3341

NATIONAL BUREAU OF STANDARDS REPORT

3341

CALORIMETER TESTS OF A PROTOTYPE 1/2 HP
FIVE-CYLINDER RADIAL COMPRESSOR

by

Minoru Fujii
C. W. Phillips
P. R. Achenbach

Report to
Office of The Quartermaster General
Department of the Army

NBS

U. S. DEPARTMENT OF COMMERCE
NATIONAL BUREAU OF STANDARDS

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NATIONAL BUREAU OF STANDARDS REPORT

NBS PROJECT		NBS REPORT
1003-20-4832	June 11, 1954	3341

Calorimeter Tests of a Prototype 1/2 HP
Five-Cylinder Radial Compressor

by

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to

Office of The Quartermaster General
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U. S. DEPARTMENT OF COMMERCE
NATIONAL BUREAU OF STANDARDS

CALORIMETER TESTS OF A PROTOTYPE 1/2 HP FIVE-CYLINDER RADIAL COMPRESSOR

by

Minoru Fujii, C. W. Phillips, and P. R. Achenbach

Abstract

Calorimeter tests were made of a prototype five-cylinder radial compressor submitted by Longstreth Enterprises to determine the suitability of this design to military applications. Special features of the compressor included a magnesium housing, five-cylinder radial construction, overhung crankshaft, special gasketing, countersunk bolts with Allen heads for the housing, and continuous circulation of oil through the compressor without the use of a sump. The calorimetric measurements indicated a linear change of capacity with speed and practically constant volumetric efficiency in the speed range up to 2300 rpm. The capacity of the compressor was one-half ton at a suction pressure of 14 psig, a discharge pressure of 117 psig and a speed of 1750 rpm. The horsepower per ton of refrigeration was found to be somewhat higher and the volumetric efficiency somewhat lower than for slow-speed compressors on similar duty, but these deviations might not be prohibitive in view of possible savings in space and weight. The magnesium housing became perforated with many small holes during the course of the tests, probably the result of corrosion of the magnesium by dichlorodifluoromethane. Means were not provided for mounting the compressor and vibration was excessive at speeds above 2300 rpm. Although the specimen might have potential value for military applications because of its small size and light weight, further development would be required to make it practical.

I. INTRODUCTION

At the request of the Office of The Quartermaster General, a study was made to determine the performance characteristics of a prototype refrigeration compressor submitted by Longstreth Enterprises of La Jolla, California. The prototype compressor incorporated a number of non-conventional features including a magnesium housing, five-cylinder radial construction, overhung crankshaft, special gasketing, and countersunk bolts with Allen heads for the housing. The purpose of this investigation was to determine the suitability of this design to military purposes.

