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USE OF FLUIDIZED BED HEAT EXCHANGERS IN HEAT PUMP SYSTEMS FOR IMPROVED PERFORMANCE

Technical Status Report
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NOTICE

P R O G R E S S R E P O R T

ABSTRACT

Performance curves have been obtained for condenser heat output, evaporator heat input and overall coefficient of performance (COP) as a function of outdoor air temperatures ranging from 10°F to 50°F and at an indoor air temperature of 65°F. Both temperatures were controlled by recycling cold air from the evaporator exhaust and mixing it in the plenum. There is good agreement, within expected error bands, with available performance data for the York heat pump. In addition, this data has been used to obtain air side heat transfer coefficients ranging from about 9 to 6 BTU/hr°F for the evaporator and 1.5 to 2.5 BTU/hr°F for the condenser. These values are being used as a basis for the design of the fluidized bed heat exchanger.

Plans for a separate test of the deicing capability of low temperature fluidized beds have been formulated.

1. SYSTEM OPERATION

Tests have been run to determine the variation in COP and heat output with outside (into the evaporator) air temperature and a constant indoor (into the condenser) air temperature of 65°F. The results of the tests are shown in Figures 1 and 2. Within the error bands shown, there is approximate agreement with data published by York. These errors were computed on the basis of the estimated accuracies listed below.

Estimated Accuracies

Temperature	∓ 0.5°F
Air flow rate	± 50 CFM (Hot)
	± 200 CFM (Cold)
Freon flow rate	± 0.05 gpm
Pressures	± 5%
Current	± 3%
Voltage	208 ∓ 2 volts

Other possible sources of error are heat losses due to air moisture content, which were not accounted for, and the effect of auxiliary fans to help push the air through the plenum. A means of controlling, or at least monitoring, humidity in the air streams, which are being dried out by the recirculation procedure, is needed. Other errors can be reduced as further experience is gained with the system and the automated data recording system is put into operation. Freon flow rate is still doubtful since it is measured at the only point in the circuit where the refrigerant should be totally in the liquid phase; that is, after it leaves the condenser. Temperatures and pressures at this point show the refrigerant to be right on the saturated liquid line in most cases so that within measurement errors it is difficult to know whether there is vapor in the line producing errors in the turbine flow meter readings.

2. COMPUTER MODEL

Programming of the model of refrigerant flow through the expansion coil is about 50 percent done. The overall model still needs further refinement before a comparison with actual data can be made.

3. FLUIDIZED BED HEAT EXCHANGER

From the system test data, heat transfer coefficients for the evaporator and condenser coils have been calculated for use in designing the fluidized bed heat exchangers. These are given in the table below.

"Outdoor Air" °F	Evaporator				Condenser			
	ΔT (LMTD)	$Q \frac{BTU}{HR}$	$\bar{U}A = \frac{Q}{\Delta T} \frac{BTU}{HR°F}$	$h = \frac{BTU}{HR°F \cdot ft^2}$	ΔT (LMTD)	$Q \frac{BTU}{HR}$	$\bar{U}A = \frac{Q}{\Delta T} \frac{BTU}{HR°F}$	$h = \frac{BTU}{HR°F \cdot ft^2}$
10	3.1	25,800	8300	8.8	44	41,300	940	1.4
15	3.0	25,800	8600	9.1	46	41,300	900	1.3
20	4.0	30,100	7500	7.9	49	49,600	1000	1.5
25	5.6	33,000	5900	6.3	49	50,500	1000	1.5
30	5.9	33,500	5700	6.1	53	59,100	920	1.3
35	6.2	36,500	5900	6.3	50	60,000	1200	1.8
40	7.3	40,400	5500	5.8	48	64,400	1300	1.9
45	8.1	51,300	6300	6.7	47	68,200	1450	2.1
50	8.1	56,000	6900	7.3	45	77,700	1700	2.5

$$A(EVAP) = 941\ ft^2$$
$$A(COND) = 682\ ft^2$$

The design tries to reproduce the same (air side) heat transfer coefficients using the enhanced heat transfer capability of the fluidized bed. This should result in a heat exchanger with a net heat transfer area that is considerably less than the existing heat exchangers, with no decrease in performance.

4. THE ICING PROBLEM

Heat pump outdoor heat exchangers, because they have to work at below freezing temperatures, accumulate frost which impairs heat transfer capability. Most commercial systems reverse the cycle so that the heat pump is an air conditioner for part of its duty cycle which decreases its efficiency as a heater. There is a possibility that icing is decreased in fluidized bed heat exchangers. A simple test, using dry ice-cooled ethylene glycol as the working fluid, is planned to investigate this possibility. This will be separate from the main test facility.

