NATIONAL BUREAU OF STANDARDS REPORT

3790

EFFECT OF ENGINE AND FAN SPEEDS ON THE CAPACITY OF THE 1/3-TON THERMO KING REFRIGERATING UNIT, MODEL Q15G

by

C. W. Phillips
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Report to

Mechanical Engineering Division
Headquarters, Quartermaster Research & Development Command
Natick, Mass.

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Heating and Air Conditioning Section
Building Technology Division

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ABSTRACT

A study was made of the effect of engine and fan speeds on the net refrigerating capacity of a plug-type refrigerating unit manufactured by the U. S. Thermo Control Company. The stock unit is belt-driven by a gasoline engine thus fixing the speed ratios between engine and compressor and between engine and evaporator and condenser fans. In addition to studying the unit as furnished the effect of independently varying the fan speeds was evaluated using a separate fan drive. The results showed that there were optimum speeds for both the compressor and fans. The maximum net refrigerating capacity was obtained at an engine speed of about 2300 rpm with the pulleys provided on the unit. This was very close to the design speed of the unit. However, the design speed of the fans was too high resulting in a loss of net capacity at all engine speeds because of the excessive amount of energy dissipated in the refrigerator by the evaporator fan. The maximum net refrigerating capacity was observed for a fan speed of about 1100 rpm at each of three engine speeds in the range from 1800 rpm to 3000 rpm. The net refrigerating capacity was increased from 4700 Btu/hr to 5350 Btu/hr by decreasing the fan speed from 2100 rpm to 1100 rpm with an engine speed of 2400 rpm in each case. The net refrigerating capacity increased almost uniformly with increase in compressor speed in the range from 1200 rpm to 2000 rpm when the fan speed was held constant indicating rather uniform volumetric efficiency and good valve performance in this range of speed. Experience with the test specimen indicated that the belt drive needed some improvement and that certain components of the engine and some of the controls should be more accessible.

1. INTRODUCTION

At the request of the Office of The Quartermaster General tests were made as outlined in letters dated January 24th, May 2, 1951 and December 1, 1952, of a 1/3-ton gasoline-engine driven refrigerating unit, Model Q15G, of the plug type manufactured by the U. S. Thermo Control Company of Minneapolis, Minnesota. It was tested in a 150 cubic foot, portable, walk-in refrigerator of the type usually equipped with this size refrigerating unit in the field. The refrigerator was employed as a calorimeter for the purpose of these tests.

The purposes of this investigation can be divided into three main parts as follows:

A. Measurement of the net refrigerating capacity of the unit at design compressor speed for a range of ambient temperatures from 70°F to 110°F and at refrigerator temperatures of 35°F, 0°F, and the lowest temperature attainable without internal load in the refrigerator.

B. Investigation of the effect of changing the compressor speed on net refrigerating capacity when the speed ratios between the compressor, and the evaporator and condenser fans remained fixed at the design value. Investigation of the effect in the net refrigerating capacity of changing the compressor speed independently of the fan speeds and vice versa.

C. Observation of the mechanical features of the refrigerating unit with regard to their suitability for military purposes.

The results observed with respect to the above three objectives are reported in Parts A, B, and C of the section on Test Results, respectively.

2. DESCRIPTION OF TEST SPECIMEN

The specimen refrigerating unit was identified as follows:

(NBS Test Specimen 54-51)
U. S. Thermo Control Co., Minneapolis, Minnesota
1/3-Ton
Model Q15G
Serial Number

The unit was of the plug-in type and used Freon 12 as the refrigerant. The compressor was identified as Thermo-King,

Model 2R, Serial Number 1051. The refrigerating unit was driven by a gasoline engine manufactured by D. W. Onan & Sons, Inc. of Minneapolis, Minnesota with the following nameplate data:

Model CK-MS
Specification 190J
Serial Number 31-430271
Maximum Horsepower 10.1 at 3,000 rpm

The condenser fan was of the propeller type made of cast aluminum. It had 10 blades, 5-5/8" long, and the diameter was 17 inches. Ambient air was drawn through the condenser, forced around the engine and compressor and discharged from the top and sides of the condensing unit section.

The evaporator fan was of the centrifugal type made of aluminum plates. The fan wheel was 14-1/2 inches in diameter and 1-3/4 inches in width. There were 17 fan blades which were 1-1/2 inches long. The air was drawn into the evaporator from the front and discharged from the top of the evaporator section.

