state represented by the point of intersection will maintain the desired condition in occupied space. However, there are cases in which the sensible heat ratio line will not intersect any cooling process curve that can be obtained with equipment that simultaneously cools and dehumidifies air, but does not reheat it. Two such sensible heat ratio lines are illustrated in Figure A1. The sensible heat ratio line 1-2 is so steep that it cannot intersect the condition curve. Although sensible heat ratio line 8-9 is not steep, it is drawn on a portion of the chart where it cannot intersect any condition curve that can be obtained with the usual equipment used to simultaneously cool and dehumidify air. Hence, for such sensible heat ratio lines, the air must be reheated after first being cooled and dehumidified. In order to obtain air in a state such as 1 on the sensible heat ratio line 1-2, the air after first being cooled and dehumidified from 4 to 5 must be reheated from 5 to 1.

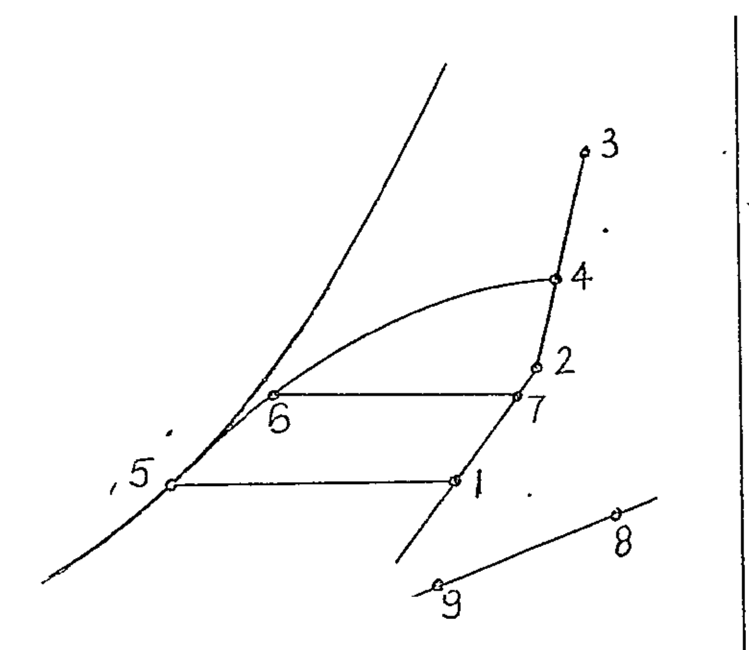

Figure A1. Psychrometric Chart showing different Sensible Heat Ratio Lines.

Another distinction between reheating for temperature control and reheating for a steep sensible heat ratio line is in the volume of air required. When reheating for a steep sensible heat ratio line, a greater volume of conditioned air must be supplied in order to absorb the sensible heat added in reheating the air supply at a time when the internal heat gain of the room is at its peak. On the other hand, when reheating for temperature control, no additional air volume above the amount needed for the maximum internal heat gain of the room need be provided.

The necessity for additional refrigerating capacity and an additional chilled air supply is the essential difference between reheating for a steep sensible heat ratio line and reheating for temperature control.

The reheating of the air can be eliminated to achieve comfort conditions for steep sensible ratio lines by using a CHEMICAL DEHUMIDIFICATION SYSTEM (CDS).

As the results of this report illustrate, a chemical dehumidification system can be utilized for comfort cooling under certain circumstances. In these cases, it results in significant reductions in overall system operating costs. Its presence also allows for the use of a smaller refrigeration plant. The following general statements may be made about the advantage of its application.

(a) On new construction, where the cost of the CDS can be partially defrayed by a reduction in air chilling equipment costs, it should be considered where the desired space condition is approximately 80°F, and 50% RH or lower, in combination with a space coding load sensible heat ratio of 0.7 or lower.

(b) On existing facilities, where the cost of the CDS cannot be defrayed by reducing the cost of auxiliary air chilling equipment, it should be considered where the desired space condition is approximately 75°F, and 50% RH or lower, in combination with a space cooling load sensible heat ratio of 0.7 or lower.

