NATIONAL BUREAU OF STANDARDS REPORT

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PERFORMANCE OF A DEVELOPMENT MODEL GENERAL ELECTRIC 3 HP CONSOLE AIR CONDITIONER FOR VANS AND PREFABRICATED BUILDINGS

by

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NATIONAL BUREAU OF STANDARDS REPORT

NBS PROJECT **NBS REPORT**

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Performance of a Development Model 3 HP Console Air Conditioner for Vans and Prefabricated Buildings

manufactured by

General Electric Company

by

Henry Karger
C. W. Phillips
P. R. Achenbach

to

Engineer Research and Development Laboratories
Fort Belvoir, Virginia

NBS

PERFORMANCE OF A DEVELOPMENT MODEL GENERAL ELECTRIC 3 HP CONSOLE AIR CONDITIONER FOR VANS AND PREFABRICATED BUILDINGS

ABSTRACT

Performance tests were made of a development model of a sectional, package-type, air conditioning unit manufactured by General Electric Company for the Research and Development Laboratories, Fort Belvoir, as a basis for further development of air conditioning equipment for vans and prefabricated buildings. Comparisons of performance were made with the unit entirely in the conditioned space, entirely outside the conditioned space, and with the unit divided into two parts - the evaporator section being mounted in the conditioned space and the condenser section in the ambient air - with a minimum of ductwork employed in each case. The total net cooling capacity ranged from 18,500 Btu/hr to 20,040 Btu/hr at ASRE Standard rating conditions for self-contained units with the capacity being 4 to 8 percent higher with the divided installation of the evaporator and condenser sections. Leakage of outside air was a significant factor that affected the dehumidifying capacity and total net cooling capacity of the unit considerably. The performance factor of the unit in Btu/hr/watt was lower than the minimum value set in the Federal Specification for self-contained air conditioners for normal closure of the fresh air dampers.

I. INTRODUCTION

In accordance with a request from the Engineer Research and Development Laboratories, Fort Belvoir, Virginia, dated September 28, 1951, tests of development models of two sectional, package-type, air conditioning units were made to determine their performance as a basis for developing air conditioning equipment for vans and prefabricated buildings for Theater of Operation use. The two units were designed for capacities in the range from 18,000 to 22,000 Btu/hr. This report presents the results obtained on a specimen manufactured by General Electric Company of Bloomfield, New Jersey. Tests were made to determine net cooling capacities, the quantity of conditioned air circulated, the quantity of air circulated through the condenser, and the starting current of the unit.

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

Capacity tests were requested at standard rating conditions of the American Society of Refrigerating Engineers with outdoor temperatures of 95°F dry bulb, 75°F wet bulb, and indoor temperatures of 80°F dry bulb, 67°F wet bulb, and at high ambient temperature conditions of 125°F dry bulb, 85°F wet bulb and indoor temperatures of 90°F dry bulb, 80°F wet bulb.

II. DESCRIPTION OF TEST SPECIMEN

The specimen air conditioning unit was identified as follows:

(NBS Test Specimen 76-52)
General Electric Company, Bloomfield, N. J.
Model 21FF20A1
Serial Number 21976976

The air conditioning unit was built for operation on alternating current, 208 volts, 3-phase, 60 cycle, 4-wire. The sectional construction of the unit permitted operation in three different ways: the entire unit in the space to be cooled with ducts connecting the condenser section to the outdoors; the entire unit on the outside of the space to be cooled with ducts connecting the evaporator section to the conditioned space; or the unit separated into two parts with the evaporator section placed inside the conditioned space and the condenser section placed outside the conditioned space. The sectional construction is evident in Fig. 1 to 4. The bottom section contained the motor and compressor; the second section contained the condenser; while the third section contained the condenser fan. These three sections were not built for ready separation. The condenser air entered the unit through the bottom section and was blown out through two outlets in the third section as shown in Fig. 2. The fourth section contained the evaporator fan, and the fifth or top section contained the evaporator coil. Fig. 3 and 4 are views of the air conditioning unit with all the face panels removed, and the five different sections of the unit can be seen very clearly. The compressor was of the open type, direct-connected to an induction motor with nominal full load speed of 1725 rpm as shown in Fig. 4. A centrifugally-operated clutch permitted the motor to start without the compressor load.