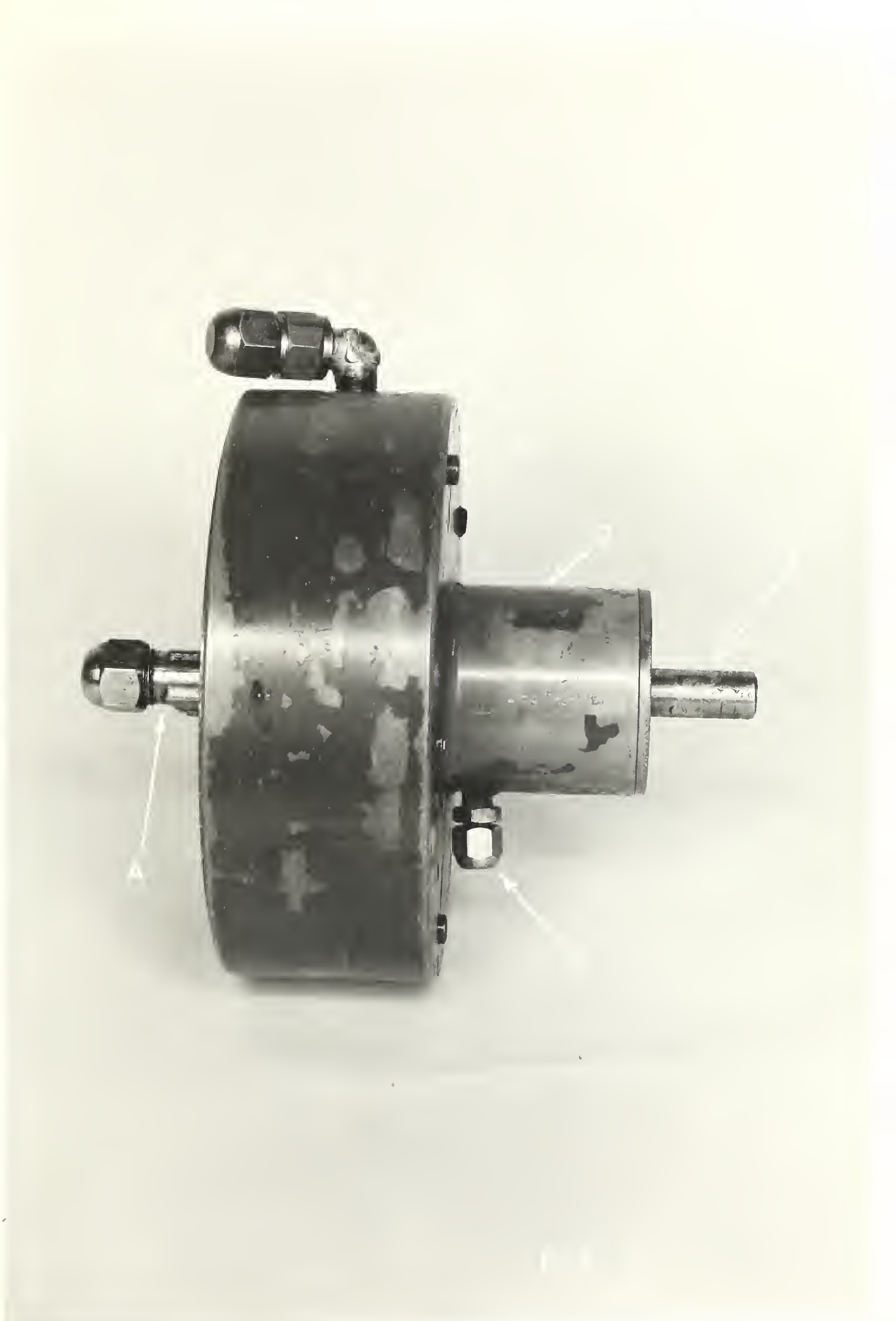
II. Description of Test Specimen

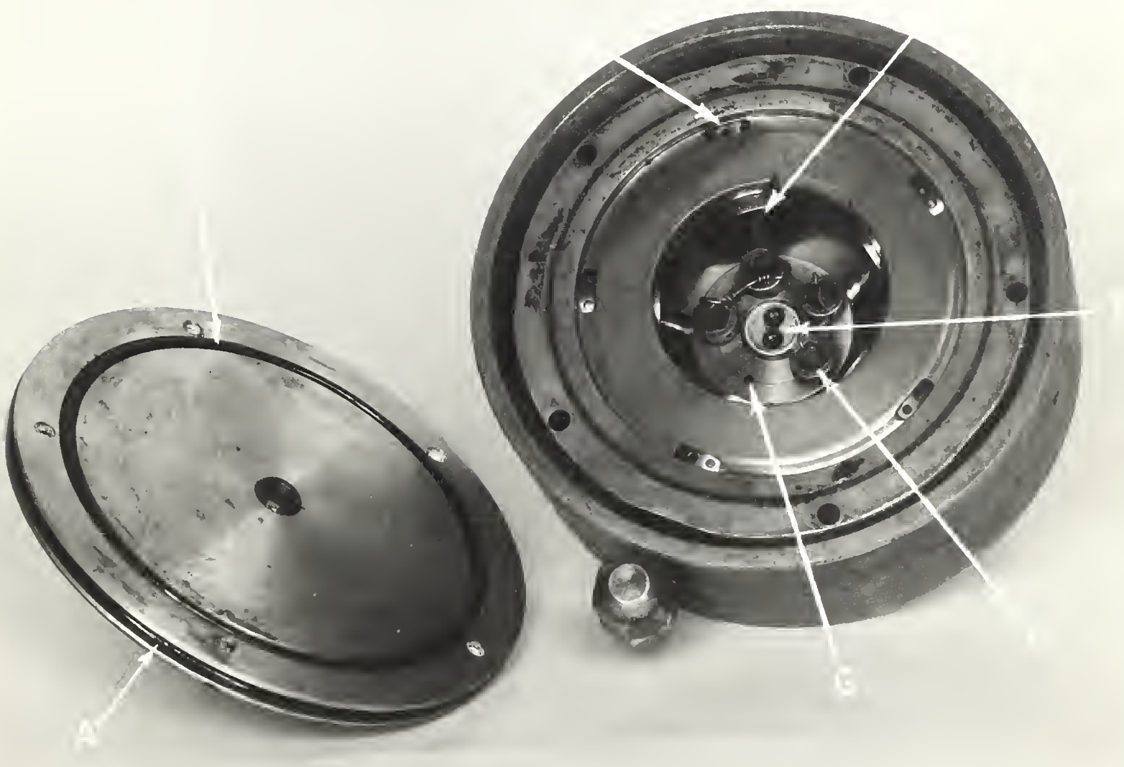
The specimen compressor was cylindrical in shape without fins. It was made of magnesium and weighed twelve pounds. An oil separator was provided as a separate accessory for the compressor. Fig. 1 shows an exterior view of the assembled unit. Fittings A, B, and C are the suction gas inlet, discharge gas outlet and oil return from the oil separator respectively. Hub D is the bearing housing and E is the shaft.

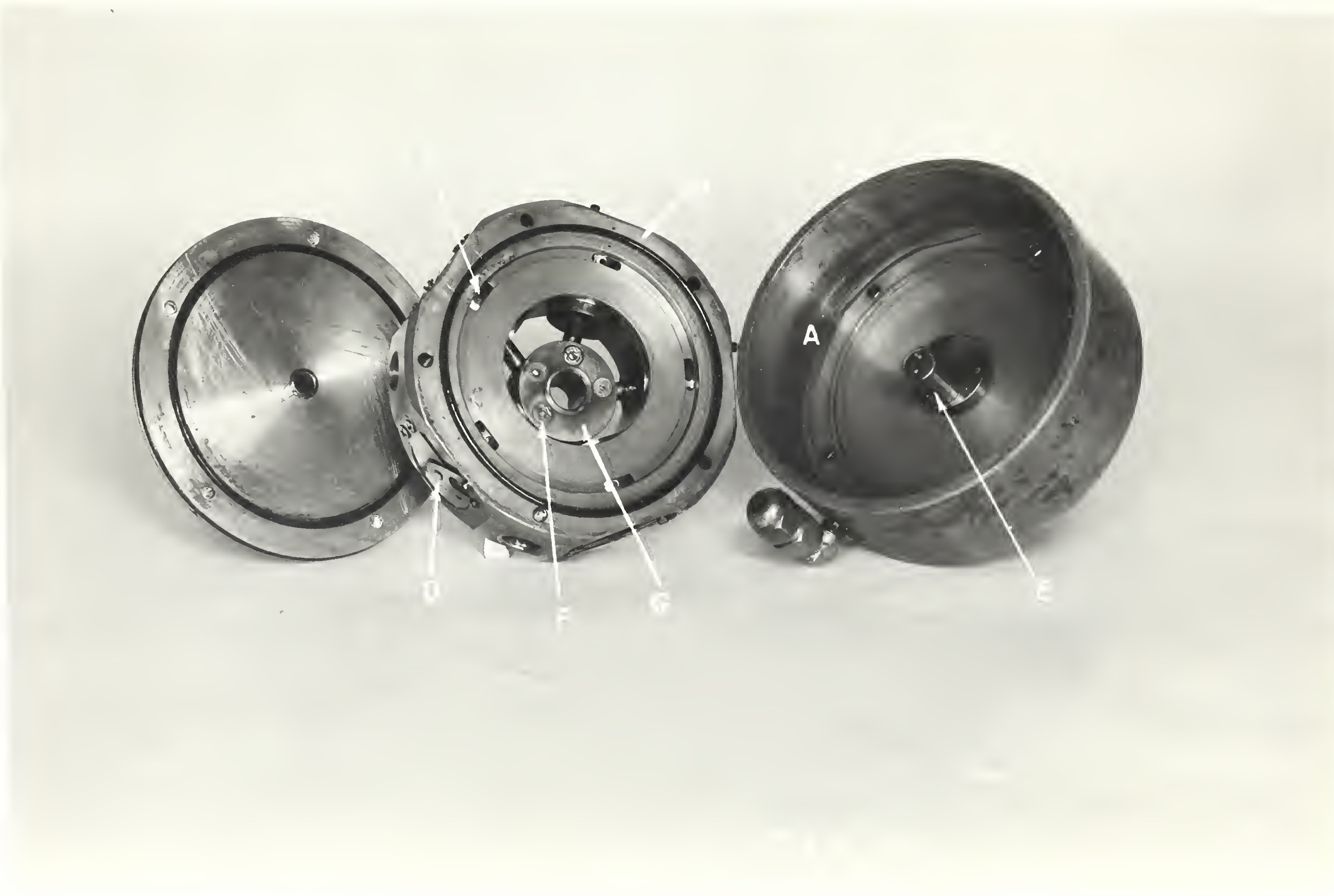
Fig. 2 shows an interior view of the compressor with the round cover plate opposite the hub removed from the compressor housing. This plate was fastened to the compressor housing with 5 Allen head bolts screwed into tapped blind holes. Synthetic rubber gaskets of the O-ring type were used at A and B to seal the casing and to separate the high pressure side and the low pressure side of the compressor. The five cylinders discharged into a common annular space just inside the periphery of the housing and the suction gas entered the crankcase at the center of the compressor. The lower end of one of the five pistons, which were arranged radially in one plane, is visible at D. The master connecting rod is shown at G and the other four link connecting rods were attached to it with link rod pins as typified at F. The end of the crankshaft is shown at E. Refrigerant gas was admitted to each cylinder from the crankcase through the port indicated at C and 20 holes around each cylinder sleeve which were uncovered at the bottom of the piston stroke. Three of these holes are visible through the port at C.

