Analysis of a Refrigerant Compressor Load Stand Incorporating Hot Gas Bypass and a Single Full Condensation Heat Exchanger

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ANALYSIS OF A REFRIGERANT COMPRESSOR LOAD STAND INCORPORATING HOT GAS BYPASS AND A SINGLE FULL CONDENSATION HEAT EXCHANGER

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ABSTRACT

A refrigerant compressor load stand is described and a theoretical analysis of its operation is presented. The purpose of the load stand is to establish and maintain pre-determined values of the suction pressure, the discharge pressure and the suction superheat. These are controlled by means of throttle valves. The power of compression is rejected to cooling water in a full condensation heat exchanger which normally operates close to ambient temperature. The refrigerant flow is divided and the smaller part, generally less than one third, passes through the heat exchanger, while the greater part bypasses it as vapour. The two streams are re-combined by adiabatic mixing. Brief details are given of such a load stand which has been built recently, together with initial operating experiences.

INTRODUCTION

A research programme with the title ´Capacity Control of Reciprocating Piston Compressors for Heat Pump or Refrigeration Applications by Speed Variation´ has been underway at Trinity College since the Summer of 1983. The purpose of this work is to study the performance of reciprocating compressors when operated at different shaft speeds. As the programme will involve a great deal of performance testing of compressors over a wide operating range of each of four main parameters, the initial phase involves the construction and development of a suitable load stand. The four main parameters to be controlled are:

- mean suction pressure
- mean discharge pressure
- suction superheat temperature
- compressor shaft speed.

The first three are parameters of the load stand, while the fourth directly influences its capacity. In addition to the controlled parameters it is intended to measure the following:

- compressor shaft speed
- compressor shaft torque (reaction torque)
- refrigerant mass flow rate (by means of an orifice plate)
- discharge vapour temperature.

It is also intended to obtain pressure-volume diagrams and valve lift measurements for a full range of operating conditions.

A load stand is proposed in which the three main cycle parameters are controlled by means of throttle valves and in which part of the refrigerant is fully condensed to the liquid state. Such a system offers several potential advantages. As some excess liquid is always present it is not necessary to vary the system charge when the operating conditions are changed. If the temperature of the condensing heat exchanger is maintained constant its thermal inertia and that of its contents should be of small consequence and a rapid system response to changes in the operating conditions should be possible. Automatic control of the throttle valves should be feasible. The load stand is a variation on those described in references (1) and (2).

In some respects the proposed system is analogous to a hydraulic dynamometer used in engine testing. The heat equivalent of the shaft power is rejected to the cooling water and the conversion from work to heat is by means of totally irreversible processes of fluid friction.

A number of computer programs and subroutines have been written in FORTRAN for the analyses which follow. The subroutines listed in reference (3) were used for the refrigerant property calculations.

THERMODYNAMIC PROCESSES

A circuit diagram to implement the basic processes of the load stand is shown in Fig. 1. It is assumed that condensation to the saturated liquid state occurs in the heat exchanger and that the saturation pressure can be maintained constant. The suction and discharge pressures of the compressor are established with respect to the heat exchanger pressure by the throttle valves D and A. By some, as yet unspecified means, part of the total refrigerant flow leaving valve A is caused to pass through the heat exchanger, while the balance is bypassed to valve D.

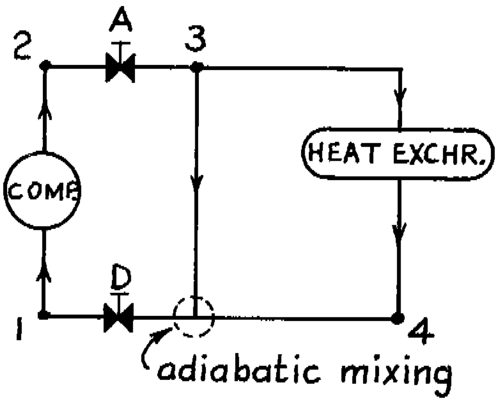

Figure 1. Load stand basic circuit

By varying the proportion of refrigerant which is bypassed the suction superheat may be controlled. The idealised thermodynamic processes are shown on a P-h diagram in Fig. 2.