The condenser and evaporator fans were mounted on opposite ends of a common shaft. The pulleys of the engine, idler, fans and compressor were connected in that order in a clockwise direction by two V-belts. The diameters of the pulleys were 6-3/4", 7", 7-3/8" and 10", respectively.

Five photographs of the specimen refrigerating unit are attached as Figures 1-5 inclusive. Figure 1 and Figure 2 show the exterior views of the condenser and evaporator sections, respectively. Figure 3 is a three-quarter view showing the condenser section with the perforated panels removed. Figure 4 is a left side view of the condenser section showing the relative locations of the engine, condenser fan, and compressor. Figure 5 is the side view of the evaporator section showing the location of the evaporator fan.

The physical dimensions of the refrigerating unit were as follows:

(1) Gross weight, without battery, lb. 583.5

		Width	Height	Depth
(2)	Overall dimension, include battery, in.	34	59-3/8	32-1/4
(3)	Condensing unit section without battery, in.	34	50	19
(4)	Evaporator section, in.	27-1/2	28	10-1/4
(5)	Plug Section, in.	28	28	3

(6) Compressor:

a)	Model	2R
b)	Serial Number	1051
c)	Type	2 cylinder vertical
d)	Bore, in.	2-1/4
e)	Stroke, in.	1-3/4
f)	Speed, rpm, design	1680
g)	Displacement, cu. in.	13.91

(7) Condenser:

a)	Width, in.	27
b)	Height, in.	23-1/2
c)	Depth, in.	1-3/4
d)	Fin size, in.	23-1/2 x 1-3/4
e)	Fin spacing	10 per inch
f)	Tube size, in.	3/8 O.D.
g)	Number of tubes	48
h)	Series or parallel	24 tubes in series per circuit, 2 circuits in parallel
i)	Primary surface area, sq. ft.	10.6
j)	Secondary surface area, sq. ft.	134.3
k)	Total surface area, sq. ft.	144.9
l)	Ratio primary to total surface, %	7.3
m)	Tube material	Copper
n)	Fin material	Aluminum

(8) Engine:

a) Model number	CK-MS
b) Specification	190J
c) Serial number	31-430271
d) Type	4 cycle 2 cylinder opposed horizontal, air-cooled
e) Material	Aluminum
f) Speed, rpm, design	2400
g) Rated maximum brake horse power at 3000 rpm	10.1
h) Lubrication	Force feed
i) Fuel system	Pump feed to carburetor
j) Ignition system	Special 12-volt motor-generator built into flywheel of engine provided automatic starting, battery charging and ignition.
k) Cooling system	Fan on armature of engine generator forced air over engine block and finned cylinder heads as directed by engine housing.
l) Governor	Pierce Governor Company - GC-1408 2 1

(9) Evaporator:

a) Width, in.	24
b) Height, in.	25
c) Depth, in.	4
d) Fin size, in.	4 x 25
e) Fin spacing	8 per inch
f) Tube size, in.	1/2 O.D.
g) Number of tubes	80
h) Series or parallel	20 tubes in series per circuit
i) Primary surface area, sq. ft.	20.9
j) Secondary surface area, sq. ft.	24.8
k) Total surface area, sq. ft.	245.7
l) Ratio primary to total surface, %	8.5
m) Tube material	Copper
n) Fin material	Aluminum

3. TEST PROCEDURE

The specimen refrigerating unit was installed in a 150 cu. ft. portable, walk-in refrigerator for the capacity tests, which was located in a test room where the ambient temperature could be controlled. Refrigerator temperature was maintained at the desired level by an electronic rheostat controlling the input to electric heaters which were used for the internal load. The heat output of these heaters was measured by means of watthour meters in the supply circuits. Temperatures were measured by means of calibrated thermocouples using an electronic, constant-balance type potentiometer.

The engine speeds were varied by adjusting the governor. The compressor, fan, idler, and engine were connected with a pair of double-faced V-belts during Part A of the tests. Therefore, the compressor and fan speeds were proportional to the engine speed as determined by the pulley ratio. During Part B of the tests, the compressor was driven by the gasoline engine and the fans were driven with a single V-belt by a 3/4 H.P. direct current motor installed on top of the condenser section. The idler was eliminated and the engine and compressor were connected with two V-belts using the original pulleys. The condensate drain fitting had to be removed in order to permit installation of these belts. The objective of this modification was to provide independent control of the fan and compressor speeds so the fan speed could be determined for each engine speed that would result in the maximum net refrigerating capacity. The fan speeds were regulated by variable external field resistance and all speeds were measured by means of a stroboscopic tachometer.