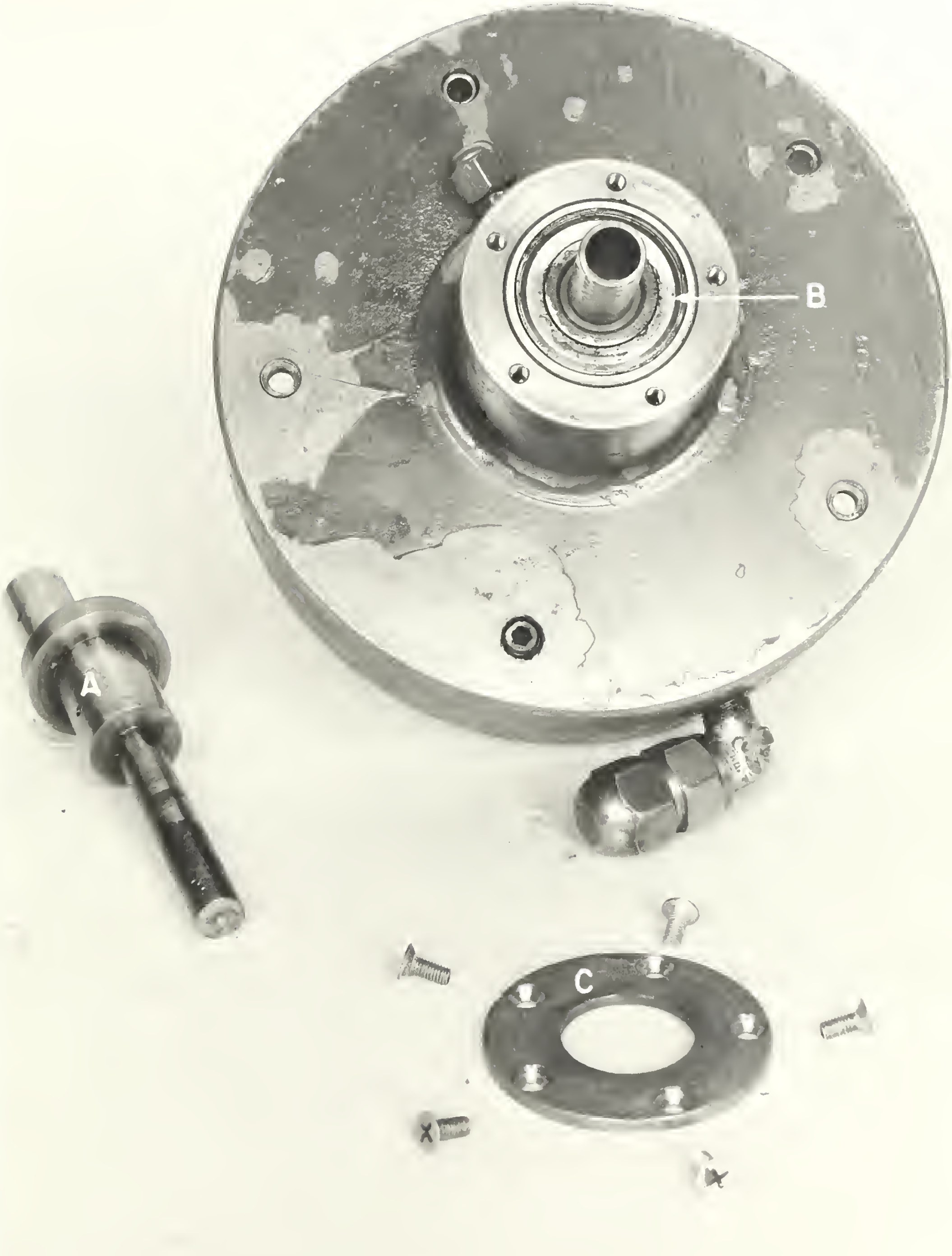
Fig. 3 shows an interior view when the cylinder block was taken out of the housing. A. The "O"-ring gasket B and suction ports C on the hub side are shown in this view. The crank pin is shown at E. The discharge valves were reed valves of the cantilever type, one of which is visible at D. In this figure F and G show the location of one of the link connecting rods and the master connecting rod, respectively.

Fig. 4 shows the journal bearing for the crankshaft at A, retainer C, and thrust ball bearing B. One rubber "O"-ring and


A


A
G


A
D
F
G
E


B
A
C

one graphite sleeve gasket combined with a spring-loaded "O"-ring were used for sealing the shaft. These parts are not shown in Fig. 4.

Forced-feed lubrication was provided utilizing the pressure differential between the oil separator and the crankcase to return the oil from the discharge line. The oil was fed to the journal bearing from the oil return, thence through a small hole in the shaft to the crank pin and finally to each cylinder through the five connecting rods.

No means of mounting was provided for the test specimen. For test purposes the compressor was mounted on a wooden block which was cut with a semi-cylindrical surface to fit the bottom half of the compressor. Steel straps were used to secure the compressor to the wood block.

Drawings and other data submitted by Longstreth Enterprises gave the principal dimensions and other physical characteristics which are listed as follows:

Table 1

Principal Dimensions and Characteristics of Specimen Compressor

Compressor manufacturer	Longstreth Enterprises La Jolla, California
Type	Reciprocating, single stage, radial
Number of cylinders	Five
Bore*	1.25 in.
Stroke*	0.75 in.
Displacement*	4.601 cu. in. per revolution
Method of lubrication	Force feed
Oil	12 fluid ounces, 300 Viscocity refrig. oil
Type of valves (discharge)	Flapper
Manufacturer's model	S-L-201
Dimensions:	
Total length of compressor	4 1/2 in.
Diameter	7 in.
Capacity	> 6000 Btu per hour
Power	1/2 Horsepower
Direction of rotation	Clockwise or counter clockwise
Design internal pressure	300 lb. per sq. in.
Upper temperature limit for discharge gas	200°F
Upper and lower temperature limit for suction gas	35°F, -20°F
Upper and lower pressure limit for suction gas	32 psig, 25" vac. Hg

*The measured bore and stroke were 1.375 in. and $\frac{49}{64}$ in. respectively, corresponding to a displacement of 5.681 cu. in. per revolution.

III. Testing Method and Procedure

A secondary refrigerant calorimeter was used to determine the capacities of the compressor. Heat load was imposed on the secondary refrigerant calorimeter by electric heaters. Suitable controls were used for maintaining the desired suction pressure, head pressure, suction temperature and other conditions. Calibrated pressure gages and watthour meters were used for observing pressures and energy consumption, respectively. Thermocouples were attached to the refrigerant tubing and compressor at selected locations for observing temperatures. The copper tubing was covered with hairfelt insulation for a distance of approximately 6" on each side of the stations of temperature measurement. Thermocouples enclosed in stainless steel wells were inserted in the suction line 6" from the compressor shell and 6" from the expansion valve to measure refrigerant temperatures. A water-cooled condenser was used for condensing the refrigerant.

A shunt-wound, direct-current motor was used to drive the compressor for most of the tests. Pulleys and belts of different sizes and a bank of rheostats in the armature circuit of this motor were used for a coarse adjustment of the motor speed. A governor, connected to the motor shaft, actuated a slide-type rheostat in the shunt field circuit of the motor to obtain more precise control of the compressor speed. Although the current flow through the armature circuit and voltage at the motor terminals were measured the power consumption of the compressor was not determined since the efficiency of the D.C. motor at various speeds was not measured. Absolute power consumption at a speed of about 1700 rpm was determined by means of a 1/2 horsepower A.C. motor as described later.

Three kinds of tests were made with the specimen compressor. For one series of tests, the compressor speed, discharge pressure and suction temperature were maintained constant, while the suction pressure was varied. For the second series of tests, suction pressure, head pressure and suction temperature were maintained constant while the compressor speed was varied. A special test was made with a 1/2 H.P. A.C. motor directly connected to the specimen compressor. The motor had previously been calibrated by means of a dynamometer.

From this calibration the brake horsepower of the compressor was determined for a limited range of suction pressure. Special tests were made at a compression ratio of 9.29 using the D.C. motor drive and at a compression ratio of 11.16 using the 1/2 hp A.C. motor.