5. PERSONNEL

For the period covered by this report — September 1, 1978 to November 30, 1978 — Professor John C. Chen participated in the project at a rate of 10 percent and Professor Robert G. Sarubbi at 15 percent.

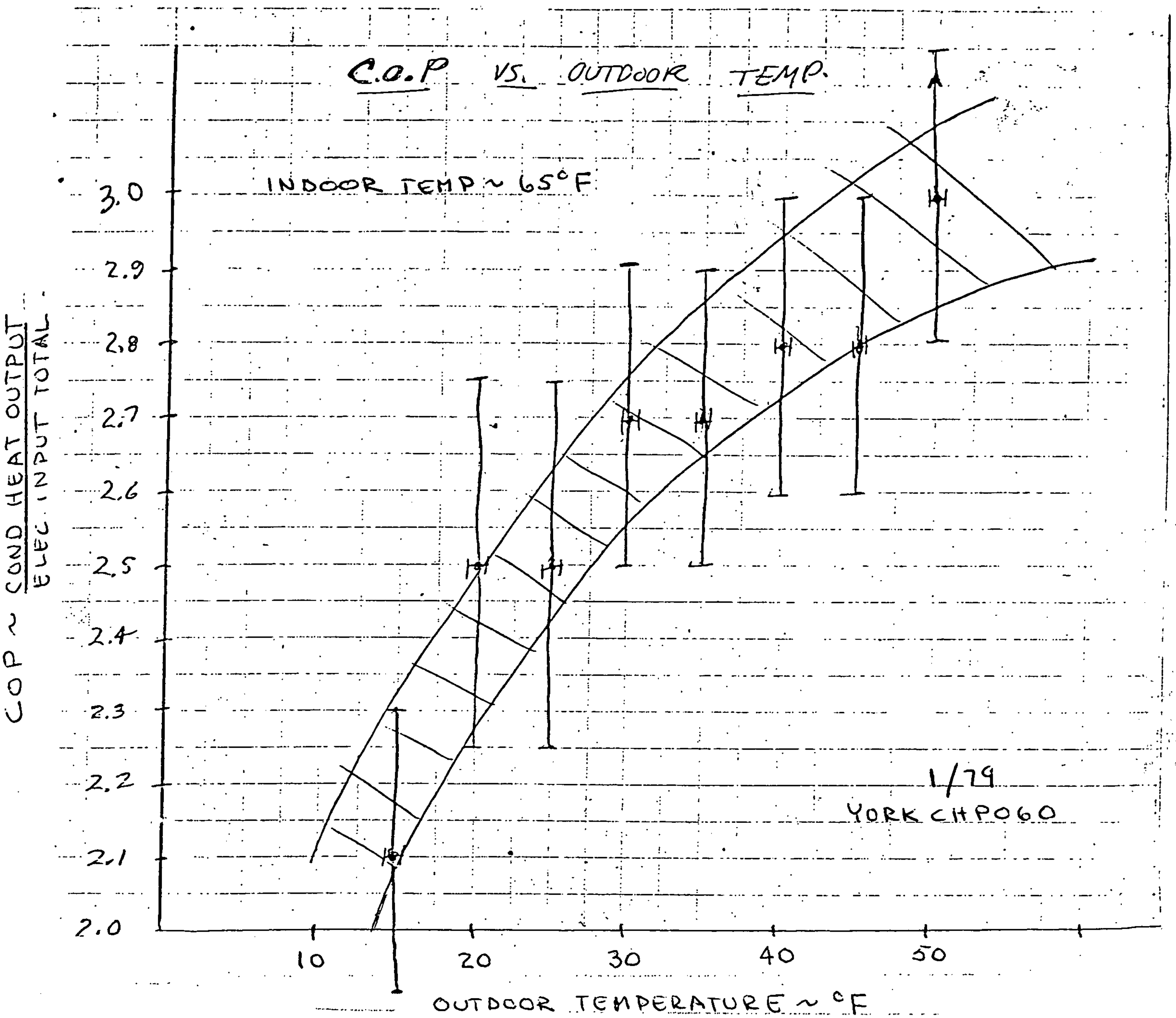
C.O.P VS. OUTDOOR TEMP.
INDOOR TEMP ~ 65°F
COP ~ COND HEAT OUTPUT / ELEC INPUT TOTAL
3.0
2.9
2.8
2.7
2.6
2.5
2.4
2.3
2.2
2.1
2.0
10
20
30
40
50
OUTDOOR TEMPERATURE ~ °F
1/79
YORK CHP060

