THE DEVELOPMENT OF A

PREFABRICATED REFRIGERATION UNIT

FOR WALK-IN TYPE FARM REFRIGERATORS

By

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INTRODUCTION

The gain in popularity of freezing as a means of food preservation has created a need for additional refrigeration on the farm. Even with the wide selection of commercial equipment to choose from, many of the units purchased have proven to be inadequate in storage and freezing capacity.

A practical solution to the problem of ample capacity and economy of installation and operation, in many cases, is the walk-in type farm refrigerator which may be constructed in part by farm labor using locally available materials. Such a unit was designed, constructed and extensively tested in the V. P. I. Agricultural Engineering Department (7, 12).

One of the major items in the initial cost of such a farm refrigerator is the installation of the refrigeration equipment. The nature of this type of installation is conducive to very inefficient use of labor, equipment and facilities. If shops equipped for refrigeration work were used to prefabricate the refrigeration equipment the labor could be used more efficiently because, (a) costly time involved in travel to and from the installation would be reduced to a minimum and (b) the assembly would be made in familiar surroundings with all the necessary tools, equipment and supplies available. Furthermore, the supplier should be able to furnish a more trouble-free installation because the assembly could be made under better working conditions and the equipment could be thoroughly inspected and adjusted before being released from the shop. All these advantages for prefabrication should make it possible for the farm operator to get a more satisfactory installation at less cost. Also, he would have a unit that would be structurally separate from the refrigerator in which it was installed. In case of emergency such a unit could be removed for repair or replacement or for use in other refrigerators.

Manufacturers of refrigeration equipment offer integral units for air conditioning installations, for use with milk coolers and in similar applications which have proven highly satisfactory. These units are designed for operation at an evaporator temperature considerably higher than that required in a walk-in refrigerator. While the gasoline-powered units designed for operation on refrigerated transport trucks came closer than any of the other units studied to meeting the requirements for use with the walk-in farm refrigerator, they were not considered to be fully satisfactory. In all cases commercial equipment currently available would require considerable modification before it could be used.

Since the integral refrigeration units used for other purposes have proven satisfactory it seemed feasible that such a unit could be designed and used on a walk-in type farm refrigerator.

OBJECTIVES

Based on the foregoing considerations, this study was undertaken with the following objectives in mind:

1. To design an integral refrigeration unit for use with the frozen food compartment of a walk-in type farm refrigerator.
2. To fabricate and install such a unit.
3. To performance test the unit and compare results of these tests with tests that have been run using larger refrigeration equipment which was installed in the conventional way.

MATERIALS AND EQUIPMENT

The Existing Refrigerator Cabinet

The two-temperature walk-in type refrigerator used was designed, constructed and tested in connection with previous experiments of the Agricultural Engineering Department at Virginia Polytechnic Institute (7, 12). The overall dimensions of the cabinet were 11 ft 1 in by 7 ft 11 in and 8 ft 4 in high. The high temperature room contains approximately 250 cu ft gross space and the low temperature compartment contains approximately 75 cu ft gross space. With the exception of the floor, the refrigerator was enclosed by an insulated compartment and facilities were available for thermostatically maintaining a fairly constant ambient temperature.

The cabinet which had been constructed on the 8-inch reinforced concrete floor of the laboratory had a wood floor covered with linoleum. The floor was insulated with 6 in of mineral wool insulation under the chill room and 12 in under the frozen food compartment. Outside walls were of four-inch cinder blocks. The vapor seal consisted of layers of Sisalkraft paper placed in hot asphalt with two layers used for the high temperature room and three layers used for the freezer or low temperature compartment. Mineral wool insulation was used with 6 inches surrounding the chill room, 12 in for the freezer. An 8 in insulated and vapor sealed wall separates the freezer and the chill room.

Entry to the cabinet is through a 3 ft by 6 ft door opening into the chill room from the outside. A 2 ft by 6 ft door opening into the chill room gives access to the freezer. Asbestos panels are used as the inside finish for the entire cabinet.

Refrigeration Machinery and Equipment

Chill Room

The performance of the chilling compartment was not being studied in this experiment. In order to properly study the operation of the freezing compartment, however, it was necessary that a temperature of approximately 35 F be maintained in the chilling room. The refrigeration equipment consisted of a 1/2 hp condensing unit connected to a ceiling mounted unit cooler with properly chosen controls providing automatic operation. The condensing unit was replaced with a 1/4 hp unit before the completion of the tests.

Low Temperature Compartment

1. Condensing Unit

The condensing unit was a Blu-Cold model PL-2F and consisted of a piston type compressor, receiver, condenser and 1/4 hp electric motor mounted on a rectangular frame 16 in by 19 in by 12 1/2 in high. Its rated capacity was 1200 Btu per hour at minus 10 F evaporator temperature and 90 F condensing temperature using Freon 12 as the refrigerant.

2. Evaporator

The evaporator consisted of four 22 in by 48 in Dole vacuum plates and a drier coil, consisting of 33 ft of 1/2 in copper tubing, connected in series. This evaporator assembly was mounted on a panel and was located in the upper portion of the low temperature compartment when the unit was installed. (See Figs. 6 and 13)

3. Expansion Valve

The tests were started with an Alco type 402 thermo expansion valve. During tests in series D-2 this valve was exchanged for a Frigidaire thermostatic expansion valve model MX-3 AC which had a maximum rating of 3000 Btu per hour.

4. Pressure Gauges

A Frigidaire type 13163-1 pressure gauge, with a range of 0-300 psi, was mounted on the control panel and connected in the liquid line near the receiver. A compound gauge, Frigidaire type AMP 9567, with a range of 30 in vacuum to 60 psi, was mounted on the control panel and connected to the suction line near the compressor. (See Figs. 7, 8 and 9).

5. Pressure Switch

A Ranco type O pressure switch was connected on the suction side of the system to provide automatic operation. (See Fig. 7)

6. Miscellaneous Equipment

Other pieces of equipment that were used in assembling the unit were a dehydrator, filter, heat exchanger, packless shut-off valves, sight glass for refrigerant line, and various fittings. Copper tubing was used to interconnect the units. One-half inch was used for the suction lines and one-fourth inch for the liquid lines. Fig. 7 shows the manner in which the refrigeration equipment was assembled.

Recording and Indicating Instruments

Following is a brief description of the various types of recording and indicating instruments used in the tests. These same instruments or similar ones were used in prior tests (7, 12), the results of which are compared with the results obtained in this experiment. Since the instruments were identical or similar the results obtained in the two experiments should give comparative values for the operating characteristics of the two units.

Recording Potentiometer: The Brown Instrument Company Model 153X60P16-X-31F1, 16-point strip-chart recording potentiometer was used. This instrument had a range of minus 40 F to 140 F and recorded the temperature indicated by copper-constantin thermocouples at 30-second intervals.

Recording Thermometer: A Bristol Model 3T500-11, Range minus 20 F to plus 120 F, 3-temperature, 24-hour chart recording thermometer was used. The thermally-sensitive bulbs were located so that the representative temperatures of the freezer, chill room and ambient inside the temperature controlled compartment were recorded.

Recording Wattmeter: A type C D 14 General Electric recording wattmeter with a maximum range of one KW was used.

Pressure Recorder: The suction pressure was recorded by a Bristol, Model 44, key wind, 24-hour chart pressure recorder. The instrument had a range of 30 in vacuum to 50 psi.

Time Meter: The portable time meter used was a General Electric Model 8KT1A4. It was designed for operation on 230 volts AC. Operating time of the attached electric motor could be read directly. To provide 230 volts for the operation of the clock, a step-up transformer was used.

Watthour Meter: A Sangamo, type H C, watthour meter was used to determine the energy consumption to the nearest hundredth KWH.

Refrigerant Flow Meter: The refrigerant flow could be measured at any time by a master enclosed flowrator manufactured by The Fischer and Porter Company. The capacity of the meter was 0-40 pounds of Freon 12 per hour.

Counter: An electro-magnetically operated counter was connected in parallel with the motor on the condensing unit to record the number of operating cycles.

PROCEDURE

As outlined in the objectives the major subdivisions of this investigation involve design, fabrication, installation and testing. In addition, correlations with the tests already run on the refrigerator are to be made wherever possible. Each of these factors will be discussed in the following paragraphs.

DESIGN

Integral refrigeration units for air conditioning, milk coolers, transport trucks, etc., are in production commercially. This would indicate that such units are feasible for these applications.

It appeared that such a unit would be desirable for use on the walk-in type farm refrigerator, therefore, one was designed for this purpose. For the most part conventional refrigeration design and design procedures were used throughout. A statement of the significant requirements and the design considerations relative thereto follows.

Heat Loads

The heat transfer losses through the walls, ceiling, floor and door of the frozen food cabinet, based on 90 F ambient and 0 F storage temperature, were calculated to be 10,072 Btu per 24 hours (Appendix B). This includes a calculated heat gain of 197 Btu per 24 hours resulting from the substitution of a 2 ft by 3 ft by 10 1/2 in insulated panel for the 12 in insulated wall of equal area. It should be noted that the increase of 197 Btu per 24 hours represents only about 1.9 per cent of the total wall losses. These losses are slightly higher than as calculated by Cristel (7) with this increase being attributed to the addition of the panel, differences in construction features and losses to the chill room were not considered in his tests since one condensing unit was used for both compartments.