The compressor capacity for each test was computed by means of the following formula, taken from ASRE Standard No. 23-R, entitled "Methods of Rating and Testing Refrigerant Compressors":

$$Q = \frac{h_{g1} - h_{f1}}{h_{g2} - h_{f2}} (Qh + Qe) \qquad (1)$$

where h_{f1} = heat content of refrigerant liquid at a temperature corresponding to the pressure of refrigerant vapor leaving compressor, Btu per lb.

h_{f2} = heat content of refrigerant liquid entering the expansion valve, Btu per lb.

h_{g1} = heat content of refrigerant vapor entering compressor, Btu per lb.

h_{g2} = heat content of refrigerant vapor leaving calorimeter, Btu per lb.

Q = compressor capacity, Btu per hr.

Q_h = calorimeter heat input, Btu per hr.

Q_l = heat leakage into calorimeter, Btu per hr.

A secondary method was employed to substantiate the compressor capacities computed by the above formula from the calorimeter data. This method consisted of measuring the volume flow of refrigerant through the calorimeter circuit per unit time. This was accomplished by using two receiving tanks in the refrigerant circuit and alternately filling one tank with liquid from the condenser, while emptying the other tank into the evaporator. The two tanks were calibrated volumetrically, and the time required to fill a certain fraction of the volume of one tank was observed. From this elapsed time the volume-flow per hour was computed. The following formula, taken from ASRE Standard No. 23-R, was then used to compute the compressor capacity for each test:

$$Q = VD \quad (1 - 0.018\, wd)\,(h_{g1} - h_{f1}) \\ - 0.44\, w\,(t_{f1} - t_{g2})$$

where

d = density of liquid refrigerant, for temperature at which flow is measured, lb per cu ft

h_{f1} = heat content of refrigerant liquid at a saturation temperature corresponding to the pressure of refrigerant vapor leaving compressor, Btu per lb.

h_{g1} = heat content of refrigerant vapor entering compressor, Btu per lb.

Q = compressor capacity, Btu per hr.

t_{f1} = saturation temperature corresponding to pressure of refrigerant vapor leaving compressor, deg F

t_{g1} = temperature of refrigerant vapor entering compressor, deg F

V = flow of mixture of liquid refrigerant and oil, cu ft per hr.

w = weight of oil per unit weight of mixture of liquid refrigerant and oil, lb per lb.

Tables in ASRE Circular No. 12 on the thermodynamic properties of dichlorodifluoromethane (F-12) were used to determine the heat contents called for in the above formula. A value of 0.02 for w in the formula was used, based on a single determination of the oil content of the refrigerant.

In preparation for each test the compressor was run for a period of more than two hours before the test period was started, in order to allow the calorimeter system to come to steady state conditions. Observations were made every twenty minutes for at least two hours under steady conditions of temperature, pressure, heat input and speed.

Tests were made with only one direction of rotation of the compressor which was counter clockwise as viewed from the hub side of the compressor.

IV. Test Results

The results obtained during the calorimeter tests of the compressor specimen are summarized in Tables 2 and 3 and various relationships are shown graphically in Fig.5 to 10, inclusive.

Fig.5 shows the relationship between compressor capacity and speed for the range from 800 to 2300 rpm. During this series of tests the suction pressure was maintained at 4.5 psig and the discharge pressure at 117 psig corresponding to a compression ratio of 6.86. The suction gas temperature was maintained at 65°F. The volumetric efficiency was constant for all practical purposes throughout this range of speed and the straight line drawn through the observed values, if extrapolated, shows that the capacity becomes zero almost simultaneously with the speed. This indicates that the effectiveness of the valve action was equally good at all speeds up to 2300 rpm.

TABLE 2

CAPACITY OF LONGSTRETH COMPRESSOR AT VARIOUS SPEEDS
WITH CONSTANT COMPRESSION RATIO

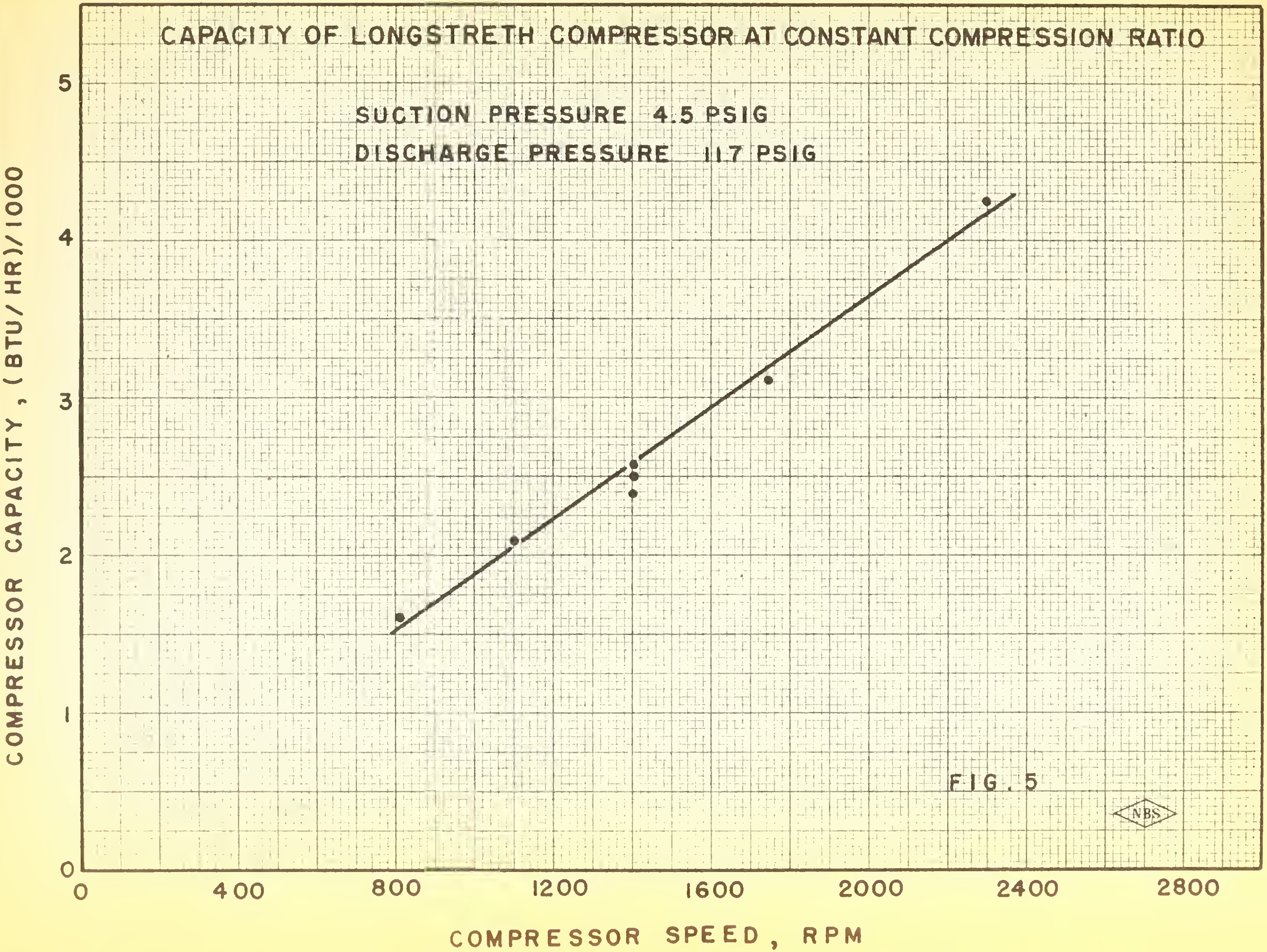
Test No.		1	2	3	4	5	6	7	8	9	10
Compressor Speed	rpm	510	790	1110	816	1103	1400	1404	1406	1743	2306
Suction Pressure	psig	21.1	21.2	21.3	4.5	4.5	4.4	4.5	4.5	4.5	4.5
Suction Pressure	psia	35.8	35.9	36.0	19.2	19.2	19.1	19.2	19.2	19.2	19.2
Discharge Pressure	psig	126	125	126	117	117	117	117	117	117	117
Discharge Pressure	psia	140.7	140.6	140.7	131.7	131.7	131.7	131.7	131.7	131.7	131.7
Compression Ratio		3.93	3.92	3.91	6.86	6.86	6.86	6.86	6.86	6.86	6.86
Refrigerant Temp at Compressor Inlet	°F	64.7	64.7	64.4	67.4	67.3	65.2	64.7	65.1	64.4	65.1
Refrigerant Temp at Exp. Valve Inlet	°F	77.8	83.9	88.9	76.3	79.0	81.7	79.9	82.3	85.6	91.1
Compressor Capacity	Btu/hr	1970	3590	4890	1600	2090	2390	2570	2500	3110	4250
Deviation of Capacity Check	%	5.6	-0.4	0.8	4.5	-1.7	4.6	4.1	6.3	6.2	5.7
Motor Input	watts	655	1019	1129	601	819	893	864	864	986	1263
Volumetric Efficiency	%	44.6	49.6	50.7	41.0	40.8	35.7	38.3	37.2	37.6	38.7
Temperature of Bearing Housing	°F	170.2	168.8	182.4	179.3	177.9	199.7	201.8	199.8	210.3	224.2
Temperature of Compressor Shell	°F	187.7	189.9	203.7	199.1	202.8	223.7	225.9	223.1	235.2	246.5
Ambient Temperature	°F	82.3	80.2	87.2	78.2	76.5	77.9	78.8	80.1	85.7	81.5
Refrigerant Flow Rate	cfm	0.75	1.29	1.85	1.10	1.48	1.64	1.77	1.72	2.15	2.93
Compressor Displacement	cfm	1.68	2.60	3.65	2.68	3.63	4.60	4.62	4.62	5.72	7.58