The electrical controls for operation of the air conditioning unit were mounted on the third section of the unit, as can be seen from Fig. 1. The controls consisted of an off-on switch, located in the center of the control panel, a four-position switch located on the right-hand side of the control panel, and a thermostat with its adjusting knob located on the left-hand side of the control panel. The four-position switch was marked "1", "2", "3", and "4".

According to the instructions furnished by the manufacturer, the unit should be started in the following sequence. With the selector switch in the No. 1 position, turning the disconnect switch to the "on" position would start the evaporator fan; then turning the selector switch to the No. 2 position would start the condenser fan. The compressor would start by turning the selector switch to No. 3 position, and it would then operate intermittently according to the demands of the thermostat of the unit. If it was desired to operate the compressor on an external control, such as a thermostat, this could be done with the selector switch in the No. 4 position.

A knob, which controlled the position of two sets of louvers, was located on the right-hand side of the fourth section of the unit. The control shaft on which the knob was mounted can be seen in Fig. 3. The two sets of louvers, one located behind the fresh air filter and the other behind the return air filter, controlled the amount of fresh air taken in by the unit by opening the fresh-air intake and at the same time closing the return-air intake from the cooled space.

Two centrifugal fans were used to circulate air through the condenser, the fan wheels being mounted on opposite ends of the shaft of the driving motor. A similar arrangement was employed for air circulation through the evaporator. These assemblies can be seen in Fig. 3 and 4. A thermostatic expansion valve was used as the refrigerant flow control.

The physical dimensions, weight and electrical data for the unit were as follows:

Height, in.	69
Width, in.	42
Depth of fourth section, including filter frames, in.	18-1/4
Depth of all other sections, in.	14
Weight, including wood shipping base, lbs.	646
Compressor Motor, HP	3
Condenser Fan Motor, HP	3/4
Evaporator Fan Motor, HP	1/3
Refrigerant charge, lbs. Freon-12	16

The unit was equipped with two air filters made of expanded aluminum mesh at the evaporator inlet and fresh air inlet, respectively. All framing and panels of the unit were made of aluminum.

III. TEST PROCEDURE

Circular No. 16-R, published by the American Society of Refrigerating Engineers, entitled "Methods of Rating and Testing Air Conditioners", was used as a general guide for the procedure to be used in determining the capacity of the air conditioning unit. This ASRE circular states that the capacity rating of self-contained air conditioning units shall be the net total room cooling effect in British Thermal Units per hour. The following apparatus was used to determine this net room cooling capacity.

A portable, 600 cu ft warehouse normally used for the refrigerated storage of various materials was used as a calorimeter. For the first series of tests the air conditioning unit was mounted in the opening normally used for the refrigerating unit for the warehouse. It was placed on the inside of the warehouse-calorimeter flush with the outside wall. The calorimeter was then sealed in such a manner that only the openings for the condenser air circuit and the fresh air opening communicated with the outside space. For the second series of tests the air conditioning unit was placed entirely on the outside of the calorimeter and short ducts were used on the evaporator air circuit into the calorimeter. For the third series of tests, the evaporator section of the air conditioning unit was separated from the condenser section and was placed on the inside of the calorimeter whereas the condenser section was placed on the outside of the calorimeter. The calorimeter was sealed so that only the fresh air opening of the evaporator section communicated with the outside of the calorimeter.