Published recommendations pertaining to the number of air changes (2) are for considerably larger refrigerators. It seemed that 10 air changes for stand-by or holding operation and 20 air changes for loading or freezing operation would be ample for this type of construction and use. This gave a 24 hour heat load of 900 Btu for stand-by and 1800 Btu when processing.

The miscellaneous load due to lights and men working in the compartment was estimated to be 480 Btu per 24 hours for stand-by and 1920 Btu per 24 hours when processing.

The recommended capacity of the freezer for 90 F ambient conditions is 30 pounds of produce per day (9). Estimated average values for the specific heat above freezing, latent heat of fusion, and specific heat below freezing were used in calculating the produce heat load (Appendix B). Vegetables were used in these calculations since more heat must be removed per pound than for meats (2, 14).

Allowing 10 per cent as a safety factor the total heat to be removed by the refrigerating equipment was calculated to be 525 Btu per hour when operating under stand-by conditions and 1080 Btu per hour when operating with a recommended freezing load (Appendix B).

The hourly heat loads were calculated on the basis of continuous compressor operation during the 12 hour freezing period. For operation during the freezing period the calculated heat load was 1080 Btu per hour with the capacity of the condensing unit being 1200 Btu per hour for the design conditions of 90 F ambient condensing temperature with minus 10 F refrigerant. For the next 12 hours the heat load was calculated to be 629 Btu per hour (Appendix B). This would be approximately 52.5 per cent of the rated capacity of the condensing unit, which would give approximately 6.3 hours of operating time in this 12 hour period.

Summarizing the above operating conditions, the condensing unit would be expected to operate approximately 18.3 hours in each 24 hour period following loading. This conforms to the recommendations for compressor operating time as given in the A.S.R.E. Data Book (2) and the A.S.R.E. Food Freezer Standards (1).

Evaporator Area

The evaporator actually performs two functions; removing the heat from the product load and maintaining the desired temperature in the storage compartment. It is desirable that these functions be performed simultaneously, therefore, the area needed for each was calculated separately. The rate of heat transfer to the refrigerant is dependent upon the type of contact between the plate and material from which heat is to be extracted. When the medium in contact with the plate is air, a heat transfer factor of 2 Btu per sq ft per hour per degree Fahrenheit temperature difference is generally recommended (2, 4) using the area of both sides of the plate. For the condition where packaged products are placed directly in contact with the plate the transfer factor is 4 Btu per sq ft per hour per degree Fahrenheit temperature difference using the area of only one side of the plate (2, 4).

A temperature difference of 5 F for refrigerant to air was used in determining the plate area needed for holding the storage compartment temperature. This is considerably lower than the 16 F difference generally recommended by the manufacturers (4) and used in the trade. The author feels justified in making this reduction because very careful estimates were made on the approximate average suction pressures maintained on many 24-hour tests. These suction pressures were converted to refrigerant temperatures and compared with the actual recorded temperatures. The value obtained was approximately 6 F.

The latent heat is the largest portion of heat to be removed from the product and this transfer occurs at approximately 30 F. Preliminary tests, on other refrigerating equipment, indicated an average refrigerant temperature of approximately 0 F for the freezing period. Therefore, 0 F to 30 F was used as the temperature difference for calculating the plate area needed for contact freezing. Having these conditions in mind the plate area may be calculated in the following manner:

$$\text{Plates needed} = \frac{\text{Load in Btu per hour}}{\text{transfer factor x TD x plate area}}$$

Since a 22 in by 48 in plate has a design area of 14.8 sq ft for air and 7.4 sq ft for contact freezing one finds the number of plates needed for stand-by conditions to be $\frac{525}{2 \times 5 \times 14.8} = 3.56$ plates,

and for freezing to be $\frac{451}{4 \times 30 \times 7.4} = 0.51$ plates.

Using the top plate for freezing the product and three other plates to maintain the storage temperature would very closely approximate the desired design condition for processing. When operating at stand-by the freezing plate would supplement the three holding plates giving more than adequate capacity.

Condensing Unit

The maximum heat load for processing was calculated to be 1080 Btu per hour (Appendix B). Using this as a basis, a standard commercially available condensing unit was selected from manufacturers' catalogs. The unit selected had a rating of 1200 Btu per hour at a minus 10 F evaporator temperature and 90 F ambient condensing temperature.

Panel

The width of commercially available evaporator plates and the six inch spacing of same determined the 2 ft width and 3 ft height of the panel. This width dimension allowed 2 in for the evaporator frame and clearance when standard 22 in plates are used. The author felt that 12 in of space should be left between the drier coils and the top plate to allow clearance for freezing turkeys or large hens. Six inches was selected as the spacing for the plates as it would allow placing two layers of packages from the frozen storage on each of the holding plates. An additional two inches was allowed for the space occupied by the evaporator framework. Four inches vertical clearance was considered necessary for installing the unit due to its length and weight. This gave a total height of 3 ft for the panel.

A rigid frame was considered essential since the unit would be transported after assembly, therefore, 2 in dressed planks were used for constructing the panel. Calculations were made to determine the loading on the panel when the assembled unit was in place. From these calculations it was determined that 1/2 in plywood had sufficient strength in bending, shear and compression to support the condensing unit. The framework supporting the evaporator plates was attached directly to the main frame of the panel and was also supported on the outer end when the unit was installed. Calculations indicated sufficient strength for the panel to support the evaporator frame when the unit was installed and also when handled or transported. The outside cover of 1/2 in plywood was extended six inches on either side and the top, to simplify the attachment of the unit to the framework of the refrigerator cabinet. Calculations were made to determine the heat losses for panels made up of standard widths of 2 in lumber. Appendix B shows an added heat load of 197 Btu per 24 hours for the 10 1/2 in panel thickness selected which used a standard 2 in by 10 in plank frame. In the author's opinion, the lesser standard widths of boards, used in constructing the frame for the panel, would give excessive heat losses and the greater widths would add undesirable weight and be subject to warping and splitting. Complete specifications for the panel are shown in Appendix D.

Supports

To prevent damage to the unit during transportation and installation it was necessary that the framework have sufficient rigidity. For the evaporator frame, (See Figs. 1 and 2), 1 1/2 in by 3/16 in angle iron was used as the base. These angles were spaced 24 in at the panel and 22 in at the outer ends. Flat iron, 1 1/2 in by 3/16 in was used for the end vertical and diagonal members to complete the frame. Center vertical members, consisting of flat iron, 1 1/2 in by 1/4 in were placed between the bottom angles and the diagonal top supports to be used for supporting the evaporator plates. Round steel rods 1/4 in in diameter were used to connect the vertical members and provide the proper plate spacing. A rack for mounting the drier coil was provided. Support for the end of the metal frame not attached to the panel was provided by wooden columns fitted at the time of installation.

The condensing unit was mounted on a welded frame prefabricated from 1 in by 1 in by 3/16 in angle iron and attached to the outer cover of the panel (see Fig. 4). Complete specifications for this construction are shown in Appendix D.

Miscellaneous

The expansion valve, pressure switch, filter, dehydrator, heat exchanger, and sight glass were selected from standard commercial articles currently available.

FABRICATION

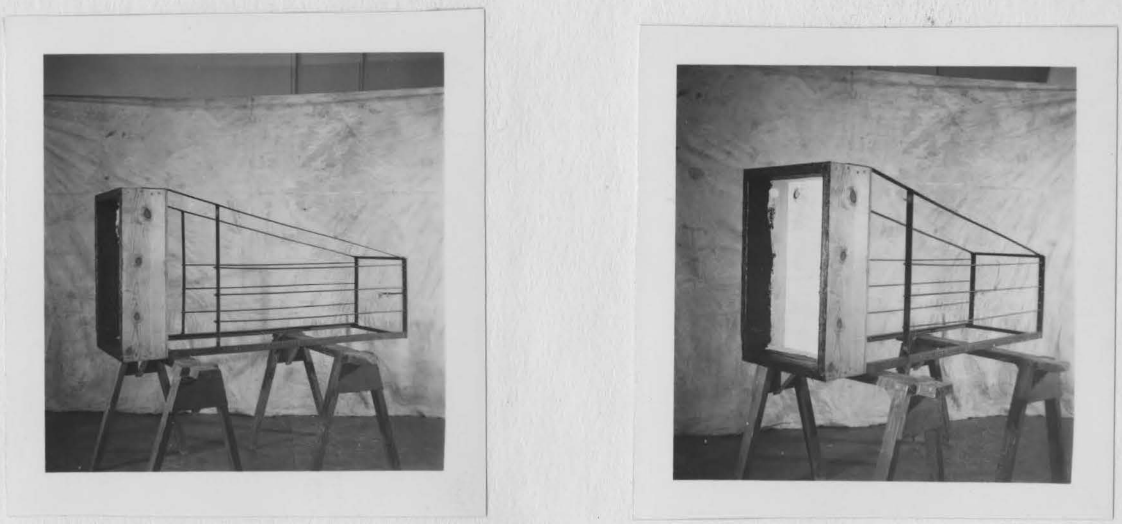
The facilities of the research shop of the Agricultural Engineering Department, Virginia Polytechnic Institute and special tools and equipment furnished by the Virginia Agricultural Experiment Station were used in the construction of this unit. Wherever possible the joints of the metal framework were welded with an electric arc welder.