After steady state test conditions were reached, they were maintained for an average period of about 5 hours while observations of pressures, temperatures, speeds, voltage, electric current and wattage were made at 30 minute intervals.

During the course of the capacity tests, some operational difficulties were encountered. They were recorded when they occurred for reporting as a part of the operating experience with this unit.

4. TEST RESULTS

Part A

In preparation for the capacity tests, a test was made to determine the heat transmission factor of the 150 cubic foot, portable, walk-in refrigerator in Btu per hour per degree temperature difference. Refrigerator temperature averaged 35.3°F and the ambient temperature averaged 136.2°F for eleven hours during steady state conditions. The heat transmission factor was computed to be 23.62 Btu/hr (°F temperature difference).

Results of the capacity tests with various engine speeds and proportional fan speeds at a refrigerator temperature of 0°F and an ambient temperature of 110°F are summarized in Table 1 and shown graphically in Fig. 6. The engine speed bore an average ratio of 1.56 to 1 to the compressor speed and 1.11 to 1 to the fan speed during these tests. However, these ratios did not remain constant for all tests indicating that belt slippage was occurring during some tests. This problem is discussed in Part C of this report.

Fig. 6 shows that the net refrigerating capacity reached a maximum value of about 4700 Btu/hr at an engine speed in the neighborhood of 2300 rpm. The net refrigerating capacity decreased for speeds above and below 2300 rpm reaching a value of 1/3-ton or 4000 Btu/hr at engine speeds of about 1760 rpm and 2920 rpm.

It would be expected that the total refrigerating capacity of the compressor would increase proportionally with engine speed if the volumetric efficiency remained constant. However, the power required to drive the evaporator fan would increase as the cube of the fan speed or the engine speed. Consequently, at some engine speed the rate of increase of compressor capacity would be equal to the rate of increase of evaporator fan power. Above this speed the net refrigerating capacity would decrease assuming that the heat transfer coefficient of the evaporator surfaces remained constant throughout the speed range. This optimum engine speed was about 2300 rpm on the test specimen which was approximately the design speed of the unit.

Table 1 shows that the comparative differences between inlet and outlet air on the evaporator and between inlet and

outlet air on the condenser remained reasonably steady in the engine speed range from 1800 rpm to 2700 rpm. This indicates that the total heat transfer in each of these components increased steadily with increase in engine speed since the air delivery of a fan in a given circuit is proportional to the fan speed. There was about a 20 percent decrease in the temperature difference between inlet and discharge air of the evaporator at an engine speed of 3000 rpm as compared to the lower engine speeds indicating a decrease in the net refrigerating capacity.

The slow rise in discharge pressure and the gradual decrease in suction pressure with increased engine speed indicate a steady increase in the total amount of heat transferred by the condenser and evaporator, respectively. The greater air flow over the condenser and evaporator surfaces does not significantly change the heat transfer coefficient inside these heat exchangers. Consequently, a greater temperature head was required to exchange more heat between the refrigerant and these surfaces.

If it were assumed that the compressor capacity was directly proportional to engine speed and the power absorbed by the evaporator fan directly proportional to the cube of the fan speed during this series of tests, the net refrigerating capacity of the unit could be expressed by an equation of the form:

$R = AS - BS^3$ where

R is the net refrigerating capacity in Btu/hr
S is the engine speed in rpm
A and B are constants which can be evaluated using the observed values of R and S for two test conditions.

Using values of R taken from the curve in Fig. 6 for S = 2100 and 2400 rpm, respectively the following relationship is obtained:

$$R = 3.09S - 0.00197S^3 \qquad (1)$$

From this equation values of the net refrigerating capacity at other engine speeds in the range of the tests

have been computed and plotted in Fig. 6. The computed values fit the curve drawn through the observed values very well except at engine speeds near 1800 rpm indicating that equation (1) adequately represents the relationship between net refrigerating capacity and speed over most of the speed range tried.

The discharge and suction pressures and the air temperatures at the outlet of evaporator and condenser were probably affected to a small extent by the belt slippage revealed during some of the tests but these effects are considered small compared to the broader capacity trends caused by compressor displacement and power comsumption of the fans.

