

Table 3 Net Refrigeration Effect Conversion Factors*

Temp. Diff.	Suction Temperature, F															
	-30	-25	-20	-15	-10	-5	0	5	10	15	20	25	30	35	40	45
10	3.94	3.80	3.69	3.60	3.52	3.45	3.39	3.32	3.27	3.21	3.15	3.12	3.07	3.04	3.00	2.97
11	3.58	3.45	3.35	3.27	3.20	3.13	3.08	3.01	2.95	2.91	2.87	2.83	2.79	2.76	2.73	2.70
12	3.28	3.17	3.08	3.00	2.93	2.87	2.82	2.76	2.71	2.67	2.63	2.59	2.56	2.53	2.50	2.48
13	3.03	2.92	2.84	2.77	2.71	2.65	2.60	2.55	2.51	2.47	2.43	2.40	2.37	2.34	2.31	2.29
14	2.81	2.71	2.63	2.57	2.51	2.45	2.41	2.37	2.33	2.29	2.26	2.23	2.2	2.17	2.14	2.12
15	2.62	2.54	2.46	2.40	2.35	2.30	2.24	2.20	2.16	2.12	2.10	2.08	2.04	2.02	2.00	1.98
16	2.42	2.34	2.31	2.25	2.18	2.12	2.07	2.03	1.99	1.96	1.94	1.92	1.88	1.86	1.85	1.83
17	2.29	2.22	2.17	2.11	2.06	2.01	1.96	1.92	1.89	1.85	1.83	1.82	1.78	1.76	1.75	1.73
18	2.17	2.10	2.05	2.00	1.95	1.90	1.85	1.82	1.79	1.75	1.74	1.72	1.69	1.67	1.66	1.64
19	2.06	2.00	1.94	1.91	1.86	1.81	1.76	1.73	1.70	1.67	1.65	1.64	1.61	1.59	1.58	1.56
20	1.97	1.90	1.85	1.80	1.76	1.72	1.68	1.65	1.62	1.59	1.58	1.56	1.53	1.51	1.50	1.48
21	1.89	1.82	1.78	1.74	1.69	1.65	1.61	1.58	1.55	1.53	1.52	1.49	1.46	1.45	1.44	1.42
22	1.81	1.75	1.71	1.66	1.62	1.58	1.54	1.51	1.49	1.46	1.45	1.43	1.40	1.39	1.38	1.36
23	1.73	1.67	1.63	1.59	1.55	1.51	1.47	1.45	1.42	1.40	1.39	1.37	1.34	1.33	1.32	1.30
24	1.65	1.60	1.56	1.52	1.48	1.44	1.41	1.38	1.36	1.33	1.32	1.31	1.28	1.27	1.26	1.24
25	1.57	1.52	1.48	1.44	1.40	1.38	1.34	1.32	1.29	1.27	1.26	1.24	1.22	1.21	1.20	1.18
26	1.51	1.47	1.43	1.40	1.36	1.33	1.29	1.27	1.25	1.22	1.21	1.20	1.18	1.17	1.16	1.14
27	1.47	1.42	1.38	1.35	1.32	1.28	1.25	1.23	1.20	1.18	1.17	1.16	1.14	1.13	1.12	1.10
28	1.41	1.37	1.33	1.30	1.27	1.24	1.20	1.18	1.16	1.14	1.13	1.12	1.10	1.09	1.08	1.06
29	1.36	1.32	1.28	1.25	1.22	1.19	1.16	1.14	1.12	1.10	1.09	1.08	1.06	1.05	1.04	1.02
30	1.31	1.27	1.23	1.20	1.18	1.15	1.12	1.10	1.08	1.06	1.05	1.04	1.02	1.01	1.00	0.99

* To obtain net refrigeration effect at other than basic rating conditions, divide basic rating by factor selected from table.

Ratings of both the compressor and the condenser must be plotted on the same basis. For example, if the condenser is rated for 15 F subcooling, the compressor rating must also be based on 15 F subcooled liquid at the evaporator or appropriate corrections as suggested by the manufacturer must be applied to the ratings. Compressor saturated discharge temperature must be corrected for pressure drop in the hot gas line to obtain the saturated condensing temperature for plotting of the air-cooled condenser rating. The pressure drop is usually assumed to be the equivalent of 2 F deg drop in saturated temperature. The condensing capacity and condensing temperature are read at the intersection of the compressor curve (Q_c) and the condenser curve (GHR). The net refrigeration effect (NRE) and power (P) demand is then read by transposing the result to the (NRE) and P curves for

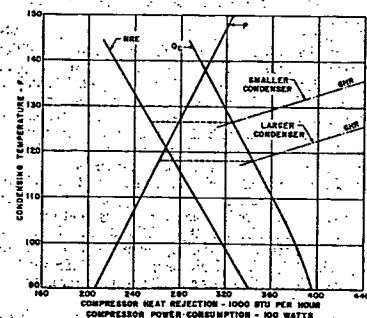


Fig. 16 Performance of Condenser-Compressor Combination

the particular compressor, as shown in Fig. 16. The same compressor, at 90 F ambient temperature, will operate at approximately 127 F condensing temperature and 250,000 Btu/hr net refrigeration effect with a power consumption of 28,000 watts, using the smaller condenser; or at 115 F condensing temperature and 270,000 Btu/hr (NRE) with a power consumption of 28,200 watts, using the next larger condenser. The ultimate condenser selection must satisfy the anticipated cooling requirements at design ambient conditions at the lowest possible annual operating cost of the total compressor-condenser system.