For each series of tests, the warehouse, with the air conditioning unit in place, was calibrated to determine the heat transmission per degree of temperature difference between the inside and outside, since this heat constituted part of the sensible heat load. Electric heaters were placed on the inside of the calorimeter to provide the remainder of the sensible heat load for the air conditioning unit. A humidifier consisting of an electric heating element immersed in water was also placed inside the calorimeter to provide the necessary humidity. The calorimeter was made as airtight as was considered necessary, and provisions were made to measure the amount of air which was introduced into or exhausted from the calorimeter by the air conditioning unit during operation. This was done by removing air from or introducing air into the calorimeter through a pipe by means of a blower. The air quantities were measured by a pitot tube placed inside of the pipe. The amount of air flowing through the pipe was regulated by a throttling valve so that the pressure difference between the inside and the outside of the

calorimeter was zero ± 0.002 in. W.G. Calibrated watthour meters were used to measure all electric energy consumption. Temperatures were measured by means of calibrated thermocouples using an electronic, constant-balance type potentiometer. Humidity measurements were made utilizing calibrated lithium-chloride-coated elements in conjunction with a micro-ammeter. Humidity measurements observed with the electric hygrometer were checked regularly with a 24" mechanical psychrometer. The condensate drained from the evaporator coil was collected as a check against the amount of water evaporated by the humidifier.

Measurements of the quantities of air circulated by the evaporator and condenser fans were made by means of a pitot tube located inside a smooth, round duct during separate tests with the air conditioning unit removed from the calorimeter. This duct was connected to the outlet of the evaporator or condenser air circuit by means of a plenum chamber and an exhaust blower was attached to the other end of the duct. The air flow through the duct was regulated by a throttling device so that the static pressure at the evaporator or condenser outlet opening in the cabinet was zero ± 0.002 in. W.G. with respect to the pressure in the test room to determine the free-air delivery of the fans.

IV. TEST RESULTS

A. Cooling Capacity

Determinations of the total net cooling capacity of the unit were made for two different sets of indoor and outdoor temperature and humidity conditions, for each of three different positions of the unit with respect to the calorimeter. Tests 1a and 1b were made with the unit entirely within the calorimeter, tests 2a and 2b were made with the unit on the outside of the calorimeter, and tests 3a and 3b were made with the unit divided as previously described. All tests except test 4a were made with the fresh air dampers closed, but not sealed. Thus the enthalpy difference between outdoor and indoor temperature conditions for any air that leaked by the damper in its closed position or through other openings in the air conditioner was not credited to the total net cooling capacity of the unit. A single test, numbered 4a, was made with the fresh air dampers sealed to determine the cooling capacity of the unit under these conditions. For this test the unit was divided, as for tests 3a and 3b. The desired test conditions, the average temperature conditions maintained during the tests, and the results are summarized in Table 1.

The total net cooling capacity, as shown in Table 1, is the sum of the net sensible and latent cooling effects. The

TABLE 1

Performance of General Electric Air Conditioner,
Model 21FF20A1, NBS Specimen 76-52

Desired Test Conditions

	a tests	b tests
Ambient Air Conditions		
Dry Bulb Temp., °F	95	125
Wet Bulb Temp., °F	75	85
Relative Humidity, %	40	20
Air Conditions at Evaporator Inlet		
Dry Bulb Temp., °F	80	90
Wet Bulb Temp., °F	67	80
Relative Humidity, %	51	65

Observed Test Conditions

Test No.	1a	2a	3a	4a	1b	2b	3b
Ambient Air Conditions							
Dry Bulb Temp., °F	95.5	94.7	94.3	94.4	125.0	125.2	124.4
Wet Bulb Temp., °F	75.8	78.5	78.3	78.7	84.2	85.4	88.0
Relative Humidity, %	41.0	49.5	49.0	50.0	19.0	21.0	24.5
Air Conditions at Condenser Inlet							
Dry Bulb Temp., °F	96.5	101.1	97.9	97.4	126.7	128.2	127.3
Wet Bulb Temp., °F	76.2	80.0	79.3	79.5	84.5	86.0	88.8
Relative Humidity, %	39.5	41.5	45.0	46.8	18.7	19.0	23.5
Air Conditions at Condenser Outlet							
Dry Bulb Temp., °F	123.7	123.9	119.5	118.9	157.1	152.2	153.8
Air Conditions at Evaporator Inlet							
Dry Bulb Temp., °F	79.9	80.4	80.2	79.1	90.8	89.8	89.0
Wet Bulb Temp., °F	66.8	67.7	67.2	66.1	80.3	80.0	79.0
Relative Humidity, %	50.7	51.8	51.8	50.2	64.5	65.0	65.2
Air Conditions at Evaporator Outlet							
Dry Bulb Temp., °F	62.5	64.2	63.2	60.6	79.7	76.8	75.5
Wet Bulb Temp., °F	----	----	----	57.0	77.2	75.4	73.6