Fig. 7 shows the hourly gasoline consumption of the unit plotted against the engine speed. The relationship was approximately linear over the range of the tests. The gasoline consumption per ton of net refrigerating capacity increased steadily over the range of speed shown in Fig. 7 and was slightly more than twice as great for a speed of 3000 rpm as for a speed of 1800 rpm.

The relation between the net refrigerating capacity of the Thermo King Q15G unit and refrigerator temperature is shown in Fig. 8 for four different ambient temperatures ranging from 70°F to 125°F. At each ambient temperature tests were made with refrigerator temperatures of 35°F, 0°F, and the lowest temperature that could be attained without internal heat load. The observed results are summarized in Table 2.

The engine was operated at a speed of about 1700 rpm for these tests corresponding to compressor and fan speeds of about 1200 rpm and 1600 rpm respectively. However, there was more than 100 rpm speed variation during the series of tests for each of the components: engine, compressor, and fans. These variations were caused by belt slippage in some cases but depended also on the amount of load on the engine. It was observed that the speed of the engine changed with engine load considerably for a given governor setting.

Fig. 8 shows that the net refrigerating capacity varied approximately in linear relation to the refrigerator temperature at each ambient temperature. The capacity curves at the four ambient temperatures are nearly parallel, converging slightly as the refrigerator temperature decreased. On the average the net refrigerating capacity increased 140 Btu/hr per degree rise in refrigerator temperature for the three higher ambient temperatures. At an ambient temperature of 125°F the net refrigerating capacity increased 117 Btu/hr for each degree rise in refrigerator temperature.

Part B

Capacity tests were made to determine the optimum fan speeds at various engine speeds and the results are summarized in Table 3. The refrigerator temperature and the ambient temperature were maintained at about 0°F and 110°F respectively during the tests. Speeds of the fan and engine were controlled independently for each test. Fan speeds were varied from 500 rpm to 2100 rpm and engine speeds from 1800 rpm to 3000 rpm at 300 rpm intervals.

Figure 9 shows the net refrigerating capacity plotted against engine speed for average fan speeds of 834 rpm, 1353 rpm and 2097 rpm. At each fan speed the unit capacity increased approximately in direct proportion to the increase in engine speed in the range from 1800 rpm to 3000 rpm. Each curve drops slightly at the higher engine speeds. This is probably accounted for by the increase in compression ratio that occurred with increase in engine speed. Table 3 shows that the discharge pressure increased and the suction pressure decreased as the engine speed increased for any constant value of fan speed. This was to be expected because greater temperature difference between air and refrigerant would be required in both evaporator and condenser to transfer greater quantities of heat as the engine and compressor speeds increased.

A comparison of Fig. 6 and Fig. 9 reveals the effect of proportional and constant fan speed on net refrigerating capacity at any given engine speed. Net refrigerating capacity is plotted against engine speed in both figures, but in Fig. 6 the fan speed was increased proportionally with the engine speed whereas in Fig. 9 the fan speed remained constant while the engine speed was increased. This comparison shows that the greater refrigerating capacity produced by increased engine and compressor speed was

quickly offset by the greater amounts of fan energy dissipated in the refrigerator by the evaporator fan.

The marked curve in Fig. 6 shows approximately how the net refrigerating capacity would increase with engine speed if the fan speed remained constant at 1500 rpm, the fan speed corresponding to the lowest value of engine speed on the solid curve. The marked curve is based on the data in Table 3. The difference in capacity represented by the two curves in Fig. 6 is a measure of the increased energy dissipated in the refrigerator by the evaporator fan when the fan speed is increased proportionally with the engine speed as compared to constant fan speed. The difference amounted to 1600 Btu/hr at an engine speed of 3000 rpm.

Figure 9 shows that the net refrigerating capacity at any given engine speed increased in the range of fan speed from 834 rpm to 1062 rpm, but began to decrease in the range of fan speed between 1062 rpm and 1353 rpm and decreased quite significantly for a fan speed of 2097 rpm. This relationship is revealed more clearly in Fig.10 where net refrigerating capacity is plotted against fan speed for three engine speeds. Fig.10 shows that the net refrigerating capacity increased with fan speed up to a value of about 1100 rpm after which it decreased rather rapidly with further increase in fan speed.