(c) In either (a) or (b), the overall applicability of CDS may be a function of the outdoor weather pattern. For instance, if the building design is such that provisions for the use of 100% outdoor air during the off-season is economical (one story buildings with roof top package units), and the weather is normally cool and dry, the operating cost offered by CDS is substantially reduced.

The psychrometric charts given in Figures A2 and A3, illustrate the process corresponding to (a) and (b), respectively. The chart given in Figure A4 illustrates the process required on the Michoud building #103, with the sensible heat ratio at 0.62. The chart given in Figure A5 illustrates the processes required to maintain 75°F, and 50% RH at various SHR's of 0.6 - 0.9. Examining the chart in Figure A5, it becomes clear that as the SHR goes from 0.9 to 0.6, (assuming a minimum practical apparatus dew point of 50°F) somewhere between a SHR of 0.7 and 0.8. By the time 0.7 is reached, 9°F of sub-cooling and reheat are required. This increases to 14°F at a SHR of 0.6

Figure A2 shows that at 80°F, 50% RH space conditions, and a SHR or 0.7, the use of CDS would be marginal, since a 52°F apparatus dew point could handle the load. In some instances however, such as in the case of office buildings, where it is desirable to minimize the amount of air supply to the space, so as to minimize overall construction costs, it is possible to handle the entire space latent load by using CDS to subdehumidify only the minimum fresh air make-up. The CDS can modulate directly with the internal latent load. Although each application will have additional variables such as duty cycle, fuel costs, indoor and outdoor design conditions, the slope of the sensible heat ratio line, the variation of the sensible heat ratio throughout all partial load conditions and the final occupied space conditions appear to be the critical parameters in economically utilizing Chemical Dehumidification Systems.