TABLE 1 (cont'd)

Test No.	1a	2a	3a	4a	1b	2b	3b
Power Consumption, Watts							
Compressor Motor	2867	2895	2795	2678	3883	3874	3759
Evap. Fan Motor	303	314	306	294	309	301	301
Cond. Fan Motor	478	493	490	483	472	476	475
Total	3648	3702	3591	3455	4664	4653	4535
Compressor Motor Current, amps	9.0	9.0	8.7	8.3	11.7	11.5	11.3
Condenser Fan Motor " "	7.0	7.1	7.2	6.9	7.0	7.1	7.1
Evaporator " " " "	3.9	4.0	3.9	3.8	4.0	3.9	3.9
Terminal Voltage	208	208	208	207	208	208	208
Moisture Load, lbs/hr							
Evaporated at humidifier	1.11	0.55	1.18	2.69	3.69	2.96	5.0
Collected off evaporator	3.75	3.50	3.50	2.5	5.25	5.0	5.5
Fresh Air introduced into calorimeter by air conditioner, cfm	71	93	92	0	71	66	91
Head Pressure, psig	195	199	189	181	293	294	287
Suction Pressure, psig	40	40	37.5	34.8	56	56.1	52.7
Net Sensible Cooling Capacity, Btu/hr	17990	17930	18790	20120	12570	13850	14910
Net Dehumidifying Capacity, Btu/hr	1180	580	1250	2850	3910	3140	5300
Total Net Cooling Capacity, Btu/hr	19170	18510	20040	22970	16480	16990	20210
Sensible Heat Ratio, %	93.8	96.8	93.8	87.6	76.3	81.5	73.8
Performance Factor, Btu/hr/watt	5.26	5.00	5.58	6.65	3.53	3.65	4.46
Total Dehumidifying Capacity, Btu/hr	3980	3710	3710	2650	5570	5300	5830
Total Heat Removed from Leakage Air, Btu/hr	2320	3800	3800	0	1160	1280	3660

discrepancy between the amount of water evaporated in the humidifier and the amount collected from the evaporator coil was due to the large amount of leakage air introduced into the calorimeter by the air conditioning unit. The additional water was extracted from this fresh air. The amount of air leakage was variable between tests depending on how securely the fresh air dampers closed in response to the control lever.

A comparison of the total net cooling capacity for inside and outside mounting of the air conditioning unit in Table 1 shows that not much difference in capacity was obtained for the two conditions. The inconsistency between this comparison for the high and low ambient temperature conditions is probably accounted for by the variations in fresh air leakage. Higher leakage was observed for the outside mounting at the lower ambient temperature condition whereas the reverse occurred at the higher ambient condition. A computation of the probable heat transfer from the condenser section cabinet (consisting of three parts) and the evaporator section cabinet (consisting of two parts) indicated that the total net cooling capacity should be about 800 to 1000 Btu/hr higher with the unit mounted outside the conditioned space. This apparent advantage of the outside location would disappear, however, if condensation occurred on the evaporator housing in the outside location or if long connecting ducts were used.

When the unit was divided into two parts as for tests 3a and 3b, the total net cooling capacity was increased from 4 to 8 percent at the lower ambient temperature and from 19 to 23 percent at the higher ambient temperature. The major portion of this increase is undoubtedly due to the fact that with the divided installation heat is not transferred from the condenser housing to the conditioned space nor is heat transferred from the ambient air to the evaporator housing. When the evaporator section was mounted on top of the condenser section, only a sheet of aluminum approximately 3/16 in. thick separated these parts.