The fabricated angle iron framework for supporting the evaporator plates was attached to the inside edges of the panel as shown in Fig. 1. Vapor seal compound was applied to the inside surface of the panel as shown in Fig. 2. The conduit for accommodating the connections between units was next installed after which the insulation was put in place, Fig. 3. The frame upon which the condensing unit was to be mounted was attached to the outside cover of the panel as shown in Fig. 4, and then three alternate layers of vapor sealing compound and vapor seal paper were applied to the inside of the panel. Fig. 5 shows the outside cover before it was put in place and fastened, using galvanized wood screws. The completed assembly is shown in Fig. 6.

The evaporator plates were put in place and tightened by means of nuts on the end of the tie rods. They were connected in series using 1/2 in outside diameter copper tubing and the necessary solder fittings. The expansion valve was installed so that the refrigerant would be fed to the bottom plate. The drier coil was fabricated from 1/2 in outside diameter copper tubing and mounted on the frame with one end connected to the outlet from the top plate and the other end carried through the opening provided in the panel and connected to the heat exchanger. This

(a) (b)

Figure 1
Views of Panel With Evaporator
Frame Attached.



(a) (b)

Figure 2
Views of Panel Showing Vapor
Seal Compound Applied.

Figure 3
Insulation in Place
Note: Conduit in Upper Left.

(a)

(b)

Figure 4
Views of Condensing Unit Frame
Attached to Panel.

Figure 5
Panel Before Final Assembly.

(a)

(b)

Figure 6
Views of Assembled Unit.

line was extended to the compressor. The dehydrator was mounted on the inside cover of the panel and the refrigerant line extended to the expansion valve and also to the outside where it was connected to the heat exchanger. From the heat exchanger the refrigerant line was extended to the receiver completing the circuit. It was not possible to test the unit before installation because certain control and test equipment had to be put in place after the unit was installed in the refrigerator. (see Figs. 7, 8, and 9). The weight of the complete assembly was 386 pounds.

INSTALLATION

An opening to the freezer compartment of the walk-in refrigerator was made through the front end wall (see Fig. 10) and framed 1/2 in larger than the panel. The wooden frame facilitated the use of screws in mounting the panel assembly. The original vapor seal was spliced and brought to the outside (Fig. 10). Vapor sealing compound was placed on the outer face of the opening and the overlap portion of the panel cover. That portion of the assembly containing the evaporator was then passed through the opening, leveled and supports fitted to the end opposite the panel. Screws extending through the overlap portion of the panel fastened the assembly to the wood frame of the outer wall (see Fig. 11).

The necessary connections were then made to the test equipment (see Figs. 7, 8, 9, and 11) and the unit was ready for operation.

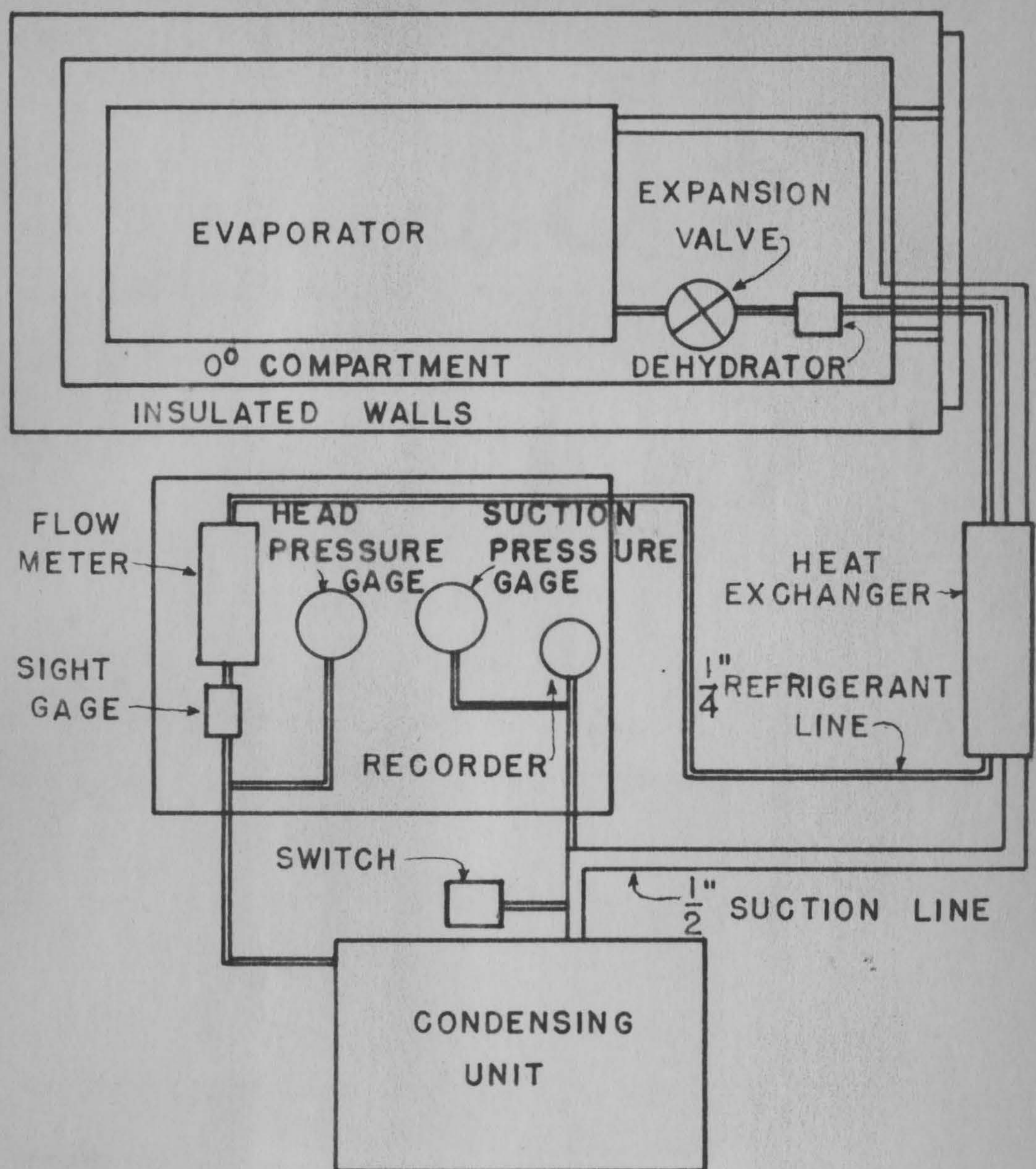


FIGURE 7

SCHEMATIC DIAGRAM OF
REFRIGERATION MACHINERY

NOT TO SCALE

Figure 8
View of Control Board Showing
Flowmeter, Pressure Gages, Counter
and Pressure Recorder.

Figure 9
View of Control Board on Insulated
Wall of Constant Temperature Compartment
With Installed Unit in Background.

Figure 10
Interior of Freezer Compartment
Note: Vapor Seal Around Opening.

(a) (b)

Figure 11
Outside of Cabinet Showing Unit
In Place.

TEST CONDITIONS

The refrigerator cabinet and equipment used in this investigation were constructed and installed as recommended for farm use. Test instruments and equipment were placed so that a minimum of interference with the operation of the unit was obtained.

Temperature

The ambient temperature of the compartment surrounding the cabinet was maintained at approximately 90 F. This is considered higher than the average daily temperatures expected for the summer months in Virginia (7).

A separate refrigeration system was used to maintain a temperature of approximately 35 F in the chill room.

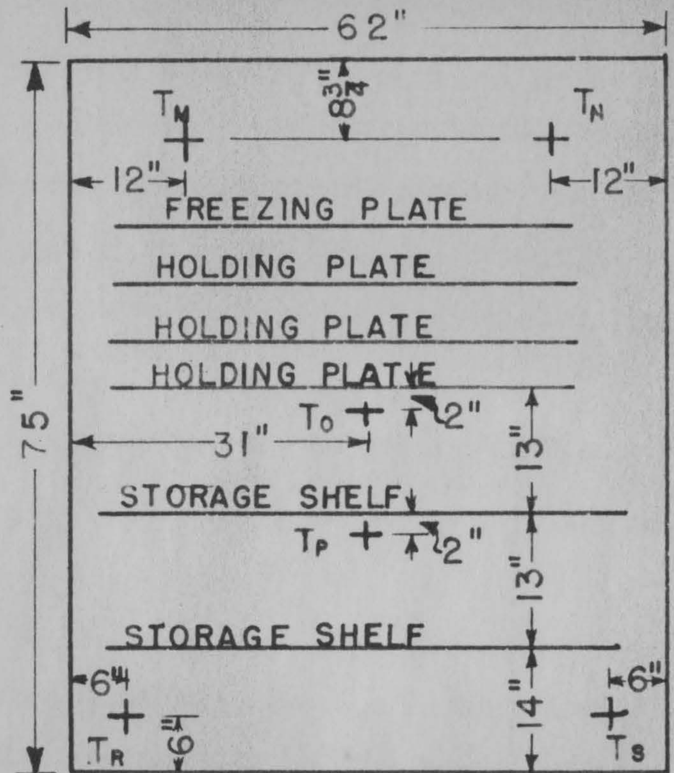
Design of the refrigeration unit for the freezer was based on an average temperature in the storage load of 0 F with minimum fluctuation desired (11).