CONTROL OF AIR-COOLED UNIT CONDENSERS

In the simplest form of control, the fan on the unit condenser starts and stops with the compressor. This method was first used for air-cooled condensing units where the condenser fan was driven by the compressor motor. It is satisfactory where the air temperature at the condenser remains constant

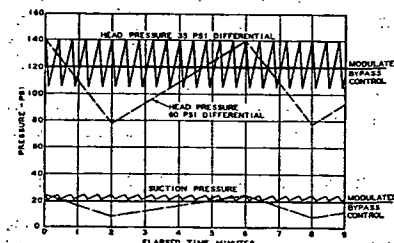


Fig. 17 Effect of Condenser Fan Cycling on Head and Suction Pressures

Condensers

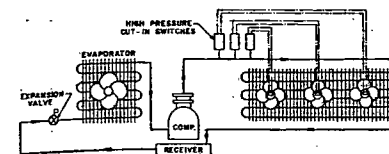


Fig. 18 Multiple Condenser Fans Controlled in Sequence

regardless of the time of day or season. This is seldom the case with remote condensers.

During winter operation, the condenser capacity is greatly increased, due to low entering air temperatures, so that the head pressure falls to an abnormally low level. While this is desirable from the point of view of compressor efficiency, the following problems arise: (1) insufficient pressure difference across the expansion valve due to low receiver pressure delivers insufficient refrigerant to the evaporator, (2) cold liquid with little or no subcooling evaporates in the liquid line, if that line runs through a warm space or must overcome a vertical lift before reaching the expansion valve; vapor at the expansion valve inlet can also result in insufficient feeding of the evaporator, (3) hot gas defrost systems cannot function without adequate head pressure at the onset of defrost, and (4) when a low-pressure switch controls compressor operation, the pressure switch may not be actuated when the pressure corresponding to the air temperature at the condenser is at or below the pressure settings of the low pressure switch.

To prevent excessively low head pressure, the condenser fan may be stopped by a reverse acting high-pressure switch whenever the head pressure falls below a predetermined limit. This control method is adequate where the condenser is protected from winds and where pronounced fluctuation of head pressure, with corresponding evaporator pressure changes, can be tolerated (Fig. 17).

Head pressure and evaporator pressure fluctuations caused by fan cycling can be minimized by using multiple fans con-

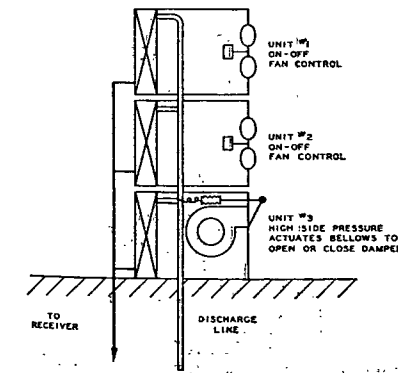
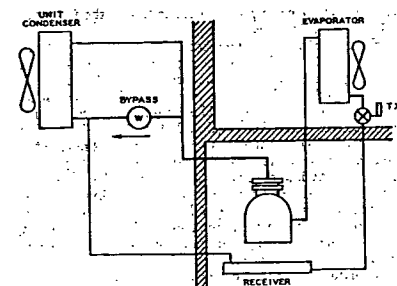


Fig. 19 Unit Condensers Installed in Parallel with Combined Fan Cycling and Damper Control

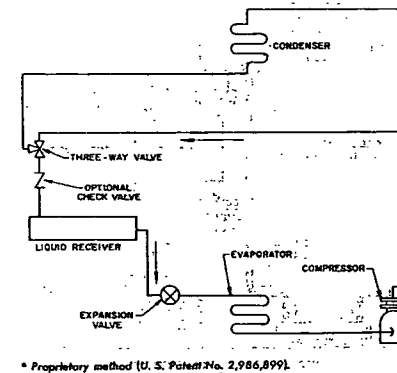


* Proprietary method (U. S. Patent No. 2,564,310).

Fig. 20 Condenser Control by the Modulated Bypass Method*

trolled in sequence. Fan control may be either in response to ambient temperatures or to condenser pressure. Head pressure control with multiple fans will minimize rapid cycling of the fan motor and rapid fluctuation in head pressure. (see Fig. 17). Fig. 18 illustrates this type of fan control.

When multiple fans are cycled in response to ambient temperature, one fan must be in operation at all times to prevent excessive condensing pressure. Thus, a two-fan unit may cycle only one fan and a three-fan unit may cycle two fans. The capacity of two-fan condensers will be reduced to approximately 55 percent capacity by cycling one fan, while the capacity of a three-fan condenser will be reduced to approximately 40 percent by cycling two fans. Further reduction in capacity to approximately 15 percent is possible by modulating the air flow through the uncontrolled fan section with speed control or with dampers on the air intake or discharge side (see Fig. 19). When used with propeller fan units, the closing of the damper may have to be limited or an over-



* Proprietary method (U. S. Patent No. 2,986,899).

Fig. 21 Modulated Bypass Control Using a Three-Way Valve