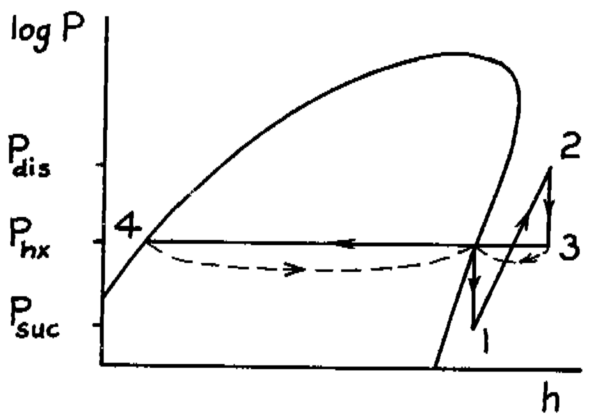

Figure 2. Idealised thermodynamic processes

By applying the Steady Flow Energy Equation the flow rate of refrigerant through the heat exchanger can be expressed as a proportion of the total as followsß

$$\frac{\dot{m}_{hx}}{\dot{m}_{tot}} = 1 - \frac{h_1 - h_4}{h_2 - h_4} \qquad (1)$$

Table 1 is a set of results of this calculation for a wide range of operating conditions for refrigerant 12 when the saturation temperature in the heat exchanger is 20 deg. C. Similar tables have been produced for other values of the heat exchanger operating temperature. For example, at a saturation temperature of 30 deg. C the proportions of flow rate which pass through the heat exchanger are higher by from 5% to 7%, depending on the pressure and superheat conditions.

The main conclusion from Table 1 is that it is possible, in principle, to produce any operating condition of interest by varying the relative amounts of refrigerant which are condensed and bypassed.

There are practical difficulties in implementing the cycle as described thus far. There will be a pressure fall, however small, in any real heat exchanger and so a corresponding pressure fall will be required in the bypass line. These pressure losses will not upset the operation of the cycle. A means of proportioning the flow is also necessary. This can be done by introducing throttle valves B and C as shown in Fig. 3.

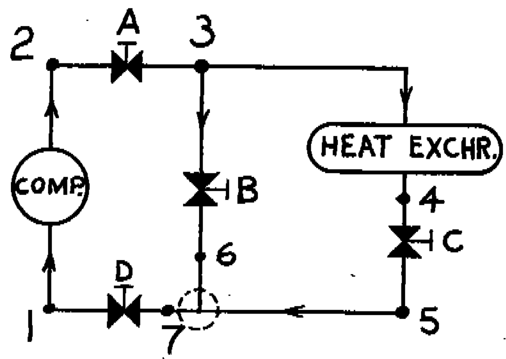

Figure 3. Circuit including bypass and liquid line throttle valves

The effect these valves have on the thermodynamic cycle is shown on a P-h diagram in Fig. 4. The inclusion of these extra throttle valves does not alter the values presented in Table 1 for the proportion of the refrigerant flow which passes through the heat exchanger. The pressure at which adiabatic mixing of the condensed liquid and bypassed vapour occurs may lie at any level between that of the heat exchanger and that at suction, depending on the settings of valves B, C and D.

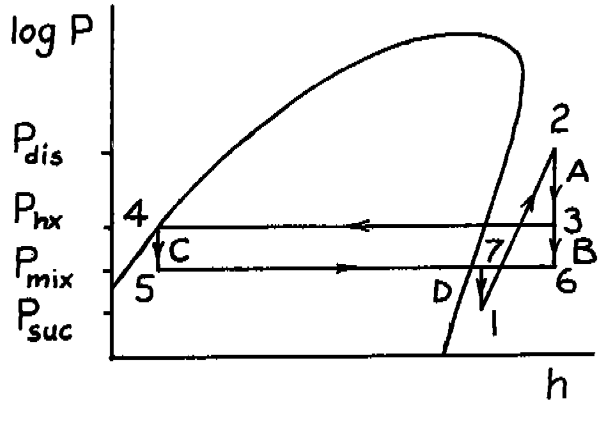

Figure 4. Thermodynamic processes, including liquid line and bypass throttling.

The Ideal Load Stand and Compressor

For the purposes of designing or sizing a load stand an ideal compressor which can be described in terms of its displacement rate and clearance ratio is a suitable model. The volumetric efficiency of such a compressor is given by

$$\eta = 1 - r_c \left[\frac{v_1}{v_2} - 1 \right] \quad (2)$$

By using this equation in conjunction with the displacement rate and the thermodynamic relationships for the cycle, the ideal mass flow rate, the proportions of refrigerant condensed and bypassed, and the rate of heat rejection in the heat exchanger can be calculated for any combination of operating conditions. The results of such calculations, assuming a displacement rate of 1 litre/sec., a clearance ratio of 3% and a heat exchanger saturation temperature of 20 deg. C are given in Tables 2 and 3. The values of heat rejection rate, given in Table 3, apply regardless of the heat exchanger operating temperature. These figures can be used in selecting a suitable heat exchanger for the load stand.