Test 4a, made with the fresh air dampers sealed, showed an increase in total net cooling capacity of 2930 Btu/hr as compared to Test 3a, made under the same conditions except for sealing the fresh air dampers. This increase in capacity should be a measure of the effect of air leakage through the unit when the condenser and evaporator sections are divided as for Tests 3a and 4a. It is of the same order of magnitude, but somewhat less than the computed total heat removed from the leakage air shown for Test 3a in Table 1.

It will be noted in Table 1 that the computed value of the total heat removed from the leakage air is less for tests 1a, 1b, and 2b than the difference between the total dehumidifying

capacity and the net dehumidifying capacity even though this difference is only a part of the total heat load caused by the leakage of ventilating air. This inconsistency indicates either that there was air leakage into the calorimeter in addition to that removed by the auxiliary blower used to remove leakage air from the conditioned space or that the total heat added by the leakage air was actually greater than that represented by the ambient air conditions and the conditions at the inlet to the evaporator. Leakage of air around some of the cabinet panels of the condenser section could explain this discrepancy for tests 1a and 1b.

Computations of the net sensible heat removal from the conditioned space based on the air circulation rate of the evaporator fan and the increase in dry-bulb temperature between the evaporator section outlet and inlet openings do not agree exactly with the values of net sensible cooling capacity reported in Table 1 derived from the calorimeter measurements using watthour meters. The computed values ranged from 8.1 to 16.1 percent higher than the net sensible cooling capacities reported in Table 1 for the seven tests. This discrepancy is probably accounted for by the fact that in many air conditioners it is difficult to obtain an accurate weighted average of the discharge air temperature because the confined conditions in the chilled air circuit make it improbable that a uniform velocity and temperature distribution can be attained at the evaporator outlet. A part of the difference between the computed and measured net sensible cooling capacities may result from the fact that the evaporator air circulation rate was not measured during each individual capacity test, but was determined in a separate test.

The electric power consumption of the air conditioner appeared to be rather high for the net cooling capacities observed. This results in a low performance factor in Btu/hr/watt as shown in Table 1. For tests made at ASRE standard rating conditions (ambient conditions 95 DB, 75 WB; room air inlet 80 DB, 67 WB), Federal Specification OO-A-372, covering self-contained air conditioning units, requires a minimum performance factor of 6.0 Btu/hr/watt for console-type air conditioners with air-cooled condensers, such as this unit. The unit reached this performance factor only in Test #4a during which the fresh-air damper was sealed.

Table 1 shows that the sensible heat ratio was above 90 percent for the tests at ASRE standard rating conditions with 51 percent relative humidity in the conditioned space. This result indicates that this unit would not be able to remove very much moisture from a space at a humidity level of 51% and suggests that the evaporator surface was not being effectively used. The Federal Specification for self-contained air

conditioning units requires that the sensible heat ratio be within the range from 70 to 80 percent for these room conditions.

B. Air Circulating Capacity

Determinations of the air circulating capacity were made for both the evaporator and condenser fans. The results of these measurements are summarized in Table 2. Federal Specification OO-A-372 for self-contained air conditioning units in this range of sizes requires a minimum of 30 cfm of air circulation per thousand Btu/hr of rated capacity. The amount of air circulated through the evaporator and condenser exceeded 30 cfm per thousand Btu/hr of observed capacity.

The ventilating air that leaked through the air conditioner with the dampers closed but not sealed was measured during all capacity tests. As shown in Table 1, the leakage ranged from 66 to 93 cfm. The latter amount is more than 10% of the total air handled by the evaporator fan, and the loss of cooling capacity caused by this unwanted air can be seen by comparing the net and total dehumidifying capacities and noting the computed total heat removed from the leakage air, as reported in Table 1.