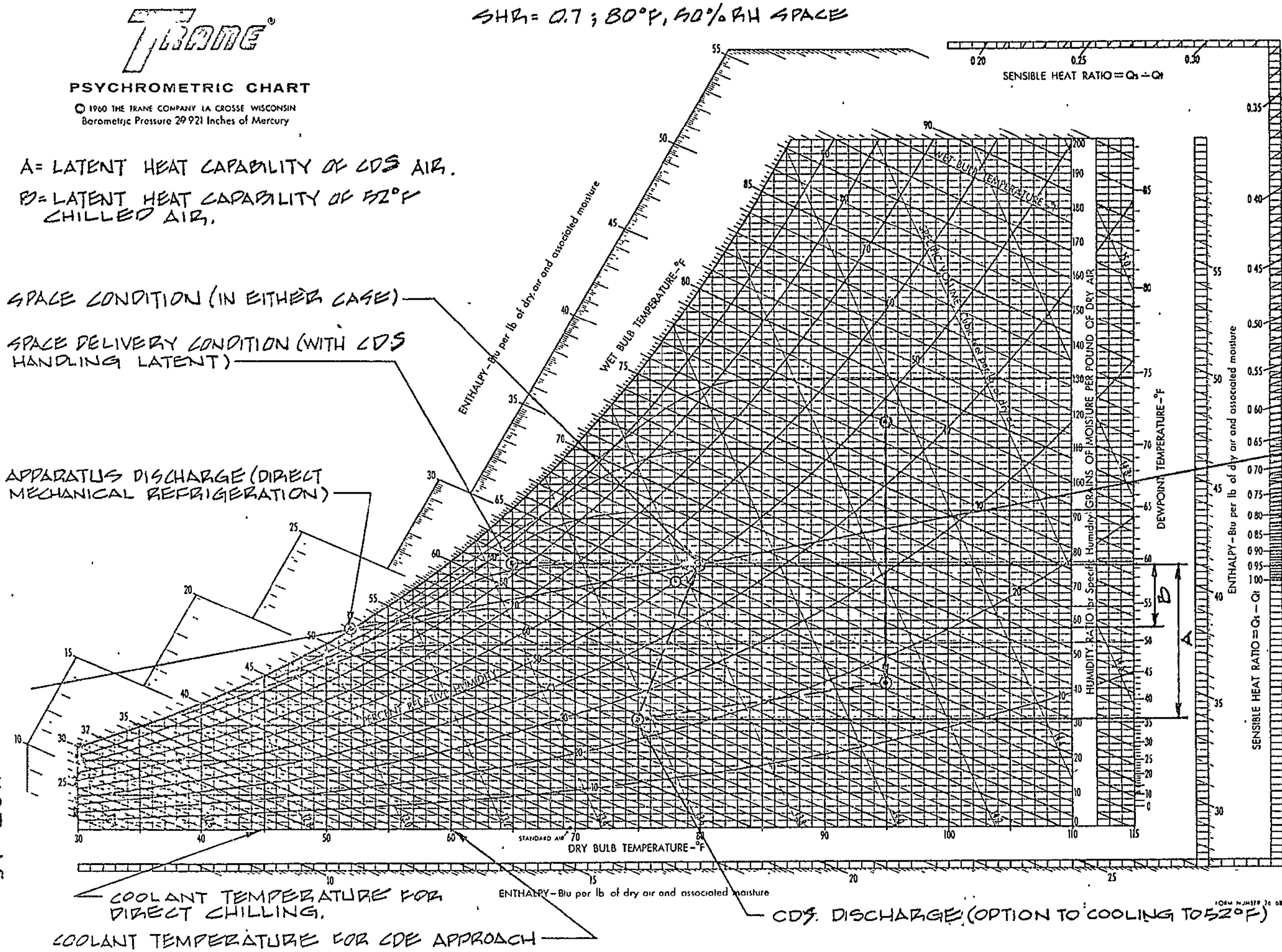
TRANE
PSYCHROMETRIC CHART
© 1960 THE TRANE COMPANY LA CROSSE WISCONSIN
Barometric Pressure 29.921 Inches of Mercury
SHR= 0.7 ; 80°F, 50% RH SPACE
A= LATENT HEAT CAPABILITY OF CDS AIR.
B= LATENT HEAT CAPABILITY OF 52°F CHILLED AIR.
SPACE CONDITION (IN EITHER CASE)
SPACE DELIVERY CONDITION (WITH CDS HANDLING LATENT)
APPARATUS DISCHARGE (DIRECT MECHANICAL REFRIGERATION)
COOLANT TEMPERATURE FOR DIRECT CHILLING.
COOLANT TEMPERATURE FOR CDE APPROACH
CDS. DISCHARGE (OPTION TO COOLING TO 52°F)
SENSIBLE HEAT RATIO = Qs ÷ Qt
ENTHALPY – Btu per lb of dry air and associated moisture
WET BULB TEMPERATURE – °F
DRY BULB TEMPERATURE – °F
DEWPOINT TEMPERATURE – °F
HUMIDITY RATIO (lbs or Specific Humidity) GRAINS OF MOISTURE PER POUND OF DRY AIR
PERCENT RELATIVE HUMIDITY
SPECIFIC VOLUME (cubic feet per lb of dry air)
STANDARD AIR
A
B

FIGURE A2.