Effective Valve Flow Areas

The value chosen for the mixing pressure is a matter of control strategy and, when this has been decided upon, the required flow areas for the four throttle valves can be calculated. The valves are modeled as orifices having a coefficient of discharge of 1 and can thus be fully described by an effective flow area. For valve C, which the refrigerant enters as a liquid, incompressible flow is assumed and the effective flow area is given by

$$A = \dot{m} \sqrt{\frac{v_u}{2 \;\Delta P}} \quad (3)$$

For valves A, B AND D the flow is compressible and, depending on the pressure ratio in each case, may be choked. In the calculations either the actual or the critical pressure ratio, whichever is the greater, is used. For the purposes of calculating the required orifice area isentropic flow is assumed, although, for the throttling process as a whole, enthalpy is the property which remains unchanged. The isentropic process is modeled as being polytropic, i.e. one for which

$$P\, v^n = \text{const.} \quad (4)$$

While this assumption may be considered reasonably valid for refrigerant vapour, even within the saturation region, the ideal gas equation should not be assumed to apply, especially when the refrigerant is saturated for part of the process. The equation for the required flow area should therefore be in terms of the properties pressure and volume, rather than, for example, pressure and temperature. The following equation is used:

$$A = \dot{m} \left[\frac{P_u}{v_u} \frac{2n}{n-1} \left(r_p^{\frac{2}{n}} - r_p^{\frac{n+1}{n}} \right) \right]^{-.5} \quad (5)$$

It should be noted that for some suction conditions point 7 in Fig. 4 lies within the saturation region on the P-h diagram. The specific volume at this point can be found since the specific enthalpy is the same as at point 1.

Values of the required effective flow areas for the valves are presented over a range of operating conditions in Table 4. It has been assumed in the table that one fifth of the pressure fall from the heat exchanger saturation pressure to the suction pressure occurs across the bypass and liquid line throttle valves B and C.

It can be noted that any change in the operating conditions requires a change in the areas of all four valves. The required flow areas for valve C, the liquid line throttle valve, are very much smaller than for the other valves which throttle vapour.

HEAT EXCHANGER SELECTION, CHARACTERISTICS AND ANALYSIS

The displacement rate of the compressor to be tested was about 3.7 litres/second and, using data from Table 3, the required maximum heat rejection rate was found to be about 2.2 kW, which also coincided with the motor rating.

A water cooled condenser heat exchanger with some capacity to hold excess liquid was desired and a shell and tube type was selected. Either a single pass or a two pass water piping arrangement was possible. The single pass arrangement was chosen in order to maximise the flow rate and to minimise the temperature change of the water in passing through the heat exchanger.

Heat Transfer Rate

The manufacturer presented the heat transfer rate as a function of the water flow rate and the temperature difference defined in terms of the saturation temperature and the water inlet temperature.

$$\Delta T = T_s - T_{wi} \tag{6}$$

Heat transfer correlations suggest that the water side heat transfer coefficient depends on the flow rate, raised to the power of 0.8 approximately. Hence, the following expression was derived for the rate of heat transfer

$$Q = \left[\frac{a\,\dot{V}^{.8}}{1 + b\,\dot{V}^{.8}}\right]\Delta T \tag{7}$$

By regression analysis of points taken from the manufacturer's graphs the following coefficients were obtained

$$a = 2.009 \tag{8}$$
$$b = 1.933 \tag{9}$$

A graph of this expression is given in Fig. 5.

Log Mean Temperature Difference

Care should be taken in applying equations 6 to 9 for flow rates lower than those quoted in the manufacturer's data (0.19 litres/sec.), as no implicit account is taken of the water exit temperature.

The temperature differences were converted to log mean values based on the refrigerant saturation temperature and the water inlet and outlet temperatures using the following equation

$$\Delta T_{\substack{log\\mean}} = \frac{(T_s - T_{wo}) - (T_s - T_{wi})}{\ln\left(\frac{T_s - T_{wo}}{T_s - T_{wi}}\right)} \tag{10}$$

The constants given in equations 8 and 9 were re-evaluated for the case where the log mean temperature difference is used in equation 7. The values are

$$a = 3.471 \tag{11}$$
$$b = 3.735 \tag{12}$$

A similar fit to that shown in Fig. 5 was obtained using these constants.