Water from the College Water System at 70 F was used for all freezing loads.

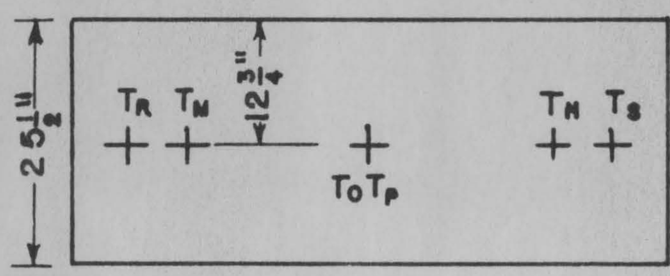
Temperature Measurements and Recordings

The interconnected plug-in panels, which were installed for prior tests, were used for connecting the recording potentiometer to the desired thermocouples inside the refrigerated cabinet. For each test two or more air temperatures inside the cabinet were recorded. The positions of these thermocouples are shown by Fig. 12. Rubber

FREEZING COMPARTMENT



ELEVATION



PLAN

FIGURE 12

THERMOCOUPLE LOCATIONS

tape was used for fastening thermocouples to points on the refrigeration equipment.

The three-temperature recording thermometer gave a constant check on the average temperature in the freezer, chill room and the compartment surrounding the cabinet.

The thermocouples used for determining the storage load temperatures were in the approximate geometrical center of specially prepared packages which were made up and frozen prior to the tests. These packages were of the same size and weight as those used for the storage load and were located so that representative temperatures of the storage load would be recorded for each series of tests.

Storage Load

The weight and placement of the storage load varied with the different series of tests (Figs. 15, 17, 19, 20, 21 and 22). In each series the weight was equally distributed and symetrically placed on the floor rack and on the two shelves. The load itself consisted of rectangular packages of ice that were frozen in preliminary tests and stored until needed.

Loading Procedure

The same procedure was followed for all the load tests. Sixteen double membrane packages were filled with 1.8 pounds of 70 F water. Special wire cages, with thermocouples attached, were placed in three of these packages. Sealing was accomplished with a standard sealing iron, but it was necessary to add paraffin to the packages having the thermocouple leads in the seal. The packages were then placed on the freezing plates as shown in Figs. 13 and 14 with the packages containing thermocouples in positions 3, 5, and 7 (Fig. 14). After the connections for the three thermocouples were made, the doors were closed and locked for the duration of the tests. Records were taken and the instruments tended immediately and in the same sequence for each of the successive tests.

Duration of the Test

Each test was for 24 hours. During this time routine checks were made on the operation of the equipment and the temperatures in the different areas. At the end of each test period the records were taken and the recording instruments tended in the same sequence used at the start of the test.

Unloading

Immediately following each load test the cabinet was opened and the freezing load removed. The plates were then cleared of frost and adjustments made on the equipment or storage load if necessary. The doors were then closed and remained so for a minimum stabalizing period of 12 hours after which the unit was considered ready for the next test.

Miscellaneous

Power used by the compressor motor was 115 volt AC furnished

Figure 13
View Showing Packages as Placed
for all Freezing Loads

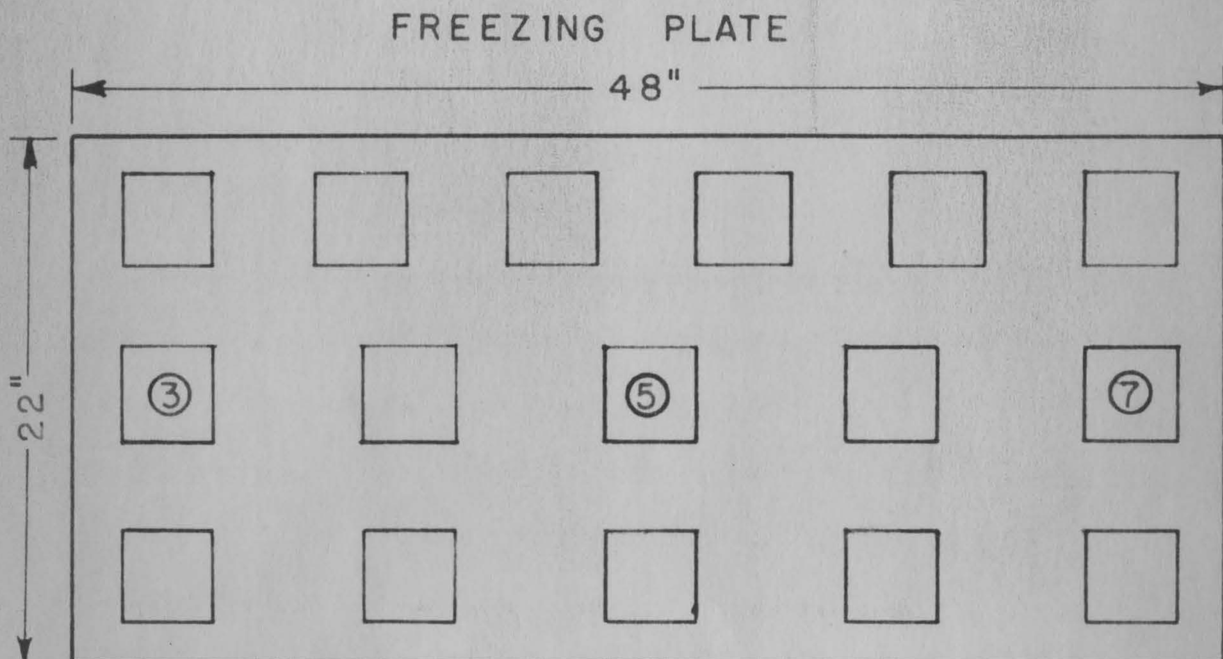


FIGURE 14

PLACEMENT OF FREEZING LOAD

by the Virginia Polytechnic Institute Department of Electric Service.

Water used for the loads was taken from the Virginia Polytechnic Institute water supply.

Plotting the temperature curves, Charts 1 to 26 inclusive (Appendix A), was done with a specially developed machine. This machine was designed and constructed by personnel of the Agricultural Experiment Station who had been engaged in work of this type.

TESTING

Performance testing of the unit consisted of a pull-down test and five series of freezing load tests with five duplications for each series. Conditions for all the load tests were the same except for the amount and location of the storage load. To provide uniformity of tests and charts, the operating time was divided into 24-hour periods.

Pull-down Test

The main purpose of this test was to observe any operating difficulties that might arise during the initial cooling period. It was believed that operating difficulties would not be experienced after the zero compartment had reached a uniform temperature and the unit had started to cycle. The author was especially concerned about the critical nature of the refrigerant charge due to the large evaporator capacity compared to the small receiver capacity of the condensing unit. This made close supervision of the equipment imperative during the pull-down period.

During the period of installation the cabinet temperature did not reach the stabilization point. However, the temperature attained was considered high enough to reveal any operating difficulties that might be experienced in an initial pull-down test. The rate of cooling was considered to be of little importance, therefore, the test was started immediately following the installation of the unit.

The unit could not be put into operation until the test equipment and instruments were installed. This made it impossible to make a preliminary run to adjust the controls, therefore, they were installed as received from the manufacturers with no further adjustment. After preliminary leak checking and charging with refrigerant, the unit was ready for operation. The equipment was kept under close observation during this test so that any operating difficulties could be corrected.

Thermocouples located at the points as shown below were connected to the recording potentiometer which provided a record of the temperature changes.

Thermocouple Number		Location
1.	Freezer	Expansion valve bulb
2.	Freezer	Top air, left
3.	Freezer	Center air, under plate
4.	Freezer	Top plate, center
5.	Chill Room	Corner opposite freezer door
6.	Chill Room	Thermostat bulb
7.	Freezer	Bottom air, left
8.	Freezer	Expansion valve
9.	Freezer	Expansion valve bulb
10.	Freezer	Top air, right
11.	Freezer	Center air, under shelf
12.	Chill Room	Expansion valve
13.	Chill Room	Expansion valve bulb
14.	Ambient	Compartment surrounding cabinet
15.	Freezer	Bottom air, right
16.	Freezer	Expansion valve

At the end of 48 hours of almost continuous operation the temperature of the compartment was minus 3 F which was considered satisfactory. The pressure switch was adjusted so that the compressor would cut out at approximately 2 psi and cut on at approximately 7 psi. Following this adjustment, the compressor operated on approximately equal off and on periods with no evidence of excessive head pressure. An inspection of the chart from the recording wattmeter gave no indication of overload during any part of the 72 hour pull-down period.

The rate of cooling inside the cabinet is indicated by Chart I (Appendix A).

Following is a summary of the operating time and power consumption:

	Total Time	Operating Time	Per cent Running Time	Power Consumption Kwh	Cycles
1st 24 hours	1440	1440	100	8.49	0
2nd 24 hours	1440	1427	99	7.15	1
3rd 24 hours	1440	980	68	4.96	41

LOAD TESTS

As explained in the section on Test Conditions, the freezing load was the same for each of the load tests. It consisted of sixteen 1.8-pound packages of water, which would be approximately equivalent to 30 pounds of vegetables. The same procedure was followed in all the freezing load tests with the only variable being the amount and placement of the storage load.