A5

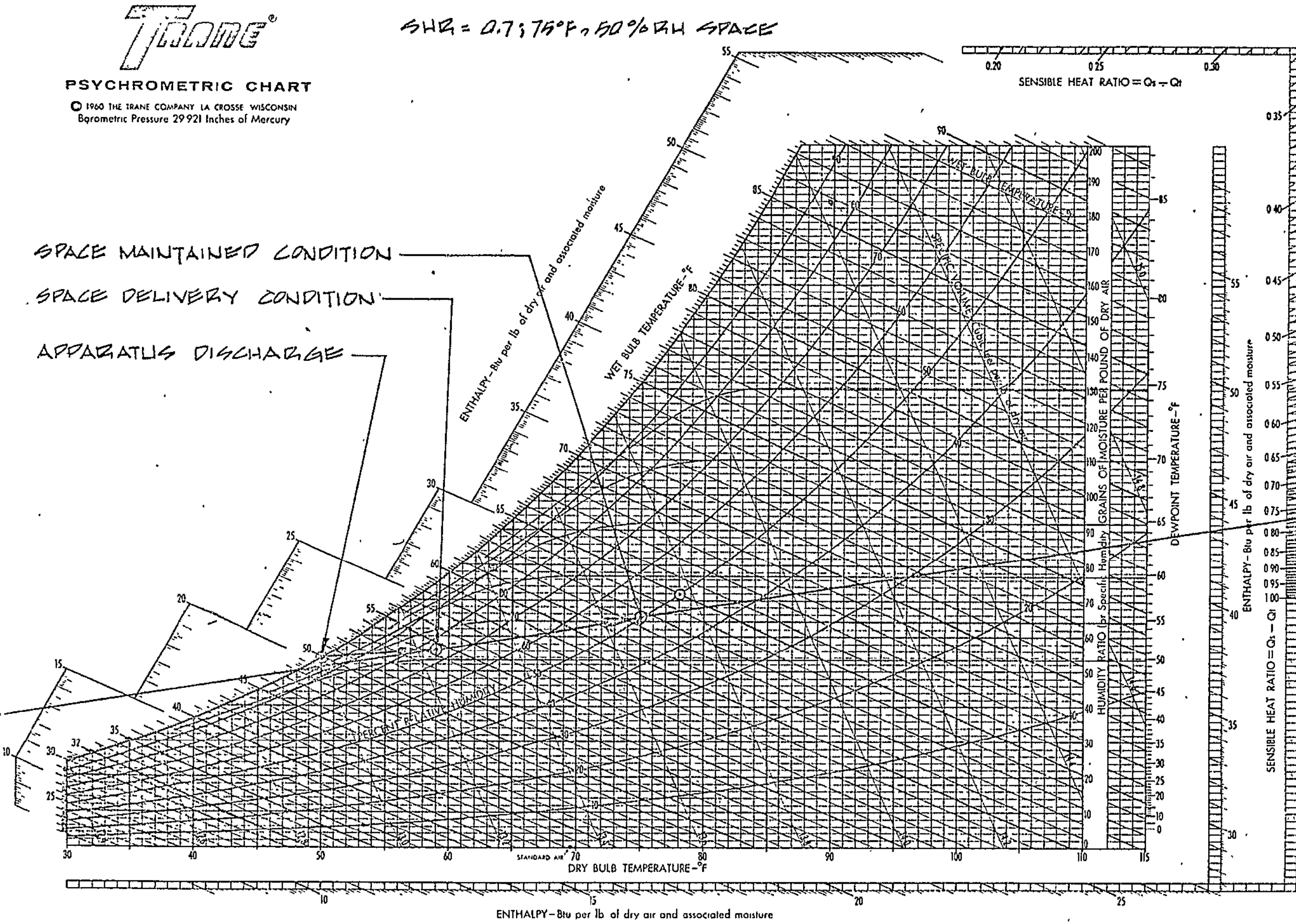
TRANE
PSYCHROMETRIC CHART
© 1960 THE TRANE COMPANY LA CROSSE WISCONSIN
Barometric Pressure 29.921 Inches of Mercury
SHR = 0.7; 75°F, 50% RH SPACE
SPACE MAINTAINED CONDITION
SPACE DELIVERY CONDITION
APPARATUS DISCHARGE
SENSIBLE HEAT RATIO = Qs ÷ Qt
ENTHALPY - Btu per lb of dry air and associated moisture
WET BULB TEMPERATURE - °F
PERCENT RELATIVE HUMIDITY
HUMIDITY RATIO (or Specific Humidity) GRAINS OF MOISTURE PER POUND OF DRY AIR
DEWPOINT TEMPERATURE - °F
DRY BULB TEMPERATURE - °F
STANDARD AIR
SENSIBLE HEAT RATIO = Qs ÷ Qt
ENTHALPY - Btu per lb of dry air and associated moisture

FIGURE A3.

SHR = 0.62; 75°F; 50% RH, AS ON MICHOUD.