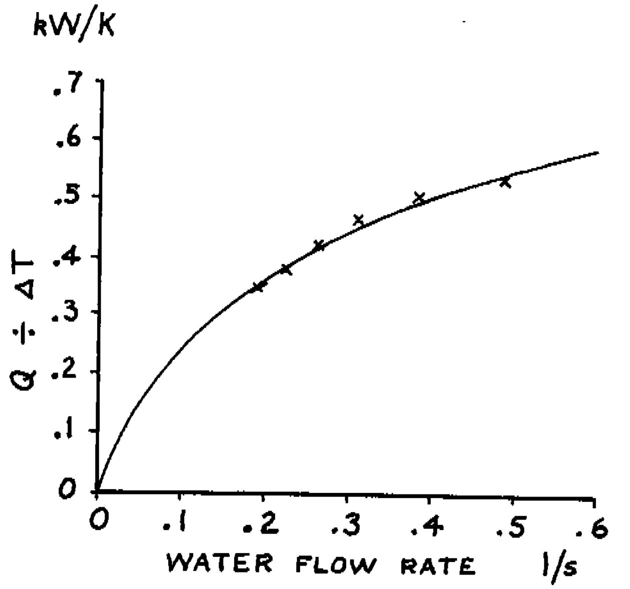

Figure 5. Heat transfer rate per degree temperature difference as a function of water flow rate.

Water Pressure Loss

It was found that the manufacturer's data could be represented reasonably closely by the following expression:

$$\Delta P = k\,\dot{V}^{j} \tag{13}$$

where $k = 1.151 \times 10^5$ (14)
and $j = 1.833$ (15)

WATER CIRCUIT ANALYSIS

The maximum available water supply rate was about 4 litres per minute and the maximum head between a constant head tank installed in the laboratory and the drain was about 1.7 metres. The water circuit which was adopted is shown diagramatically in Figure 6. Valve E controls the rate of water flow to drain and hence the saturation temperature within the heat exchanger.

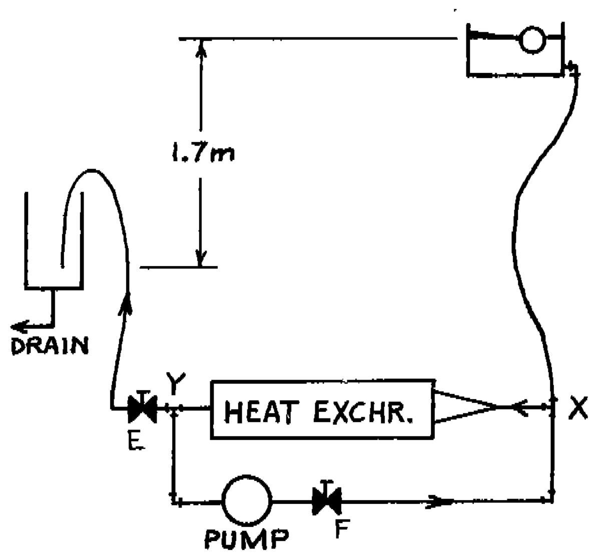

Figure 6. Heat exchanger water circuit

The equation representing the pressure loss through the heat exchanger has already been mentioned. The Darcy equation was used to determine pressure losses in the pipes and approximate empirical loss coefficients were used to take account of the bends and tees.

A standard domestic heating circulation pump, having three speeds, was used and its pressure versus flow rate characteristics were obtained from the suppliers.

The maximum flow rate through the heat exchanger will occur when there is no flow to drain and is limited by the available head between points X and Y in Figure 6. This flow rate was calculated to be about 20 litres per minute. At its lowest speed the pump would produce a flow rate of about 16 litres per minute and at the second speed a flow rate of about 21 litres per minute. Thus the calculations indicated that a flow to drain was possible at the lowest pump speed, or at a higher speed with throttle valve F partially closed.

The conclusion from the analysis was that a considerable increase in the flow rate through the condenser could be produced by recirculation, but, that this was not essential to the operation of the system. One effect of a high recirculation rate is that subcooling of the refrigerant in the heat exchanger is minimised.