Table 3 shows that the compression ratio decreased as the fan speed was increased for any constant value of engine speed. This was the result of better heat transfer in both evaporator and condenser. The pumping capacity of the compressor increased as the compression ratios decreased even though the compressor speed remained constant and the increase in compressor capacity was greater than the increase in evaporator fan power for fan speeds up to 1100 rpm. Above this fan speed the energy dissipated by the evaporator fan increased more rapidly than the compressor capacity so the net refrigerating capacity decreased.

If the data in Table 3 are plotted to show the relation of compression ratio to fan speed at constant compressor speed, it is revealed that the curve approximates a straight line. Calorimeter studies of other compressors have shown that the capacity of a compressor decreases almost linearly with increase in compression ratio. These two facts can be combined

in the following equation to show how the total refrigerating capacity of the compressor in the Thermo King unit might be affected by the fan speeds.

$$R_t = A - BS \quad \text{where}$$

R_t is the total refrigerating capacity of the compressor

S is the fan speed

A and B are constants

However, the net refrigerating effect of the unit would be equal to the difference between the total refrigerating capacity of the compressor and the heat dissipated by the evaporator fan inside the refrigerator. But since the fan power is proportional to the cube of the fan speed, the net refrigerating capacity of the unit could be expressed by an equation of the form:

$$R = A + BS - CS^3 \quad \text{where}$$

R is the net refrigerating capacity in Btu/hr

S is the fan speed in rpm

A, B and C are constants which can be evaluated using the observed values of R and S for three test conditions at constant engine speed.

Using values of R taken from the curve in Fig.10 for S = 800, 1300 and 2100 rpm respectively, and an engine speed of 3030 rpm, the following relationship is obtained:

$$R = 4572 + 1.326S - 0.26 \times 10^{-6} S^3 \qquad (2)$$

From this equation values of the net refrigerating capacity at other fan speeds in the range between 800 and 2100 rpm have been computed and plotted in Fig.10. The computed values lie near the curve drawn through the observed values for an engine speed of 3030 rpm, although the curvature of the line through the computed values is slightly different and the fan speed

corresponding to the maximum value of capacity is a little higher for equation 2 than for the observed data. The relatively good agreement of the computed and observed values indicates that an equation similar to equation 2 represents the relation between net refrigerating capacity and fan speed of the Thermo King unit at constant engine speed.

If the theoretical relation between volumetric efficiency and compression ratio is used instead of the linear relation assumed in equation (2), an equation of the following form results:

$$R = A - BP^{\frac{1}{1.06}} - CS^{3} \quad \text{where} \qquad (3)$$

R is the net refrigerating capacity in Btu/hr

S is the fan speed in rpm

P is the compression ratio of the compressor

A, B, and C are constants which can be evaluated if three sets of observed values of R, P, and S are substituted in the equation.

It was found that the curve described by equation (3) does not fit the observed data plotted in Fig.10 any better than the curve for equation (2) and equation (3) requires a knowledge of the relation between compression ratio and fan speed for the unit under test. A more comprehensive study of the performance characteristics of the various components in the Thermo King unit might reveal an equation that better describes the relation between net refrigerating capacity and fan speed than either equation (2) or (3).

Part C

A number of defects and mechanical failures were observed during the course of the tests. These are described below.

The joint between rubber and glass in the sight glass in the liquid refrigerant line was loose causing the unit to lose its refrigerant charge during the storage period prior to the tests. A close coupling of the double flare nut type between

the receiver valve and the sight glass split during the tests causing loss of refrigerant.

Considerable difficulty was experienced with the belt drives. When the unit was first started for test, it was found that the belts were loose and the compressor pulley was not aligned with the engine pulley. As a result the belts did not ride down in the groove of the compressor pulley. The belts became loose twice during the course of the tests indicating that the automatic belt-tightener did not function properly. The spring furnished on the belt-tightener was replaced with a turnbuckle and a short spring one inch in diameter made of 1/8-inch tempered wire. This modification prevented further difficulty with belt slippage. It was observed, however, that the arc of belt contact on the compressor pulley was small because of the relative positions of the pulleys on the engine, compressor, fans and the idler.

The battery charging rate was initially observed to be 12.5 amperes. It was reduced to 2.5 amperes in accordance with the manufacturer's instructions. It was observed that the batteries required daily filling with water even after the charging rate was reduced.

Considerable maintenance was required on the distributor points to keep the engine in operation. The distributor was not readily accessible. Considering the frequent need for servicing, attention should be given to a relocation of some of these components.