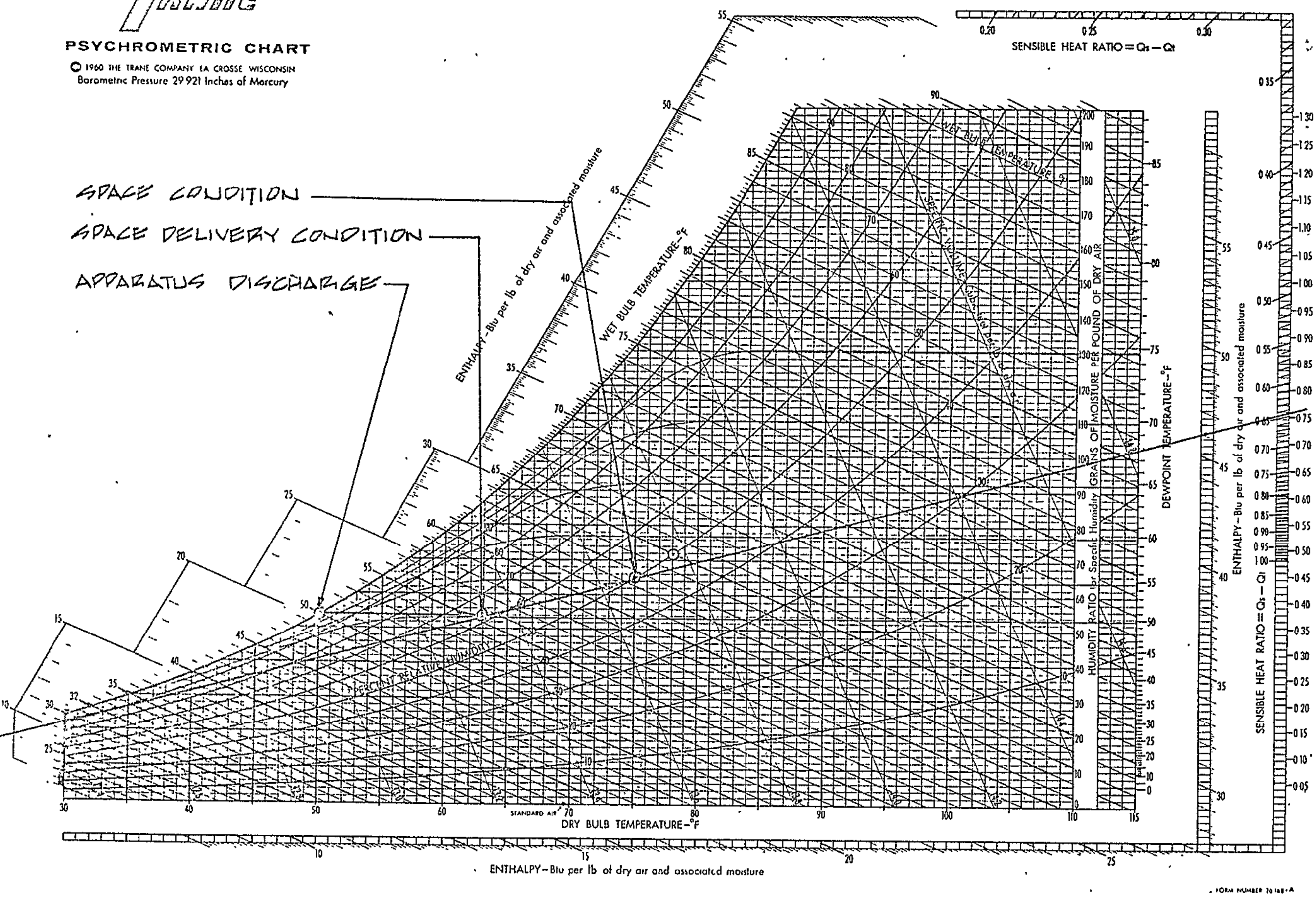

FIGURE A4

A7

EFFECT OF SHR ON PSYCHROMETRIC PROCESS