Thermocouple locations that were common for all the freezing load tests were given the same thermocouple number and the recorded temperatures were plotted as the same numbered curve on the Freezing Rate Charts (Appendix A). The following relationship, between the thermocouple numbers and the plotted curves, was used for these common positions.

Thermocouple Number	Curve Number	Location
14	1	Ambient temperature
6	2	Chill room, thermostat bulb
12	3	Freezer, top air, right
11	4	Freezer air, under shelf
3	5	Freezing load, left
5	6	Freezing load, center
7	7	Freezing load, right

Recorded temperatures at other locations are given for the respective storage load conditions.

<u>No Storage Load, SERIES D-2</u>

These tests were run to determine the operating characteristics of the unit with no storage load and to make adjustments on the equipment. The operation of the expansion valve was not considered satisfactory, therefore, it was changed after the second test. Performance of the unit after this change of expansion valves was considered satisfactory. The rate of freezing and the temperatures maintained in the cabinet are shown on Charts 2 to 6 inclusive (Appendix A). Thermocouples were located at the following positions and used for recording the temperatures in this series of tests.

Thermocouple Number	Location
1. Freezer	Expansion valve bulb
2. Freezer	Top air, left
3. Freezing load	Left
4. Freezer	Center air, under plate
5. Freezing load	Center
6. Chill Room	Thermostat bulb
7. Freezing load	Right
8. Freezer	Expansion valve
9. Freezer	Expansion valve bulb
10. Freezer	Refrigerant line before X valve
11. Freezer	Center air, under shelf
12. Freezer	Top air, right
13. Freezer	Bottom air, left
14. Ambient	Compartment surrounding cabinet
15. Freezer	Bottom air, right
16. Freezer	Expansion valve

277-pound Storage Load, SERIES D-3

The storage load was equally divided and distributed symetrically on the slatted false floor and two shelves of the freezing compartment. This distribution is shown by Fig. 15. Three frozen packages containing thermocouples numbers one, four, and sixteen were placed so that average temperatures of the load on the lower left, center, and top right of the storage area would be recorded. The temperature variation between these three locations was very small, amounting to less than 1 F in all cases, therefore, only one curve was plotted for the storage load. This is shown on Charts 7 to 11 inclusive (Appendix A) by curve number four. No other changes were made in the thermocouple locations as listed for tests in Series D-2. The refrigeration equipment for the chill room failed to perform satisfactorily during three of the tests. These records were discarded and duplicate tests were run to replace them.

The performance of the refrigeration unit on the freezer was considered satisfactory.

Figure 15
View of 277-pound Storage Load
Used in Tests of SERIES D-3.

Figure 16
View of Evaporator Plates and
Freezing Load at End of Tests
in SERIES D-3.

554-pound Storage Load, SERIES D-4

The maximum storage load used by Cristel (7) was 554 pounds. This is not the maximum capacity of the zero storage compartment but will be used since a comparison will be made of the freezing rates for the two units. This 554 pound load was distributed equally between the shelves as indicated in Fig. 17.

Thermocouple locations were the same as indicated for the tests in Series D-3.

Curves representing the temperature changes are shown on Charts 12 to 16 inclusive (Appendix A).

Figure 17
View of 554-pound Storage Load
Used in Tests of SERIES D-4.

Figure 18
View of Evaporator Plates and Freezing
Load at End of Tests in SERIES D-4.

554-pound Storage Load With 174 pounds on Plates, SERIES D-5

The cycling of the refrigeration equipment is caused by variations of the refrigerant temperature in the plates. Observations made of the temperatures maintained in preliminary tests indicated that this fluctuation of temperature might be evident in any products left on the plates. In addition to fluctuation of temperature it was also thought that such storage might effect the performance of the refrigeration unit or the temperature of the main storage load. Since products could be very easily stored in the space between the holding plates or left on the top plate after freezing, the tests in this Series and Series D-6 were run to determine the extent of such temperature variations and the effect on the performance of the refrigeration equipment.

Thirty-two packages of the storage load were transferred from each of the storage shelves to the holding plates. Fig. 19 shows the storage load after the transfer was made. A view of the plate storage and freezing load is shown in Fig. 20. The packages were placed in four rows of eight packages each on the three holding plates.

Thermocouple number four had given the representative temperature of the shelf storage during the prior test. The package containing thermocouple number one was moved to the bottom plate as the second package in the third row of the plate storage and the package containing thermocouple number 16 was used as the seventh package in the second row of the storage on the top plate. The temperature of the storage load on the center plate was recorded by thermocouple number 13 which was in the fifth package of the second row of storage on that plate. Rows were numbered from front to rear and the packages, in these rows, from left to right for the placement of the plate storage. The positions of the thermocouples used in these tests are as follows:

Thermocouple Number	Location
1. Freezer	Plate storage, lower left
2. Freezer	Top air, left
3. Freezing load	Left
4. Freezer	Shelf storage, center
5. Freezing load	Center
6. Chill Room	Thermostat bulb
7. Freezing load	Right
8. Freezer	Expansion valve
9. Freezer	Expansion valve bulb
10. Freezer	Refrigerant line before expansion valve
11. Freezer	Center air, under shelf
12. Freezer	Top air, right
13. Freezer	Plate storage, center
14. Ambient	Compartment surrounding cabinet
15. Freezer	Bottom air, right
16. Freezer	Plate storage, upper right

The difference in temperature of thermocouple numbers 1, 13, and 16 never exceeded 1 F, therefore, the records of only one, number 13, were was plotted as shown by curve number eight on Charts 17 to 21 inclusive

(Appendix A). The freezing load and procedure were the same as in preceding tests.

Figure 19
View of Storage Load on Shelves as Used in Tests of SERIES D-5.

Figure 20
Evaporator Plates Showing Loading and Storage as Used in Tests of SERIES D-5.

554-pound Storage Load with 290 pounds on Plates, SERIES D-6

The maximum storage that could be placed on the holding plates was used for this series of tests. It consisted of 64 packages on each of the two top plates and 32 packages on the bottom plate. A change in the type of evaporator plates after the frame was constructed introduced a variation in the spacing of the bottom plate, which was not detected until this test. This decrease in spacing of 5/8 in made it impossible to place more than one layer of packages on this plate, therefore, the storage load on the slatted floor remained the same as in the tests of Series D-5 but 32 packages were transferred from the other two storage shelves to the holding plates. This placement of the storage load is shown by Figs. 21 and 22. Thermocouples numbers 8 and 9 were moved to the packages directly over thermocouples numbers 13 and 16. It was thought that the temperatures in the two layers of packages might be different. Less than 1 F variation was noted in the temperatures of thermocouples 8 and 9 and, therefore, the temperature of number 8 was plotted as curve number 9 on Charts 22 to 26 inclusive (Appendix A). The remaining thermocouple locations were the same as indicated for the tests in Series D-5.

The test procedure followed was the same as for the preceding series.

Figure 21
View of Storage Load on Shelves
as Used in Tests of SERIES D-6.

Figure 22
Evaporator Plates Showing Loading and
Storage as Used in Tests of SERIES D-6.

GENERAL DISCUSSION

DESIGN

Recommended procedure for determining the heat loads and rate of heat removal, conventional types and sizes of refrigeration equipment, and standard practices in assembling the unit were used in the design. With minor adjustments, the equipment performed satisfactorily.

An attempt was made to determine the temperature difference that might be expected between the refrigerant and the temperature maintained in the cabinet. In order to do this, the recorded suction pressure and the temperature in the coldest part of the cabinet were used. Values for each of these criteria were taken on the even hour marks of the charts for the five tests in each series. The pressure was converted to temperature and the values subtracted. This gave sixty values for each series of tests which had the following arithmetic averages:

Series D-2	No storage load	6.5 F
Series D-2	1/2 storage load	7.0 F
Series D-4	Full storage load	6.6 F
Series D-5	Full storage (one layer each plate)	8.9 F
Series D-6	Full storage (maximum plate)	9.9 F

The 24-hour average temperature difference for all loading and storage condition was 7.8 F. The author feels that the values obtained closely approximate the existing conditions.

The panel had sufficient strength. The framework supporting the refrigeration equipment was not perfectly rigid but the copper tubing interconnecting the system easily absorbed any movements caused by handling.

The unit could be crated for shipment provided necessary precautions were taken to avoid any shifting of parts that might damage the refrigerant lines. Construction of the evaporator frame could be simplified by eliminating the diagonal braces and making the framework rectangular.

The panel had sufficient strength to support the attached frameworks, for the evaporator plates and the condensing unit, when the assembly was moved or transported and after installation.

Detail drawings of the panel and supporting frames for the evaporator plates and condensing unit are shown in Appendix D.

FABRICATION

Since there is very little probability of modifying a unit of this type in the field, welding was used wherever possible in the construction of the unit. The author believed that this method of construction would be more economical and also give a framework with greater rigidity. The unit could be given a rust proof coating more easily.