TABLE 3

CAPACITY OF LONGSTRETH COMPRESSOR AT VARIOUS COMPRESSION RATIOS WITH CONSTANT SPEED

Test No.		1	2	3	4	5	6	7	8[a]
Compressor Speed	rpm	1743	1751	1752	1762	1757	1757	1750	1690
Suction Pressure	psig	4.5	9.2	14.7	21.1	29.0	37.0	5.0	-3.0
Suction Pressure	psia	19.2	23.9	29.4	35.8	43.7	51.7	19.7	11.7
Discharge Pressure	psig	117	117	117	117	117.5	117	168.4	117
Discharge Pressure	psia	131.7	131.7	131.7	131.7	132.2	131.7	183.1	131.7
Compression Ratio		6.86	5.51	4.48	3.68	3.03	2.55	9.29	11.16
Refrigerant Temp at Compressor Inlet,	°F	64.4	65.6	65.0	64.5	65.1	65.0	64.8	65.0
Refrigerant Temp at Exp. Valve Inlet,	°F	85.6	87.9	92.8	90.7	92.7	91.2	87.2	79.9
Compressor Capacity	Btu/hr	3110	4410	5900	7950	10320	13160	1830	1220
Deviation of Capacity Check	%	6.2	3.1	5.3	6.2	6.4	5.5	-5.7	7.1
Motor Input	watts	986	1055	1271	1462	1644	1848	952	-
Volumetric Efficiency	%	37.6	42.9	46.7	50.9	54.2	57.8	24.5	25.4
Temperature of Bearing Housing	°F	210.3	209.0	200.4	189.6	178.9	166.3	234.5	210.0
Temperature of Compressor Shell	°F	235.3	232.6	222.6	210.1	200.6	188.1	257.8	227.6
Ambient Temperature	°F	85.7	78.9	80.8	81.5	82.5	79.4	78.6	72.5
Refrigerant Flow Rate	cfm	2.15	2.47	2.69	2.95	3.13	3.34	1.41	1.41
Compressor Displacement	cfm	5.72	5.76	5.76	5.79	5.78	5.78	5.76	5.56

[a] Compressor driven by 1/2 hp A.C. motor direct-connected



CAPACITY OF LONGSTRETH COMPRESSOR AT CONSTANT COMPRESSION RATIO
SUCTION PRESSURE 4.5 PSIG
DISCHARGE PRESSURE 117 PSIG
COMPRESSOR CAPACITY, (BTU/HR)/1000
COMPRESSOR SPEED, RPM
0
1
2
3
4
5
400
800
1200
1600
2000
2400
2800
NBS

FIG. 5

Fig 6 shows that the relationship between compressor capacity and suction pressure was linear for a constant discharge pressure. This figure reveals that the capacity of 6000 Btu/hr claimed by the manufacturer was attained at a suction pressure of 14.3 psig corresponding to an evaporator temperature of about 10°F. If the capacity curve in Fig 6 were extrapolated as a straight line the indicated capacity would be zero at a suction pressure of 10 in. H_g vacuum.

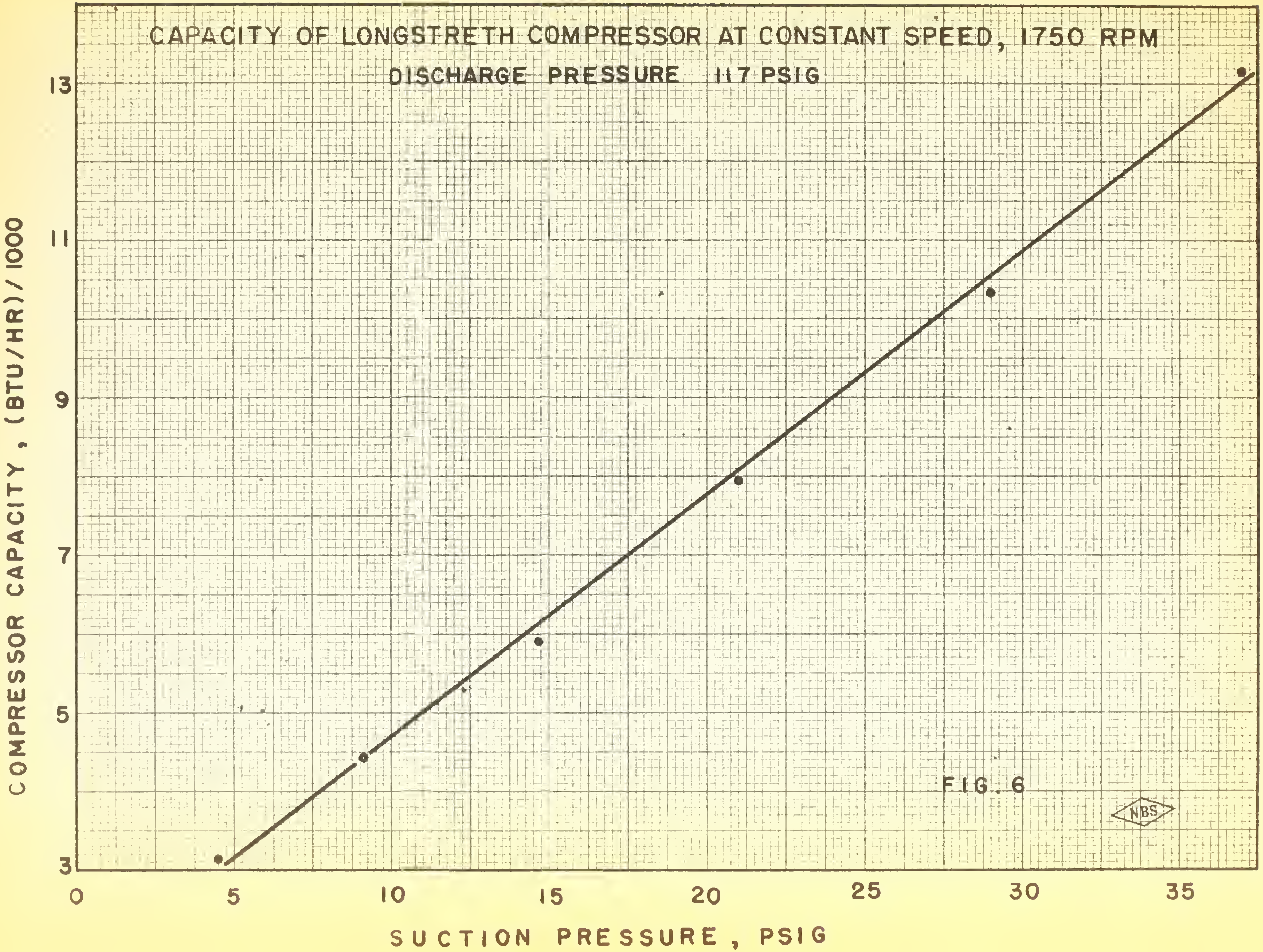
Fig 7 shows the power consumption of the Longstreth compressor for a limited range of suction pressure as determined by using a calibrated 1/2 hp A.C. motor as a direct drive. This curve shows that a 1/2 hp motor would only be adequate for driving this compressor at suction pressures of zero gage or lower. Using the data from Fig 7 and a minor extrapolation of the curve in Fig 6 it can be shown that the brake horsepower per ton of refrigeration for the specimen compressor was about 3.75 for a suction pressure of 2.5 psig and a discharge pressure of 117 psig. By comparison, the brake horsepower per ton observed on a larger slow-speed compressor was 2.8 for the same compression ratio.