The manually-operated hot gas defrost valve was not readily accessible. It was located in a place which caused the operator to be burned by the refrigerant line at the condenser inlet when opening the valve. A relocation of the thermometer on the control panel and changing the position of the defrost valve handle would remove this hazard.

The metal side panel had to be removed from the unit to start the engine, adjust the thermostat, or fill the gasoline tank. If these functions could be performed through an access opening, the operator would be more likely to leave the side panel on the unit.

5. DISCUSSION AND CONCLUSIONS

This study of the relation between net refrigerating capacity and engine and fan speeds for the 1/3-ton Thermo King refrigerating unit, Model Q15G, showed that there were optimum speeds for both components. The maximum net refrigerating capacity was obtained at an engine speed of about 2300 rpm with the pulleys provided on the unit. This speed was very close to the design speed of the unit. However, the design speed of the fans was too high resulting in a loss of net capacity at all engine speeds, because of the excessive energy dissipated in the refrigerator by the evaporator fan. The maximum net refrigerating capacity of the unit was observed for a fan speed of about 1100 rpm at each of three engine speeds, viz, 1800 rpm, 2400 rpm and 3000 rpm. The net refrigerating capacity was increased from 4700 Btu/hr to 5350 Btu/hr by decreasing the fan speed from 2100 rpm to 1100 rpm with an engine speed of 2400 rpm in each case. Therefore, it is recommended that the pulley on the fan shaft be reduced in diameter to provide a fan speed of approximately 1100 rpm. at the design engine speed, thereby increasing the capacity and reducing the load on the engine.

The net refrigerating capacity increased almost uniformly with increase in compressor speed in the range from 1200 rpm to 2000 rpm when the fan speed was held constant indicating rather uniform volumetric efficiency and good valve performance in this range of speed. These results suggest the possibility of using the same compressor for more than one duty by adjusting the compressor speed to the anticipated load.

Experience with the test specimen indicated that the belt drive needed some improvement and that accessibility had been sacrificed to compactness in a few important respects.

EFFECT OF COMPRESSOR SPEED ON UNIT
CAPACITY AT CONSTANT REFRIGERATOR AND AMBIENT TEMPERATURES

Test No.	1	2	3	4	5	6
Net Refrigerating, capacity, Btu/hr	4060	4670	4690	4470	3940	3780
Engine Speed, r.p.m.	1785	2099	2364	2693	2978	2974
Compressor Speed, r.p.m.	1120	1339	1526	1801	1910	1935
Fan Speed, r.p.m.	1590	1858	2108	2428	2912	2656
Ratio of Engine Speed to Comp. Speed	1.59	1.64	1.55	1.49	1.56	1.54
Ratio of Engine Speed to Fan Speed	1.12	1.18	1.12	1.11	1.02	1.12
Gasoline Consumption, lb/hr	3.2	4.0	4.8	6.2	7.4	6.5
Discharge Pressure, psig.	205	211	214	219	226	226
Suction Pressure, psig.	4.4	5.0	4.2	3.3	2.9	2.9
Avg. Refrigerator Temp.,°F	0.0	0.2	0.1	0.2	0.3	0.2
Avg. Ambient Temp., °F	110.7	109.9	109.0	110.9	110.7	110.3
Suction Gas Temp., °F	82.2	76.5	78.7	78.7	78.9	78.8
Temp. of Air Entering Condenser, °F	117.5	118.2	119.2	117.9	117.9	118.1
Temp. of Air Leaving Condenser, °F	124.6	126.3	126.7	125.9	125.8	126.5
Temp. of Air Entering Evaporator, °F	1.7	1.9	2.5	1.5	1.6	1.7
Temp. of Air Leaving Evaporator, °F	-3.8	-3.1	-2.9	-3.6	-2.6	-2.2

TABLE 1

THE NATIONAL BUREAU OF STANDARDS

TEST NO.