FIGURE 1

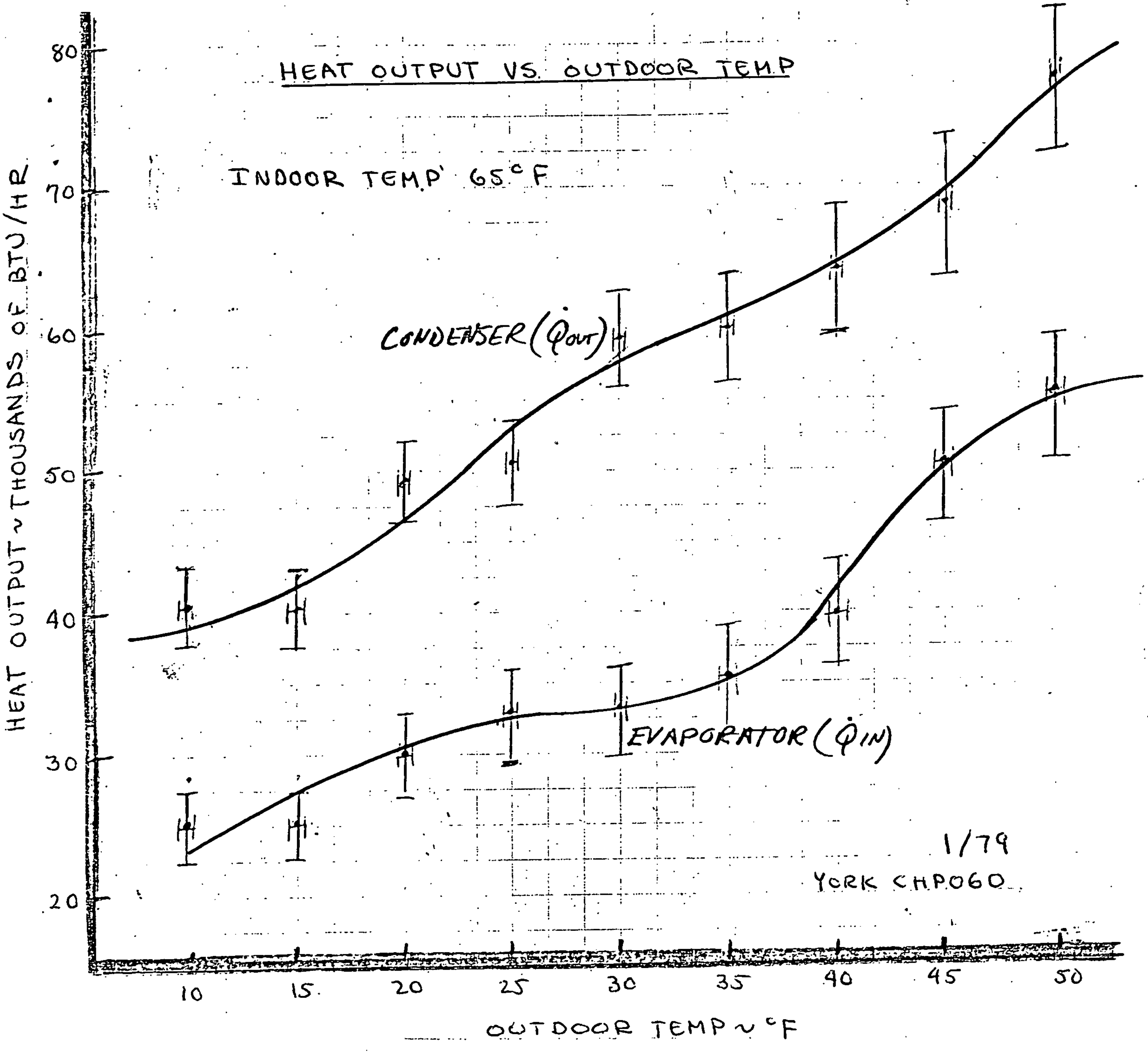
HEAT OUTPUT VS. OUTDOOR TEMP
INDOOR TEMP 65°F
CONDENSER ($\dot{Q}_{OUT}$)
EVAPORATOR ($\dot{Q}_{IN}$)
1/79
YORK CHP060
HEAT OUTPUT ~ THOUSANDS OF BTU/HR
OUTDOOR TEMP ~ °F
80
70
60
50
40
30
20
10
15
20
25
30
35
40
45
50

FIGURE 2