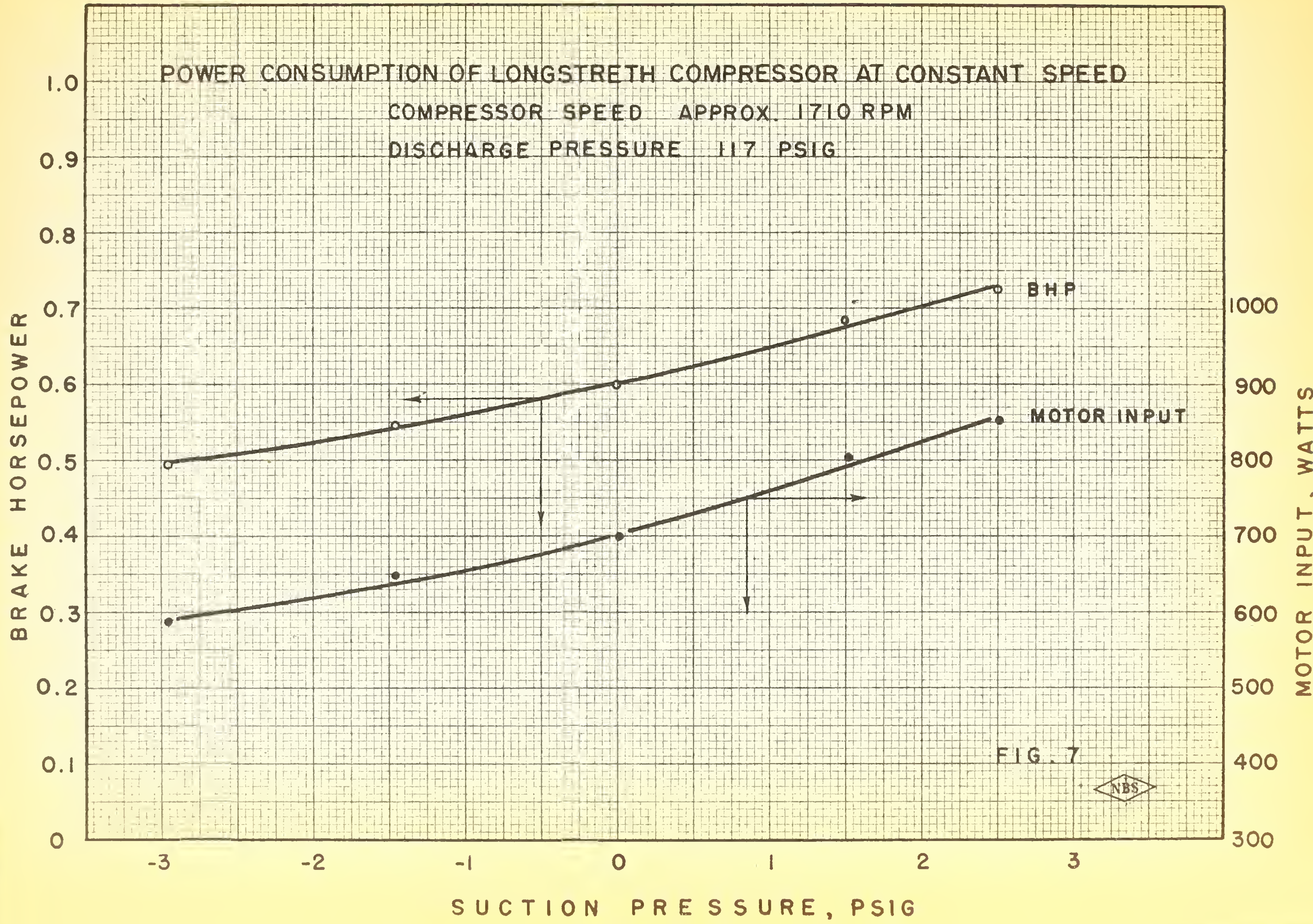
Fig 8 shows the temperatures of the bearing housing and the compressor shell for a range of compressor speed. These temperatures increased in a linear relationship with compressor speed at constant ambient temperature. The scattering of the plotted values at a speed of 1750 rpm resulted from the variation in suction pressure selected for different tests at this speed.

The relation between volumetric efficiency and compression ratio at constant speed is shown in Fig 9. The volumetric efficiency ranged from about 38% for a compression ratio of 7 to about 58% for a compression ratio of 2.5. The volumetric efficiency of the Longstreth compressor at a compression ratio of 5 was comparable to that of another prototype compressor tested at the same speed during an earlier investigation, but it is as much as 20% lower than for slow-speed compressors designed for similar capacity.

Fig 10 shows the volumetric efficiency plotted against compressor speed for a fixed compression ratio. Fig 10 corroborates the conclusion drawn from Fig 5 that the volumetric efficiency was practically constant for the range of speed from 800 rpm to 2300 rpm.