Net Refrg.
Engine Spe
Compressor
Fan Speed
Gasoline (
Avg. Refrg
Avg. Ambie
Discharge
Suction Pr
Compressic
Suction Ga
Temp. of A
ing Conde
Temp. of A
ing Conde
Temp. of A
ing Evapc
Temp. of A
ing Evapc

*Vacuum,

EFFECT OF AMBIENT TEMPERATURE AND REFRIGERATOR TEMPERATURE ON UNIT CAPACITY
AT CONSTANT COMPRESSOR AND FAN SPEEDS

TEST NO.	Units	1	2	3	4	5	6	7	8	9	10	11	12
Net Refrg. Capacity	Btu/hr	10,570	5,280	5,440	2,230	10,040	4,360	2,530	8,620	3,990	2,770	7,490	3040
Engine Speed	r.p.m.	1,726	1,804	1,779	1,801	1,701	1,741	1,764	1,613	1,699	1,726	1,698	1708
Compressor Speed	r.p.m.	1,127	1,257	1,191	1,186	1,205	1,067	1,136	1,151	1,215	1,225	1,199	1211
Fan Speed	r.p.m.	1,529	1,657	1,601	1,581	1,605	1,447	1,521	1,526	1,605	1,626	1,601	1608
Gasoline Consumption	lb/hr	3.4	3.1	3.2	2.7	3.6	3.1	2.8	4.0	3.6	3.3	4.1	3.3
Avg. Refrg. Temp.	°F	36.4	1.3	0.7	-22.7	35.2	0.7	-16.3	34.8	0.1	-6.7	35.5	-2.5
Avg. Ambient Temp.	°F	71.3	71.1	71.5	71.5	91.0	91.4	90.8	109.8	110.2	110.5	124.1	126.3
Discharge Pressure	psig.	170	130	127	116	174	165	153	225	188	188	261	224
Suction Pressure	psig.	17.2	4.3	4.6	6.0*	17.0	5.5	0.5*	19.0	6.5	2.8	19.0	5.5
Compression Ratio		5.80	7.64	7.37	11.20	5.96	8.66	11.70	7.12	9.59	11.65	8.20	11.87
Suction Gas Temp.	°F	64.6	57.9	55.8	62.1	72.6	70.2	74.3	81.7	79.4	85.2	88.1	90.9
Temp. of Air Entering Condenser	°F	84.4	77.9	79.1	76.8	100.8	97.9	95.7	119.3	117.2	116.1	133.9	131.9
Temp. of Air Leaving Condenser	°F	91.1	82.0	81.9	77.6	108.0	100.6	96.1	126.8	120.2	117.7	140.6	134.7
Temp. of Air Entering Evaporator	°F	38.6	2.6	1.7	-21.7	37.5	2.1	-15.4	36.8	1.0	-5.6	37.1	-1.6
Temp. of Air Leaving Evaporator	°F	22.8	-4.9	-5.9	-25.1	22.7	-4.6	-18.8	23.9	-4.8	-9.7	26.0	-5.6

*Vacuum, In. Hg.

TABLE 2

THE NATIONAL BUREAU OF STANDARDS

Nominal Engine

Test No.

Net Refrigerat-
ing Capacity
Engine Speed
Compressor Spe
Fan Speed
Gasoline Con-
sumption
Avg. Refrigera
Temperature
Avg. Ambient
Temperature
Discharge Pres-
sure
Suction Pressu
Compression Ra
Suction Gas Te
perature
Temperature of
Entering Con
ser
Temperature of
Air Leaving
Condenser
Air Temperatur
Diff. across
Condenser
Temperature of
Air Entering
Evaporator
Temperature of
Air Leaving
Evaporator
Air Temperatur
Diff. Across
Evaporator