THE LOAD STAND AS BUILT

A load stand has been built in accordance with the analyses presented. Standard hand operated diaphragm-sealed refrigeration shut-off valves were used for throttling. Valves A, B and D were three quarter inch flare and valve C was quarter inch flare. It was found difficult to control the suction superheat temperature and this was attributed to the insensitivity of valve C. This was replaced with a one eight inch needle valve and it was then found possible to achieve and maintain desired operating conditions. However, the operating experience to date is very limited.

Figure 7 is a photograph of the load stand. A sight glass located upstream of the liquid line throttle valve, C, is used to ensure that liquid is present.

CONCLUSIONS

A straightforward analysis of the load stand has been presented and the results indicate that it is feasible in principle. From the analysis and the limited operating experience to date it is felt that purpose designed throttle valves of higher sensitivity will be required.

NOTATION

A	=	effective flow area of specified valve, sq. m
a,b	=	constants
h	=	specific enthalpy of refrigerant at specified state, J/kg
j,k	=	constants in pressure loss equation
$\dot{m}$	=	mass flow rate of refrigerant, kg/s
n	=	polytropic index for a reversible adiabatic process of the refrigerant
P	=	pressure, N/(sq. m)
ΔP	=	pressure difference, N/(sq. m)
Q	=	rate of heat transfer, kW
r_c	=	clearance ratio (= compressor clearance volume divided by compressor swept volume)
r_p	=	pressure ratio or critical pressure ratio for a valve
T	=	temperature, K
ΔT	=	temperature difference, K
$\dot{V}$	=	volume flow rate of water, litres per second
v	=	specific volume at specified state, (cu. m)/kg
η	=	compressor volumetric efficiency

Subscripts

dis	discharge
hx	heat exchanger
mix	mixing
s	refrigerant saturation property
suc	suction

tot	total
u	upstream property
wi	water inlet property
wo	water outlet property
1 - 7	refrigerant equilibrium states (Fig. 4)

REFERENCES

1. Sodel, W., "Introduction to Computer Simulation of Positive Displacement Compressors", Purdue University, 1972.

2. Marriott, L.W.; F.R. Lady; L.L. Evans, "Accelerated Test for Rating Positive Displacement Compressors Using Computer Control", Proceedings of the 1974 Purdue Compressor Technology Conference.

3. Kartsounes, G.T.; R.A. Erth, (1971), "Computer Calculation of the Thermodynamic Properties of Refrigerants 12, 22 and 502", ASHRAE Transactions, vol. 77, part II, pages 88-103.