The diagonal braces and taper on the evaporator frame complicated the construction of the unit with the diagonals interfering in loading operations. A rectangular frame of sufficient width and height to accommodate the evaporator plates would be simplier to construct if sufficient strength could be obtained. The drier coil could then be mounted on the top part of the frame.

Galvanized screws used in fabricating the panel and for attaching the frames were satisfactory.

The frame was originally designed to accommodate four 22 in by 54 in "Yoder" evaporator plates. This type of plate required support for the edges which was furnished by the 1/4 in round rods. When the "Dole" type plates were substituted, the overhanging edges covered the rods. This gave 5 3/8 in spacing for the bottom plate instead of the 6 in spacing used for the others. This condition was not detected until the unit was put under test.

The solder connections which were made using 95-5 solder, did not leak at any time during the tests. Flare fittings used for connecting the control and recording equipment in the refrigerant circuit did leak initially, but after tightening, no further leaks were detected. Although silver solder would have given a stronger joint, the 95-5 type solder gave satisfactory service and was considerably cheaper and easier to use.

INSTALLATION

Most of the difficulties encountered in the installation of the unit would not exist in a new installation. The correct size of opening could be left during the initial construction of the cabinet. Connections to the testing recorders, etc., would not be necessary. The floor plan of the refrigerator cabinet could be oriented before construction so that the opening for the assembled unit would be clear of outside obstructions that might interfere with its installation.

The vapor seal of the refrigerator cabinet could be extended to the outside of the concrete block wall very easily during the initial construction. The six inches overlap of vapor seals used between the refrigerator cabinet and the panel apparently gave a vapor-tight joint.

The opening which allowed the refrigerant lines to pass through the panel was not sealed initially. Since considerable moisture entered the freezer in a very short time this condition was corrected by filling the conduit containing the refrigerant lines with the compound used for the vapor seal on the panel.

The expansion valve was initially fastened to the top part of the metal framework supporting the plates. This seemed to effect the operation of the valve due to the added mass of the frame. By unfastening the expansion valve from the evaporator frame, satisfactory operation was thereafter obtained.

The position of the heat exchanger, which was mounted on the outside of the panel, seemed to be satisfactory. Mounting in this position allowed part of the air passing through the condenser to come in contact with the heat exchanger, which would be at a lower temperature. The condensed moisture dropped to the floor without interfering with the operation of the condensing unit. Mounting the heat exchanger in this position would also tend to increase the superheat of the vapor entering the compressor, thereby decreasing the chance of "slugging" or liquid refrigerant entering the compressor.

Other accessories were mounted as recommended by the manufacturer.

TESTING

The records taken on the installed unit were for the purpose of determining the freezing rate, holding ability and power consumption under the storage conditions outlined in the procedure. Comparison will be made wherever possible with the tests run by Cristel (7), which utilized a system whereby the evaporators in both the chilling room and the frozen food compartment were operated from a single compressor.

Pull-Down, SERIES D-1

It is evident that the refrigerant charge in this unit is very critical due to the small receiver capacity. In view of this fact some method of automatic protection against excessive heat pressures, such as a high pressure cut-out switch, should be provided.

It is believed that the rate of heat removal was satisfactory since the temperature in the warmest part of the cabinet was lowered from approximately 73 F to 10 F in 18 hours.

The chart of the recording wattmeter did not indicate an overload of the 1/4 HP electric motor at any time during this test. This indicates that the evaporator area used was not excessive for a 1/4 HP condensing unit.

Freezing Load Tests, SERIES D-2, D-3, D-4, D-5, and D-6

The following observations are considered important in arriving at any conclusions on the performance of this refrigeration unit.

From Tables I to V inclusive (Appendix C), it will be noted that the longest time required to lower the freezing load from 70 F to 10 F was 10 hours and 59 minutes. This was in Series D-2 which had no storage load in the freezer cabinet. The arithmetic average time in this series for all packages to reach 10 F was 9 hours and 55 minutes. The shortest recorded time for any of the packages to reach 10 F was 7 hours and 40 minutes which was in Series D-3 with 277 pounds of storage load on the shelves. The ASRE Food Freezer Standards (1) states that the freezing load shall attain a temperature of 10 F in 20 hours. The unit definitely had sufficient capacity to meet these standards since the maximum time required for any package to reach 10 F was 10 hours and 59 minutes (Table I, Appendix C).

Considerable variation will be noted in the time required for the temperature of the packages to reach 32 F. The position of the package on the freezing plate did not seem to influence the elapsed time for the different loading conditions, however, shifting the storage load from the shelves to the plate gave an apparent trend of shortening the time required for this heat removal.

A rather wide range is shown in the time required to remove the latent heat of fusion and reach 30 F. This most likely was due to the impurities in the water and the differences in the contact areas between the plates and the freezing packages. This is shown more clearly by comparing the standard deviations, as shown in Table IX (Appendix C), for these tests. After approximately 13 hours of operation the freezing packages had all reached temperature stabilization at approximately 8 F. The cycling of the refrigerating equipment caused a temperature fluctuation in the load, of approximately 2 F, for the remainder of the 24 hour test.

The freezing rates in the load tests are summarized in Table IX (Appendix C). It will be noted that the average time to reach 10 F varied from 8 hours 40 minutes to 9 hours 55 minutes. The comparatively high standard deviation of 38.6 minutes, in the time for the freezing load to reach 10 F, might be attributed to the change of expansion valves between tests in Series D-2. The author feels that less significance should be attached to the results of these tests since they were performed with no storage load which is a condition that will seldom be encountered in actual operation. The arithmetic average times required to lower the load temperature to 10 F in the remaining tests show a range of 26 minutes with a maximum standard deviation of 29.8 minutes.

A comparison with the data obtained by Cristel (7) and shown on Table VI (Appendix C) gives an average time of 15 hours 7 minutes with a standard deviation of 202 minutes to lower the load in 12 packages, in four different tests, from 60 F to 10 F. This indicates approximately 6 hours more time required to remove less heat. Operating conditions of plate spacing, storage load, freezing load, and ambient temperature used in obtaining data for Table VI (Appendix C) were the same as used in Series D-4. However, the freezing plates were of different design.

The average power consumed in the tests in Series D-3, D-4, D-5, and D-6 varied approximately in the same proportion as the time required for removing the heat from the loads. Table VII (Appendix C) shows that the average power consumption for these tests varied less than 0.2 Kw-hr in an arithmetic average operating time of 20 hours 32 minutes.

The cycling of the refrigeration equipment, in each of the 24 hour freezing load test periods, seemed to be effected by the amount and placement of the storage load (Table VII, Appendix C). When 277 pounds of shelf storage load was used, the arithmetic average was 14.4 operating cycles. When this shelf storage load was increased to 554 pounds the number of cycles decreased to an arithmetic average of 11.8. Transferring 174 pounds of storage load to the holding plates, the arithmetic average decreased to 8.4 operating cycles. When the holding plate storage was increased to 290 pounds there was very little change in the number of operating cycles. Observations of the pressure recordings for all the freezing load tests indicated continuous operation of the condensing unit for approximately 13 hours after the load was placed on the freezing plates. For the remaining 11 hours of the tests, the condensing unit cycled in approximately equal periods of operation and idleness.

Temperature variations in the compartment and load are shown by Table VIII (Appendix C). A fluctuation in storage load temperature of approximately 2 F was obtained with 277-pound storage load and on each of the tests using plate storage. With the full storage load on the shelves, the maximum fluctuation was only 1 F. Table VIII (Appendix C) does not give the maximum fluctuation for the plate storage in Series D-5 and D-6. An inspection of Charts 17 to 26 inclusive (Appendix A) shows a variation of approximately 5 F in the plate storage load in the first 12 hours of the test when one layer of storage was on the plates with a slightly less variation in temperature when the second layer was added. The second layer of plate storage was slower than the contact layer in attaining the maximum temperature rise during the freezing period, with this lag also evident in the period following freezing when the temperatures were decreasing. These changes in temperatures are shown as curves 8 and 9 on Charts 22 to 26 inclusive (Appendix A).

The air temperature inside the compartment decreased at approximately the same rate for all tests except when maximum plate storage was used.

CONCLUSIONS

It must be emphasized that the conclusions on certain phases of this broad investigation are based on possible trends or tendencies that need further investigation. With these limitations in mind the data obtained was evaluated and the following conclusions appear to be warranted.

(1) An integral refrigeration unit for the zero storage compartment of a walk-in type refrigerator was designed using standard practices heretofore followed.

(2) Standard sizes of refrigeration equipment was installed with tools commonly used in the refrigeration trade.

(3) Fabrication of the supporting frames by metal workers would be advisable.

(4) This unit was installed without special handling devices.

(5) The assembly was designed so that it could be transported provided the care normally afforded refrigeration units was given it.

(6) Vibration of the unit caused no apparent difficulties with the vapor seal or other structural features.

(7) Due to the small receiver capacity such a unit should be provided with a high pressure control.

(8) A comparison of the refrigerant pressures and the temperatures maintained in the freezer compartment gave an arithmetic average temperature differential of approximately 6 F to 8 F for test periods.

(9) The rate of heat removal from the product load was faster with the 1/4 HP integral refrigeration unit than for a 1/2 HP unit furnishing

all the refrigeration for the complete walk-in cabinet, which included an additional chill room load of approximately 800 Btu per hour. This faster freezing rate for the smaller unit might be attributed to its continuous operation whereas the larger unit had cycles of operation. The different types of evaporator plates used for the two units might also contribute to this faster freezing rate obtained by use of the smaller unit.