EFFECT OF FAN SPEED ON NET REFRIGERATING CAPACITY AT CONSTANT ENGINE SPEED

Nominal Engine Speed, RPM		1800			2100			2400							2700	3000		
Test No.	Units	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
Net Refrigerating Capacity	Btu/hr	4220	4498	3374	4972	4264	3785	4427	5016	5063	5304	5408	4721	4210	4584	5484	5669	4993
Engine Speed	r.p.m.	1841	1828	1809	2107	2095	2089	2435	2400	2421	2428	2396	2435	2400	2687	3030	3045	3014
Compressor Speed	r.p.m.	1230	1210	1213	1388	1390	1387	1600	1580	1595	1608	1570	1601	1589	1763	2005	2015	1996
Fan Speed	r.p.m.	828	1354	2114	1355	1718	2089	515	675	831	1013	1062	1718	2084	2110	844	1350	2086
Gasoline Consumption	lb/hr	3.1	3.3	2.9	3.5	3.6	3.7	3.9	4.0	3.9	3.9	4.2	4.3	4.0	4.6	4.8	4.8	5.0
Avg. Refrigerator Temperature	°F	-1.3	0.0	-0.1	-0.1	-0.3	-0.4	0.1	0.0	-0.1	-0.5	-0.2	1.3	-0.2	-0.5	0.1	-0.3	0.2
Avg. Ambient Temperature	°F	104.8	109.8	111.6	107.4	110.9	111.2	107.9	107.2	110.0	109.3	105.6	111.5	111.6	110.7	110.8	109.9	108.0
Discharge Pressure	psig	194	189	187	192	187	186	194	194	196	193	191	192	195	190	201	193	203
Suction Pressure	psig	2.4	3.5	5.0	3.0	3.5	3.6	1.1"Hg	1.9	1.5	2.2	1.6	2.9	2.7	1.8	1.0"Hg	0.7"Hg	2.2
Compression Ratio		12.27	11.25	10.28	11.73	11.13	11.02	14.83	12.64	13.08	12.35	12.70	11.80	12.11	12.47	15.30	14.55	12.95
Suction Gas Temperature	°F	73.2	75.9	79.2	72.7	76.6	74.3	70.5	78.4	69.7	72.4	72.6	73.8	79.4	77.2	73.3	75.6	74.8
Temperature of Air Entering Condenser	°F	118.8	118.0	118.0	118.9	117.5	117.6	118.3	119.1	117.2	117.9	118.9	120.1	118.9	117.6	118.5	118.2	119.0
Temperature of Air Leaving Condenser	°F	131.3	125.5	123.1	128.2	124.8	122.6	133.3	131.6	130.0	129.2	128.8	129.3	126.2	124.9	131.6	127.1	128.6
Air Temperature Diff. across Condenser	°F	12.5	7.5	5.1	9.3	7.3	5.0	15.0	12.5	12.8	11.3	9.9	9.2	7.3	7.3	13.1	8.9	9.6
Temperature of Air Entering Evaporator	°F	0.1	1.0	0.7	0.8	1.1	0.6	-	-	2.1	0.9	1.0	2.6	0.7	0.3	1.1	0.5	1.5
Temperature of Air Leaving Evaporator	°F	-9.7	-6.3	-2.8	-7.3	-4.2	-3.5	-15.5	-7.3	-11.3	-9.8	-9.6	-3.2	-3.6	-5.0	-12.8	-9.0	-4.3
Air Temperature Diff. Across Evaporator	°F	9.8	7.3	3.5	8.1	5.3	4.1	-	-	13.4	10.7	10.6	5.8	4.3	5.3	13.9	9.5	5.8

TABLE 3

THE NATIONAL BUREAU OF STANDARDS

Functions and Activities

The functions of the National Bureau of Standards are set forth in the Act of Congress, March 3, 1901, as amended by Congress in Public Law 619, 1950. These include the development and maintenance of the national standards of measurement and the provision of means and methods for making measurements consistent with these standards; the determination of physical constants and properties of materials; the development of methods and instruments for testing materials, devices, and structures; advisory services to Government Agencies on scientific and technical problems; invention and development of devices to serve special needs of the Government; and the development of standard practices, codes, and specifications. The work includes basic and applied research, development, engineering, instrumentation, testing, evaluation, calibration services, and various consultation and information services. A major portion of the Bureau's work is performed for other Government Agencies, particularly the Department of Defense and the Atomic Energy Commission. The scope of activities is suggested by the listing of divisions and sections on the inside of the front cover.

Reports and Publications

The results of the Bureau's work take the form of either actual equipment and devices or published papers and reports. Reports are issued to the sponsoring agency of a particular project or program. Published papers appear either in the Bureau's own series of publications or in the journals of professional and scientific societies. The Bureau itself publishes three monthly periodicals, available from the Government Printing Office: The Journal of Research, which presents complete papers reporting technical investigations; the Technical News Bulletin, which presents summary and preliminary reports on work in progress; and Basic Radio Propagation Predictions, which provides data for determining the best frequencies to use for radio communications throughout the world. There are also five series of nonperiodical publications: The Applied Mathematics Series, Circulars, Handbooks, Building Materials and Structures Reports, and Miscellaneous Publications.

Information on the Bureau's publications can be found in NBS Circular 460, Publications of the National Bureau of Standards ($1.00). Information on calibration services and fees can be found in NBS Circular 483, Testing by the National Bureau of Standards (25 cents). Both are available from the Government Printing Office. Inquiries regarding the Bureau's reports and publications should be addressed to the Office of Scientific Publications, National Bureau of Standards, Washington 25, D. C.

NBS