TABLE 2

Air Circulating Capacity of GE Unit

	Air capacity	Temp. of air during fan tests	
	cfm	°F DB	°F WB
Evaporator Fan	881	79	58
Condenser Fan	1004	78	60

Federal Specification OO-A-372 requires that air conditioners be equipped with provisions to admit 5 cfm of fresh air per 1000 Btu/hr cooling capacity. For the unit tested this would amount to about 100 cfm of fresh air or only a little more than the leakage air observed on this unit. Thus the heat load caused by the air leakage should probably not be considered as totally wasted capacity, but should be viewed as an irreducible minimum whereas the fresh air dampers should permit adjustment of the quantity of fresh air below as well as above the value of 5 cfm per 1000 Btu/hr of rated capacity. The two interlocking sets of dampers on the evaporator section permitted adjustment of the quantity of fresh air from the minimum values shown in Table 1 to 100 percent fresh air.

C. Electrical Tests

A Brush oscillograph was used to measure the instantaneous

starting current for the entire unit and of the evaporator fan alone. The highest current observed for the entire unit was 166 amperes, and for the evaporator fan, 56 amperes. The time of starting was 0.25 seconds for the compressor and 0.52 seconds for the evaporator fan.

D. Construction Features

To permit checking the amount of condensate drained from the evaporator coil in relation to the amount of water evaporated in the humidifier, the condensate drain was connected by means of rubber tubing to a receptacle outside the calorimeter. It was found that, as submitted, the condensate drain was connected to the evaporator fan which of course would put the condensate back into the evaporator air stream. This condensate drain should have been connected into the condenser fan, and representatives of the manufacturer stated that in production this error would be corrected.

The primary thermostat furnished with the unit was equipped with a capillary line too short to permit mounting the sensing bulb in its intended place in the return air stream. According to the manufacturer this would also be corrected in production, and before the tests were completed a satisfactory thermostat was furnished.

During the testing of the specimen, the silica gel dehydrator installed in the liquid line gradually became restricted, making it necessary to re-run several of the tests after replacement with a new dehydrator.

Each time the compressor was started after extended periods of shut down severe rapping or knocking was noted in the compressor. This was believed to have been caused by slugs of oil entering the cylinders. Although the knocking was considered extreme the compressor did not appear to have been damaged during the course of testing. The knocking did not continue more than 10 or 15 seconds after starting, and the severity was dependent on a number of variables, length of shut down time apparently being the most significant.

Erratic expansion valve operation observed during some of the first test runs was traced to the gradual plugging of the dehydrator. No expansion valve difficulty was observed after the new dehydrator was installed and the original power element, which had been replaced in investigating the difficulty, was reinstalled and operated properly.

Head pressures observed during the tests at high ambient temperatures were excessive, according to a representative of the manufacturer. Under his supervision the unit was discharged,

evacuated and recharged, but no appreciable change in operating head pressure was observed following this procedure.

Several spot-welds which attached the grilles to the condenser air inlet opening broke loose during operation and instrumentation of the unit.

V. DISCUSSION AND CONCLUSION

The results of these tests show that the total net cooling capacity of the air conditioner was highest when it was divided sectionally, i.e. the evaporator section in the cooled space and the condenser section outside of the cooled space. The reduction in capacity caused by mounting the entire unit inside or outside the conditioned space would probably exceed the 4 to 8 percent difference observed during the tests if any appreciable amount of ductwork was used in either the condenser or evaporator air circuit.

The performance factor of the unit was below the minimum value used in the Federal Specification on self-contained air conditioning units when it was tested with the fresh air dampers in the closed position. Consideration should be given to redesign for tighter closure of the fresh air dampers to permit better control of the quantity of fresh air.

It should be noted from Fig. 4, that the evaporator fan pushes the air through the evaporator coil rather than pulling it through as is a common type of construction. Pushing the air through the coil in the confined space of the evaporator section may tend to channelize the air and reduce the amount of coil area effectively used for heat transfer.