According to information received from the manufacturer, the compressor was designed to operate at speeds in the range from 1000 rpm to 4000 rpm. At speeds higher than 2300 rpm, however, vibration of the specimen was so severe that the refrigerant lines broke repeatedly and fittings developed leaks. A better means of mounting


CAPACITY OF LONGSTRETH COMPRESSOR AT CONSTANT SPEED, 1750 RPM
DISCHARGE PRESSURE 117 PSIG
COMPRESSOR CAPACITY, (BTU/HR)/1000
13
11
9
7
5
3
0
5
10
15
20
25
30
35
SUCTION PRESSURE, PSIG
NBS
FIG. 6


POWER CONSUMPTION OF LONGSTRETH COMPRESSOR AT CONSTANT SPEED
COMPRESSOR SPEED APPROX. 1710 RPM
DISCHARGE PRESSURE 117 PSIG
BHP
MOTOR INPUT
BRAKE HORSEPOWER
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
MOTOR INPUT, WATTS
300
400
500
600
700
800
900
1000
SUCTION PRESSURE, PSIG
-3
-2
-1
0
1
2
3
NBS
FIG. 7

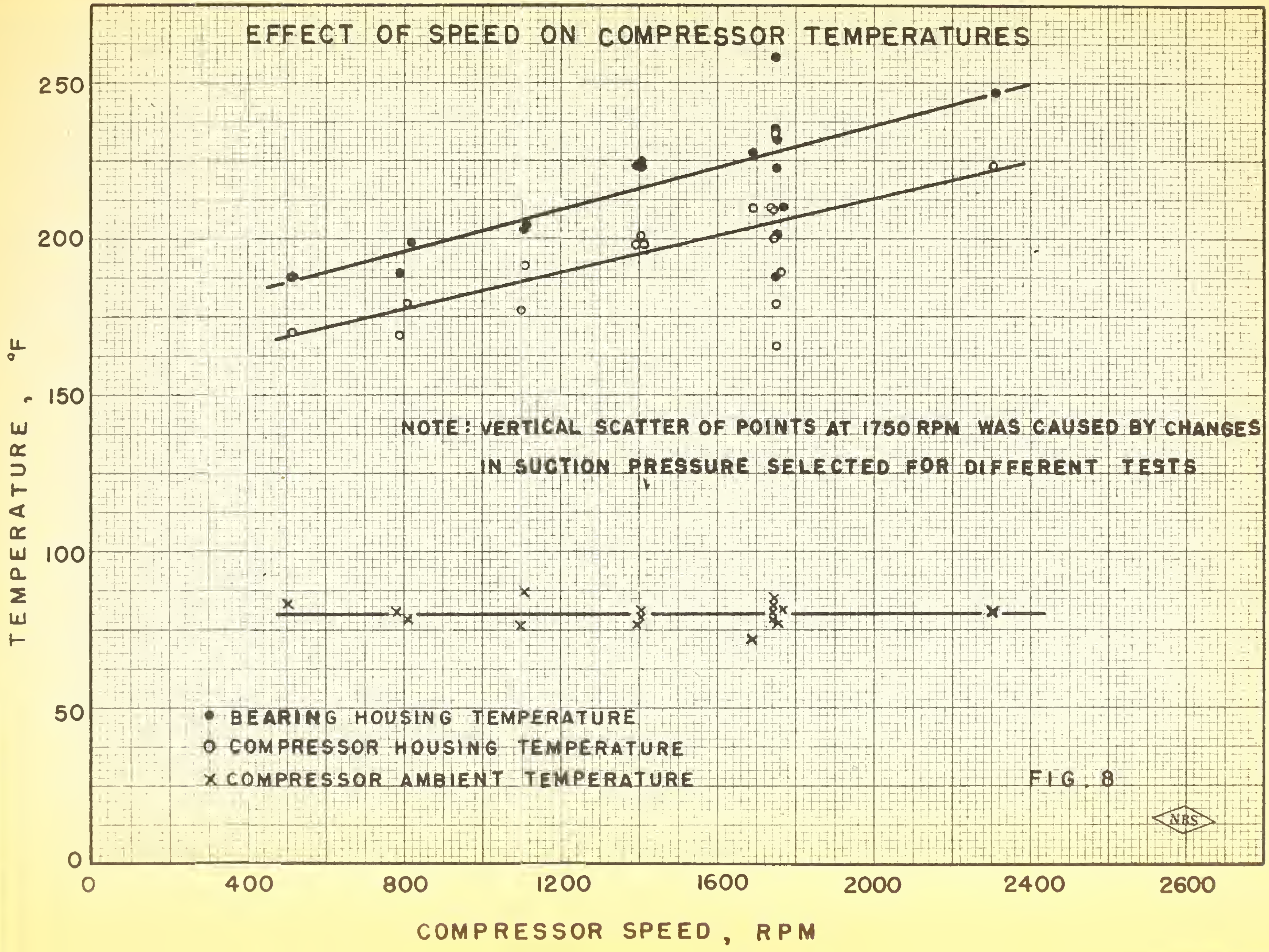

EFFECT OF SPEED ON COMPRESSOR TEMPERATURES
NOTE: VERTICAL SCATTER OF POINTS AT 1750 RPM WAS CAUSED BY CHANGES IN SUCTION PRESSURE SELECTED FOR DIFFERENT TESTS
• BEARING HOUSING TEMPERATURE
O COMPRESSOR HOUSING TEMPERATURE
X COMPRESSOR AMBIENT TEMPERATURE
TEMPERATURE, °F
COMPRESSOR SPEED, RPM
0
50
100
150
200
250
400
800
1200
1600
2000
2400
2600
NBS