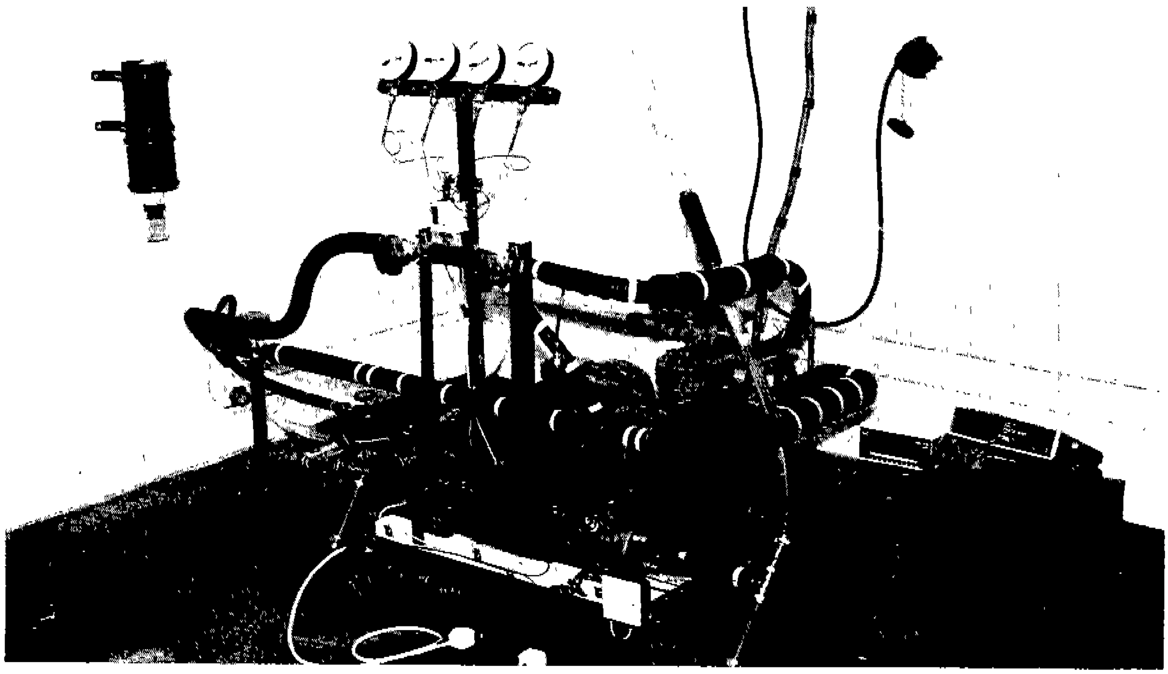

Figure 7. The refrigerant compressor load stand

HEAT EXCHANGER MASS FLOW RATE AS A PROPORTION OF THE TOTAL COMPRESSOR MASS FLOW RATE

REFRIGERANT NO.: 12

HEAT EXCHANGER OPERATING TEMP., DEG. C: 20.0

SUCTION SUPERHEAT, K: 0.0

SUCTION SAT. TEMP. DEG. C	DISCHARGE SATURATION TEMP., DEG. C 20.0	30.0	40.0	50.0	60.0	70.0
-50.0	.2972	.3204	.3410	.3594	.3758	.3904
-30.0	.2017	.2281	.2517	.2724	.2907	.3071
-10.0	.1145	.1437	.1695	.1923	.2127	.2308
10.0	.0363	.0677	.0956	.1204	.1424	.1620

SUCTION SUPERHEAT, K: 10.0

SUCTION SAT. TEMP. DEG. C	DISCHARGE SATURATION TEMP., DEG. C 20.0	30.0	40.0	50.0	60.0	70.0
-50.0	.2967	.3200	.3406	.3590	.3757	.3906
-30.0	.2017	.2281	.2515	.2723	.2909	.3076
-10.0	.1148	.1439	.1696	.1929	.2134	.2318
10.0	.0361	.0680	.0961	.1211	.1437	.1635

SUCTION SUPERHEAT, K: 20.0

SUCTION SAT. TEMP. DEG. C	DISCHARGE SATURATION TEMP., DEG. C 20.0	30.0	40.0	50.0	60.0	70.0
-50.0	.2959	.3195	.3402	.3587	.3753	.3902
-30.0	.2013	.2277	.2512	.2720	.2910	.3079
-10.0	.1146	.1439	.1699	.1930	.2137	.2322
10.0	.0362	.0681	.0963	.1215	.1439	.1640

SUCTION SUPERHEAT, K: 30.0

SUCTION SAT. TEMP. DEG. C	DISCHARGE SATURATION TEMP., DEG. C 20.0	30.0	40.0	50.0	60.0	70.0
-50.0	.2954	.3187	.3394	.3579	.3745	.3896
-30.0	.2007	.2274	.2509	.2719	.2908	.3077
-10.0	.1143	.1436	.1696	.1931	.2139	.2327
10.0	.0361	.0680	.0963	.1215	.1440	.1646

Table 1

HEAT EXCHANGER MASS FLOW RATE IN KG/S/1000 FOR AN IDEAL COMPRESSOR OF GIVEN DISPLACEMENT RATE AND CLEARANCE RATIO

REFRIGERANT NO.: 12

HEAT EXCHANGER OPERATING TEMP., DEG. C: 20.0

COMPRESSOR DISPLACEMENT RATE, LITRES PER SECOND: 1.00
CLEARANCE RATIO: .030

SUCTION SUPERHEAT, K: 0.0

SUCTION SAT. TEMP. DEG. C	DISCHARGE SATURATION TEMP., DEG. C 20.0	30.0	40.0	50.0	60.0	70.0
-50.0	.5401	.5002	.4259	.3128	.1559	.0000
-30.0	1.1192	1.2036	1.2452	1.2374	1.1788	1.0635
-10.0	1.4308	1.7546	2.0107	2.2008	2.3243	2.3786
10.0	.8781	1.6188	2.2537	2.7850	3.2183	3.5558