FIG. 1

FIG. 2

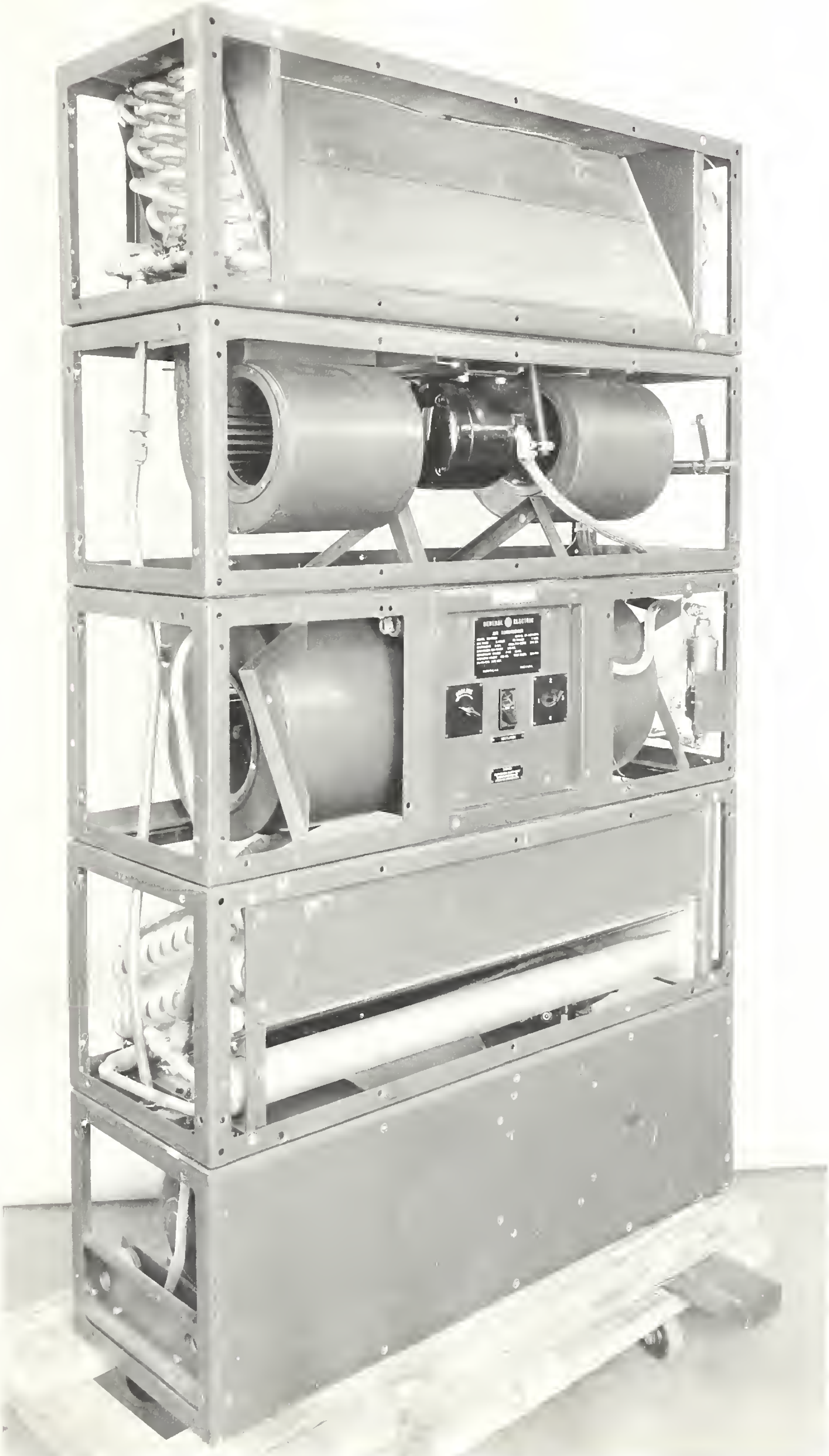

FIG. 3