FIG. 8

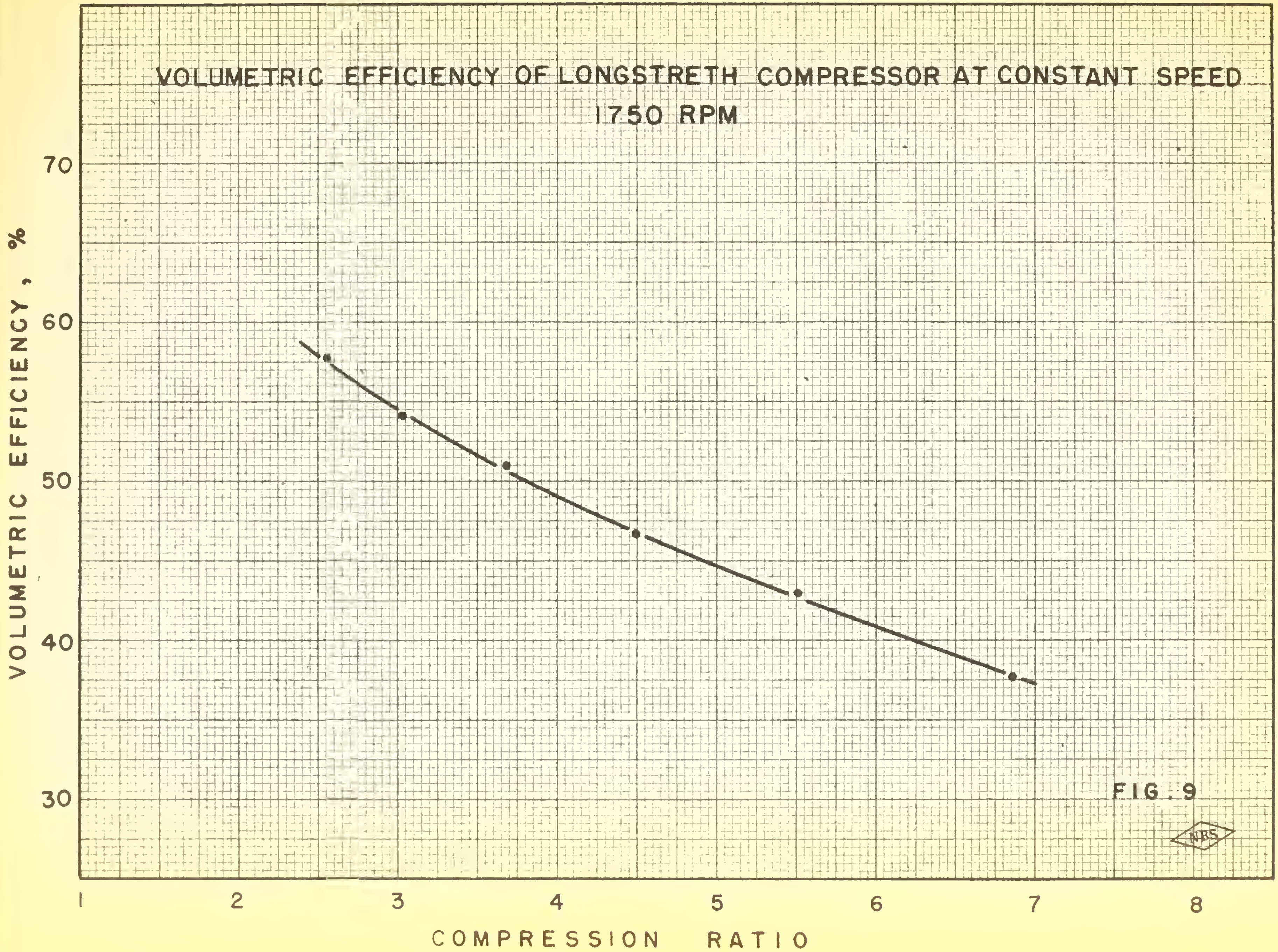

VOLUMETRIC EFFICIENCY OF LONGSTRETH COMPRESSOR AT CONSTANT SPEED
1750 RPM
70
60
50
40
30
VOLUMETRIC EFFICIENCY, %
1
2
3
4
5
6
7
8
COMPRESSION RATIO
NBS

FIG. 9

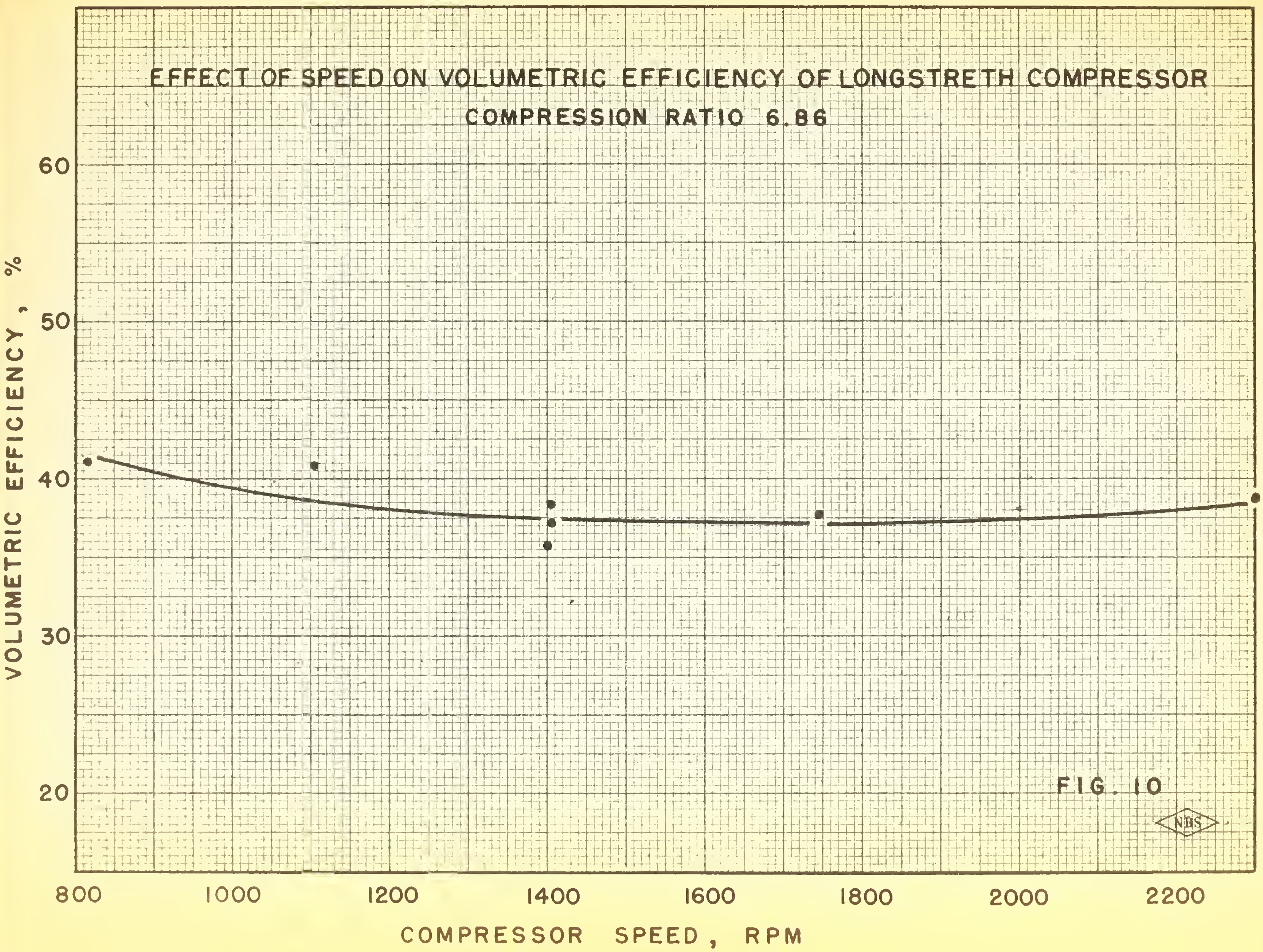

EFFECT OF SPEED ON VOLUMETRIC EFFICIENCY OF LONGSTRETH COMPRESSOR
COMPRESSION RATIO 6.86
VOLUMETRIC EFFICIENCY, %
60
50
40
30
20
800
1000
1200
1400
1600
1800
2000
2200
COMPRESSOR SPEED, RPM

FIG. 10

the compressor was being constructed when a leak test revealed refrigerant loss through a large number of pores in the housing of the compressor. The pores were found in the cylindrical surface of the magnesium compressor housing and could barely be observed by a binocular which had a magnification of one hundred. The leakage through each pore was minute, but the total amount of leakage was considerable. It was not determined whether the pores were present in the magnesium casting originally or whether they were caused by corrosion.

V. Discussion and Conclusions

The prototype compressor appeared to have potential value to the Office of The Quartermaster General because of its relatively light weight and small size. There were, however, some disadvantages. The power consumption per unit capacity was relatively high. This characteristic may have been due in part to the absence of suction valves. The suction ports were not uncovered to admit refrigerant to the cylinders until the piston reached the bottom of the cylinder thus increasing the amount of work done during each suction stroke. The vibration of the prototype was excessive. The absence of any means of mounting the compressor may have been responsible in part for this excessive vibration, but it is believed that better balancing of the compressor is also needed.

The Data Book of the American Society of Refrigerating Engineers recognizes that Freon-12 is corrosive to magnesium. It is probable, therefore, that corrosion caused the porosity of the magnesium compressor housing that was observed at the end of the tests.

The compressor housing and shaft bearing operated at relatively high temperatures even when air was circulated over the compressor by an auxiliary fan. The compressor was not provided with fins for extended cooling surface. The wooden supports used for mounting the prototype during the tests probably reduced the heat transfer from the lower half of the housing somewhat.

An oil separator is an essential accessory to this compressor because lubrication of all moving parts depends on a steady flow of oil from the compressor, through the separator, and back to the oil inlet connection on the bearing housing.

The design of the compressor was considered well-suited to mass production because most of the parts could be made with automatic machine tools.

THE NATIONAL BUREAU OF STANDARDS

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