(10) Doubling the amount of shelf storage load increased the arithmetic average time required for freezing by approximately 25 minutes.

(11) Transferring one layer of the shelf storage to the holding plates apparently did not effect the time required for freezing.

(12) When two layers of storage load was transferred to the holding plates the arithmetic average time required for freezing was decreased by approximately 21 minutes.

(13) Storage on the holding plates effected the cycling of the condensing unit after approximately minus 4 F was reached, thereby, giving longer periods of operation, with fewer cycles for the remainder of the 24 hour test period.

(14) Transferring storage load to the holding plates seemed to increase the amount of power consumed in the 24 hour test periods, with this increase evident in both plate load conditions. The increase was approximately 0.2 Kw-hr which was approximately four per cent of the total power consumed.

(15) A maximum variation of 2 F in the temperature of the shelf

storage load was observed. Transferring part of this load to the holding plates apparently did not effect this temperature variation.

(16) Maximum variation of plate storage temperature was 5 F with one layer of storage on each holding plate.

(17) The arithmetic average operating time was increased approximately 52 minutes, when either 174 pounds and 290 pounds of the storage load was transferred to the holding plates.

SUGGESTIONS FOR FUTURE STUDY

This study indicated that an integral refrigeration unit for the zero storage compartment of a walk-in type farm refrigerator is feasible. However, definite improvements that can be made on the installed unit are evident. The tests conducted on the unit indicate a definite need for further study and observations during the conduct of the test suggest other technicalities that would possibly warrant further study. The author, therefore, suggests that future research be conducted along the following lines.

(1) Simplify the design of the frame used for supporting the evaporator plates.

(2) Investigate the possibility of replacing the drier coil with a heat exchanger.

(3) Design, construct, install, and test such a unit for use in the chill room.

(4) Study the effect that overload conditions might have on the unit.

(5) Determine more accurately the temperature difference maintained between the refrigerant and the air inside the freezer.

(6) Investigate in more detail the effect of placement of the storage load on the performance of the unit.

(7) Investigate the possibility of replacing the expansion valve with a capillary tube.

ACKNOWLEDGEMENT

To the Virginia Agricultural Experiment Station, recognition is given for supplying the necessary materials and equipment used in the study. The staff of the VPI Agricultural Engineering Department co-operated with many valuable suggestions, criticisms and instructions. The assistance of Associate Professor U. F. Earp of the Agricultural Engineering Department, the project supervisor, is cheerfully acknowledged. Without his aid the project would have been impossible.