SUCTION SUPERHEAT, K: 10.0

SUCTION SAT. TEMP. DEG. C	DISCHARGE SATURATION TEMP., DEG. C 20.0	30.0	40.0	50.0	60.0	70.0
-50.0	.5144	.4769	.4071	.3012	.1554	.0000
-30.0	1.0683	1.1498	1.1893	1.1849	1.1335	1.0306
-10.0	1.3678	1.6763	1.9209	2.1083	2.2313	2.2907
10.0	.8316	1.5455	2.1538	2.6652	3.0924	3.4205

SUCTION SUPERHEAT, K: 20.0

SUCTION SAT. TEMP. DEG. C	DISCHARGE SATURATION TEMP., DEG. C 20.0	30.0	40.0	50.0	60.0	70.0
-50.0	.4903	.4556	.3898	.2903	.1531	.0000
-30.0	1.0209	1.0995	1.1383	1.1359	1.0911	.9977
-10.0	1.3066	1.6043	1.8418	2.0209	2.1420	2.2038
10.0	.7948	1.4780	2.0613	2.5532	2.9593	3.2828

SUCTION SUPERHEAT, K: 30.0

SUCTION SAT. TEMP. DEG. C	DISCHARGE SATURATION TEMP., DEG. C 20.0	30.0	40.0	50.0	60.0	70.0
-50.0	.4689	.4355	.3732	.2791	.1497	.0000
-30.0	.9764	1.0539	1.0923	1.0918	1.0502	.9644
-10.0	1.2504	1.5361	1.7645	1.9415	2.0611	2.1254
10.0	.7594	1.4139	1.9736	2.4465	2.8381	3.1611

Table 2

HEAT EXCHANGER HEAT TRANSFER RATE IN KW FOR AN IDEAL COMPRESSOR OF GIVEN DISPLACEMENT RATE AND CLEARANCE RATIO

REFRIGERANT NO.: 12

COMPRESSOR DISPLACEMENT RATE, LITRES PER SECOND: 1.00
CLEARANCE RATIO: .030

SUCTION SUPERHEAT, K: 0.0

SUCTION SAT. TEMP. DEG. C	DISCHARGE SATURATION TEMP., DEG. C 20.0	30.0	40.0	50.0	60.0	70.0
-50.0	.0846	.0810	.0711	.0537	.0275	.0000
-30.0	.1673	.1861	.1986	.2029	.1983	.1831
-10.0	.2073	.2629	.3106	.3496	.3788	.3968
10.0	.1247	.2376	.3411	.4333	.5136	.5807

SUCTION SUPERHEAT, K: 10.0

SUCTION SAT. TEMP. DEG. C	DISCHARGE SATURATION TEMP., DEG. C 20.0	30.0	40.0	50.0	60.0	70.0
-50.0	.0844	.0810	.0713	.0543	.0287	.0000
-30.0	.1674	.1863	.1988	.2037	.2000	.1862
-10.0	.2079	.2634	.3112	.3514	.3817	.4012
10.0	.1239	.2381	.3422	.4355	.5187	.5872

SUCTION SUPERHEAT, K: 20.0

SUCTION SAT. TEMP. DEG. C	DISCHARGE SATURATION TEMP., DEG. C 20.0	30.0	40.0	50.0	60.0	70.0
-50.0	.0842	.0810	.0715	.0547	.0296	.0000
-30.0	.1674	.1864	.1991	.2043	.2015	.1888
-10.0	.2078	.2639	.3124	.3526	.3836	.4041
10.0	.1240	.2384	.3429	.4369	.5197	.5903

SUCTION SUPERHEAT, K: 30.0

SUCTION SAT. TEMP. DEG. C	DISCHARGE SATURATION TEMP., DEG. C 20.0	30.0	40.0	50.0	60.0	70.0
-50.0	.0842	.0809	.0715	.0550	.0303	.0000
-30.0	.1672	.1867	.1996	.2053	.2027	.1907
-10.0	.2077	.2639	.3126	.3539	.3857	.4074
10.0	.1237	.2382	.3429	.4373	.5207	.5943

Table 3

EFFECTIVE FLOW AREAS OF VALVES ON THE TEST STAND FOR AN IDEAL
COMPRESSOR OF GIVEN DISPLACEMENT RATE AND CLEARANCE RATIO
**** UNITS SQUARE MM ****

SYMBOLS C - CHOKED FLOW, I - IMPOSSIBLE OPERATING CONDITION, O - VALVE FULLY OPEN
Z - ZERO COMPRESSOR FLOW

IT IS ASSUMED THAT THE ADIABATIC MIXING PRESSURE (=P7=P6=P5) LIES BETWEEN THE HEAT EXCHANGER PRESSURE (=P3=P4) AND THE SUCTION PRESSURE (=P1) AND THAT
P7 = P3 - (P3-P1)*(.20)