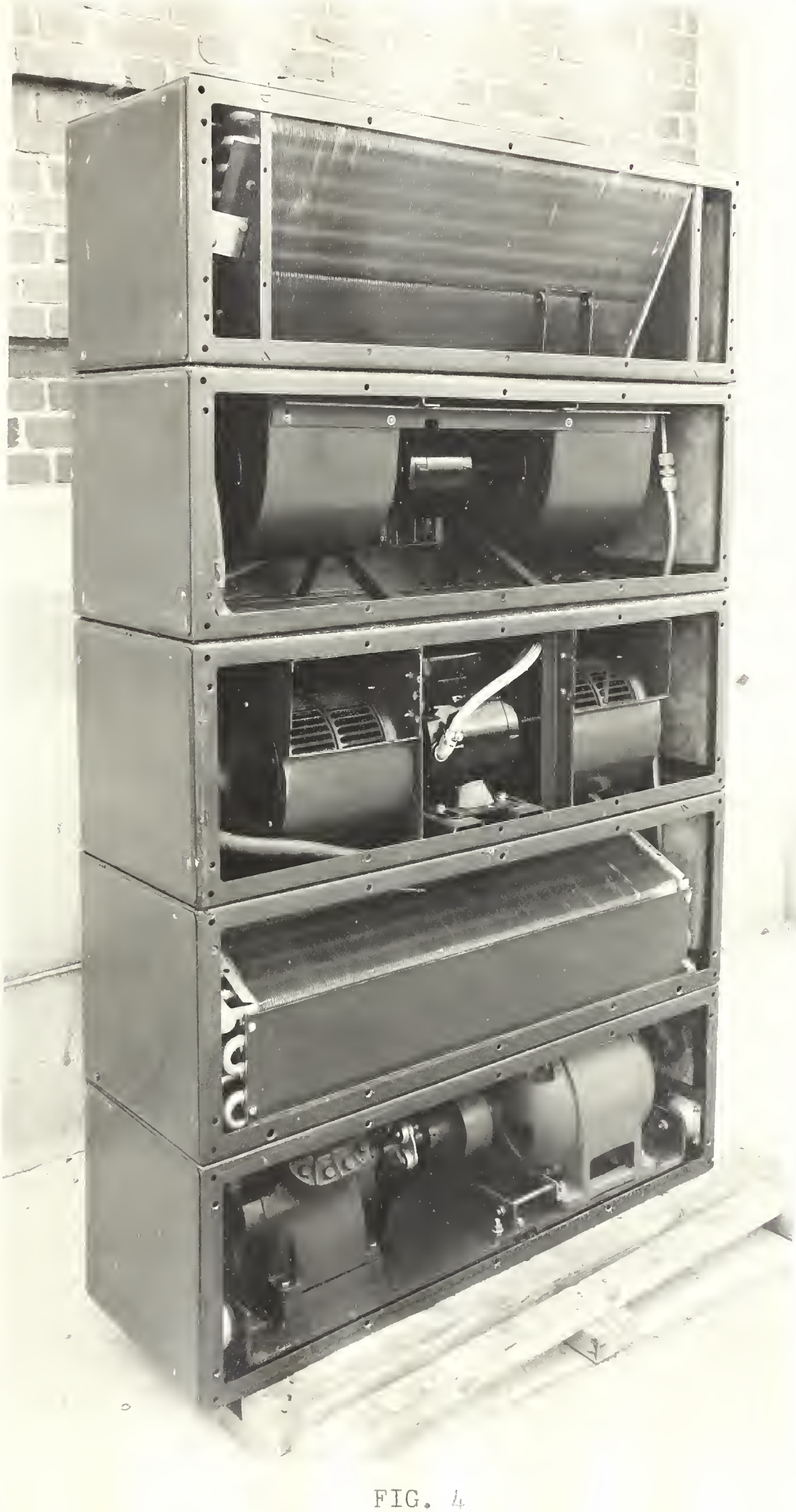

FIG. 4

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