REFRIGERANT NO.: 12 ADIABATIC INDEX (BASED ON P AND V): 1.180
HEAT EXCHANGER OPERATING TEMP., DEG. C: 20.0
COMPRESSOR DISPLACEMENT RATE, LITRES PER SECOND: 1.00
CLEARANCE RATIO: .030

*** VALVE A ***

SUCTION SUPERHEAT, K: 0.0

SUCTION SAT. TEMP. DEG. C	DISCHARGE SAT. TEMP., DEG. C 20.0	30.0	40.0	50.0	60.0	70.0
-30.0	99.999 O	1.677	1.102	0.798 C	0.568 C	0.391 C
-10.0	99.999 O	3.815	2.597	1.971 C	1.501 C	1.139 C
10.0	99.999 O	7.386	5.094	3.933 C	3.060 C	2.390 C

SUCTION SUPERHEAT, K: 30.0

SUCTION SAT. TEMP. DEG. C	DISCHARGE SAT. TEMP., DEG. C 20.0	30.0	40.0	50.0	60.0	70.0
-30.0	99.999 O	1.581	1.044	0.761 C	0.549 C	0.386 C
-10.0	99.999 O	3.592	2.455	1.874 C	1.438 C	1.104 C
10.0	99.999 O	6.918	4.789	3.716 C	2.910 C	2.293 C

*** VALVE B ***

SUCTION SUPERHEAT, K: 0.0

SUCTION SAT. TEMP. DEG. C	DISCHARGE SAT. TEMP., DEG. C 20.0	30.0	40.0	50.0	60.0	70.0
-30.0	2.082	1.949	1.799	1.630	1.435	1.212
-10.0	5.741	5.525	5.290	5.033	4.746	4.422
10.0	17.705	17.231	16.733	16.205	15.636	15.012

SUCTION SUPERHEAT, K: 30.0

SUCTION SAT. TEMP. DEG. C	DISCHARGE SAT. TEMP., DEG. C 20.0	30.0	40.0	50.0	60.0	70.0
-30.0	1.958	1.832	1.693	1.538	1.364	1.168
-10.0	5.391	5.183	4.961	4.721	4.461	4.175
10.0	16.539	16.069	15.588	15.089	14.563	13.999

*** VALVE C ***

SUCTION SUPERHEAT, K: 0.0

SUCTION SAT. TEMP. DEG. C	DISCHARGE SAT. TEMP., DEG. C 20.0	30.0	40.0	50.0	60.0	70.0
-30.0	0.071	0.076	0.079	0.079	0.075	0.068
-10.0	0.105	0.129	0.148	0.162	0.171	0.175
10.0	0.100	0.185	0.258	0.318	0.368	0.406

SUCTION SUPERHEAT, K: 30.0

SUCTION SAT. TEMP. DEG. C	DISCHARGE SAT. TEMP., DEG. C 20.0	30.0	40.0	50.0	60.0	70.0
-30.0	0.062	0.067	0.069	0.069	0.067	0.061
-10.0	0.092	0.113	0.130	0.143	0.151	0.156
10.0	0.087	0.162	0.226	0.280	0.324	0.361

*** VALVE D ***

SUCTION SUPERHEAT, K: 0.0

SUCTION SAT. TEMP. DEG. C	DISCHARGE SAT. TEMP., DEG. C 20.0	30.0	40.0	50.0	60.0	70.0
-30.0	2.234 C	2.125 C	1.991 C	1.829 C	1.632 C	1.394 C
-10.0	4.945 C	4.834 C	4.697 C	4.530 C	4.326 C	4.079 C
10.0	10.467	10.342	10.188	10.000	9.770	9.489

SUCTION SUPERHEAT, K: 30.0

SUCTION SAT. TEMP. DEG. C	DISCHARGE SAT. TEMP., DEG. C 20.0	30.0	40.0	50.0	60.0	70.0
-30.0	2.089 C	1.989 C	1.869 C	1.724 C	1.551 C	1.345 C
-10.0	4.633 C	4.531 C	4.408 C	4.260 C	4.082 C	3.870 C
10.0	9.773	9.659	9.520	9.353	9.153	8.915

Table 4