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APPENDIX A

FREEZING RATES FOR LOAD TESTS

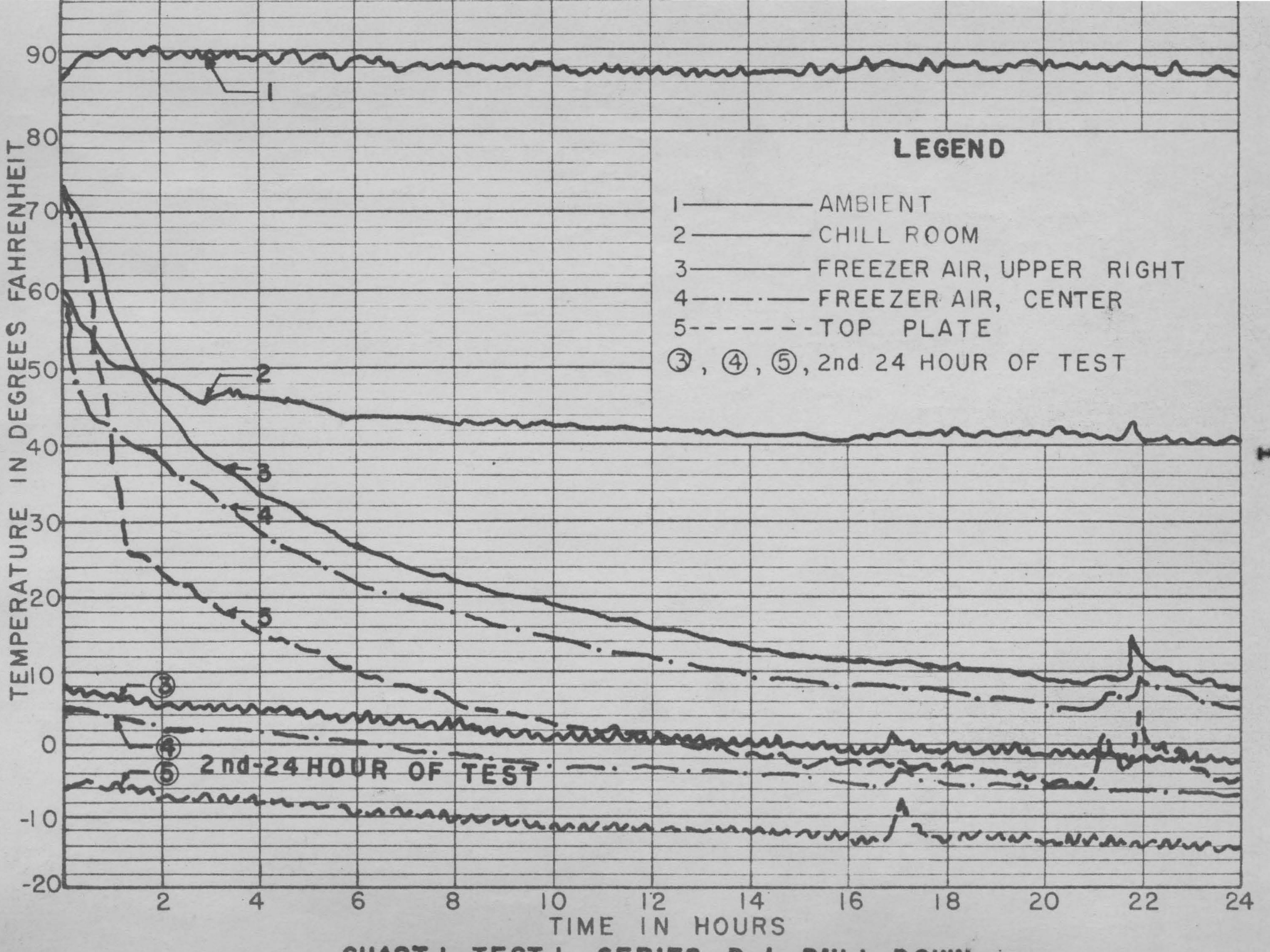


CHART I TEST I SERIES D-1 PULL-DOWN

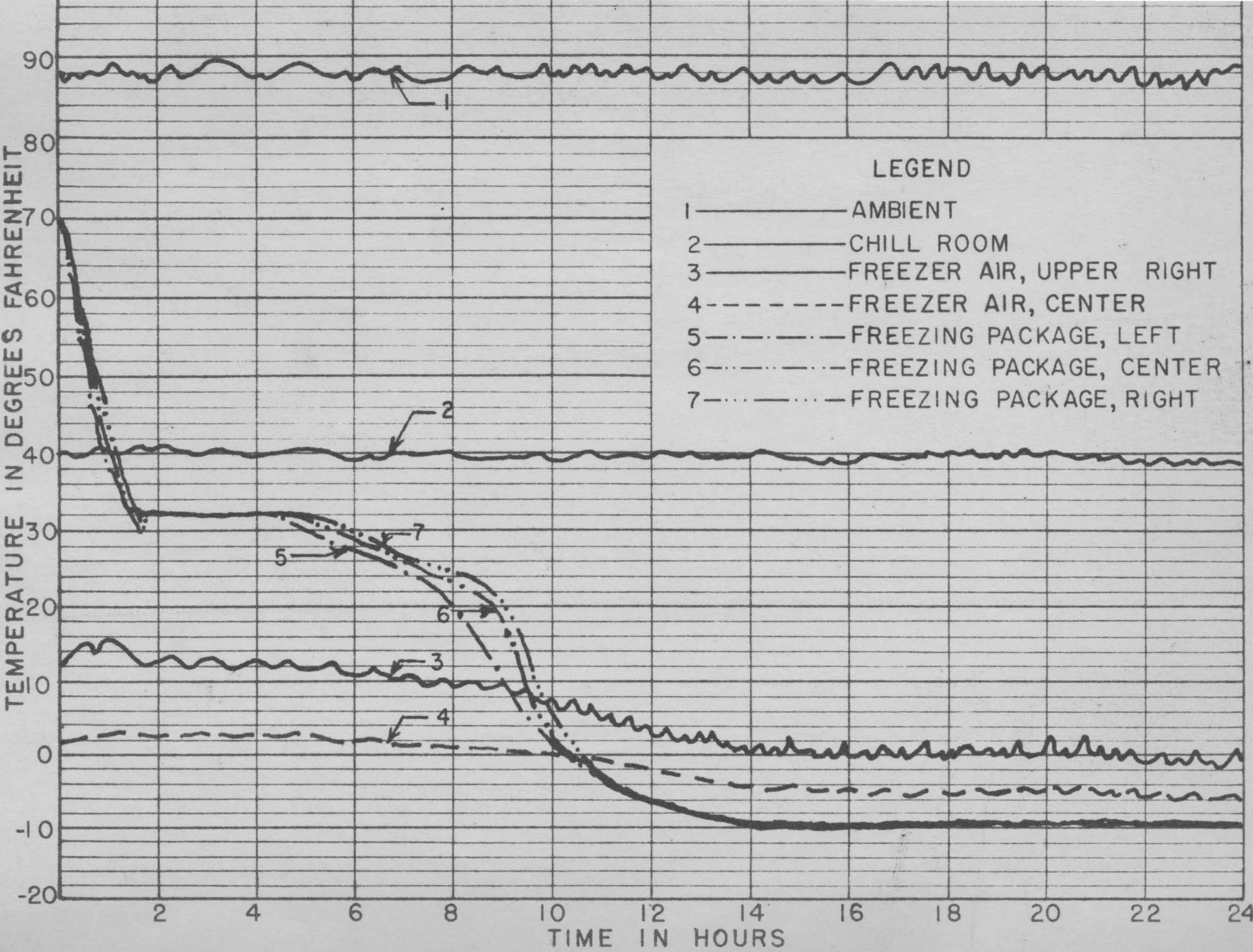


CHART 2 TEST I SERIES D-2 LOAD, NO STORAGE

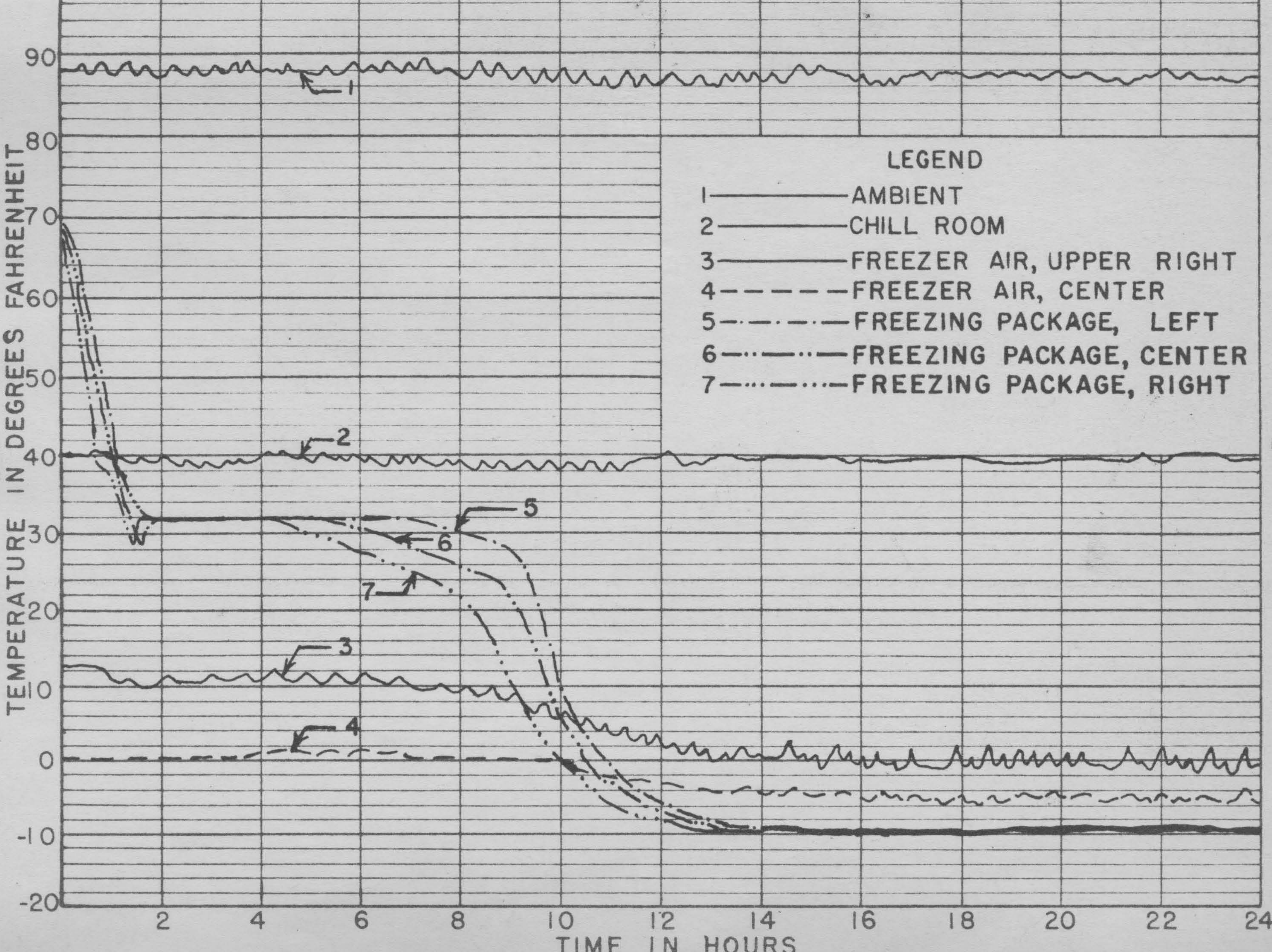


CHART 3 TEST 2 SERIES D-2 LOAD, NO STORAGE

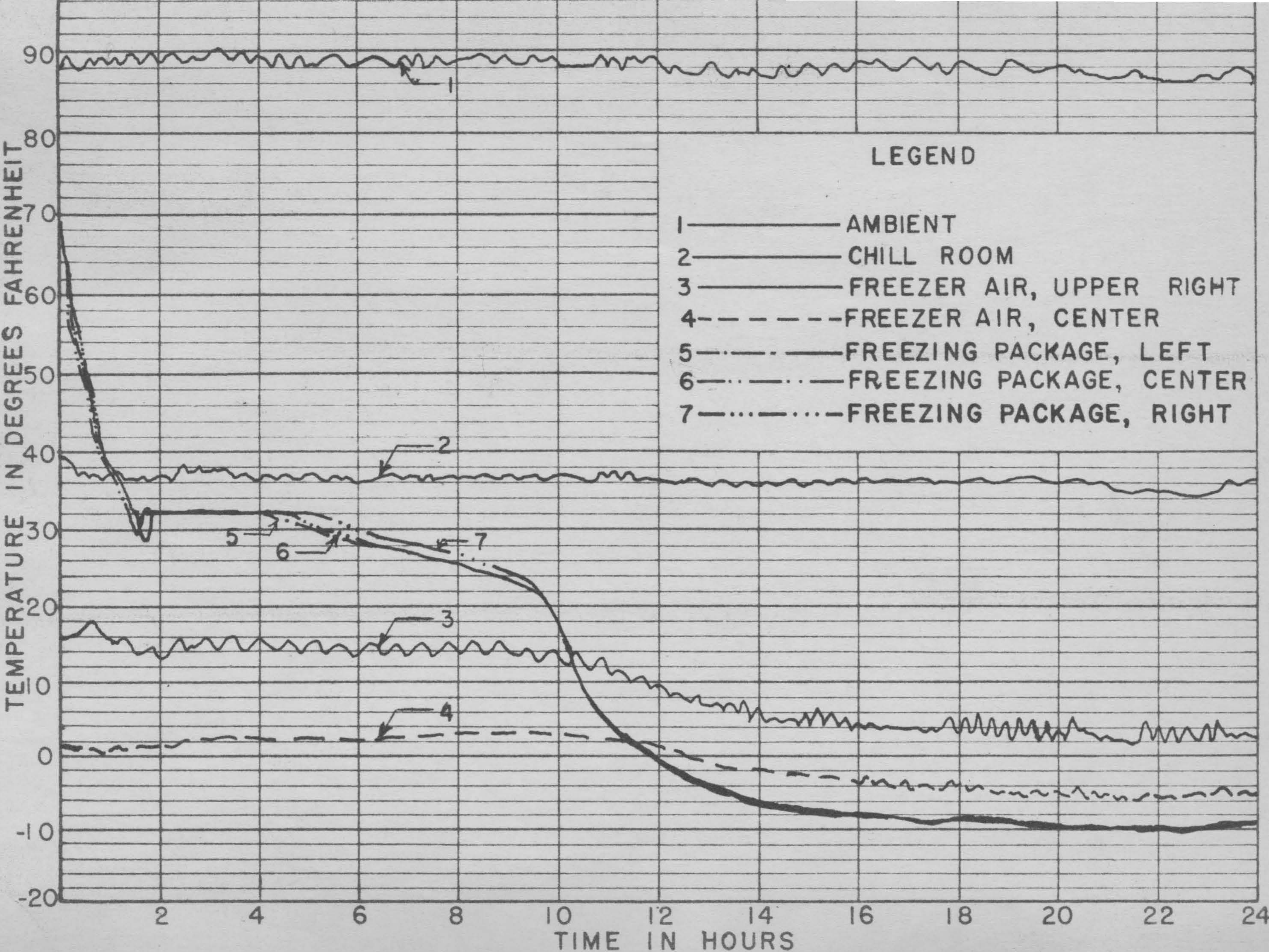


CHART 4 TEST 3 SERIES D-2 LOAD, NO STORAGE

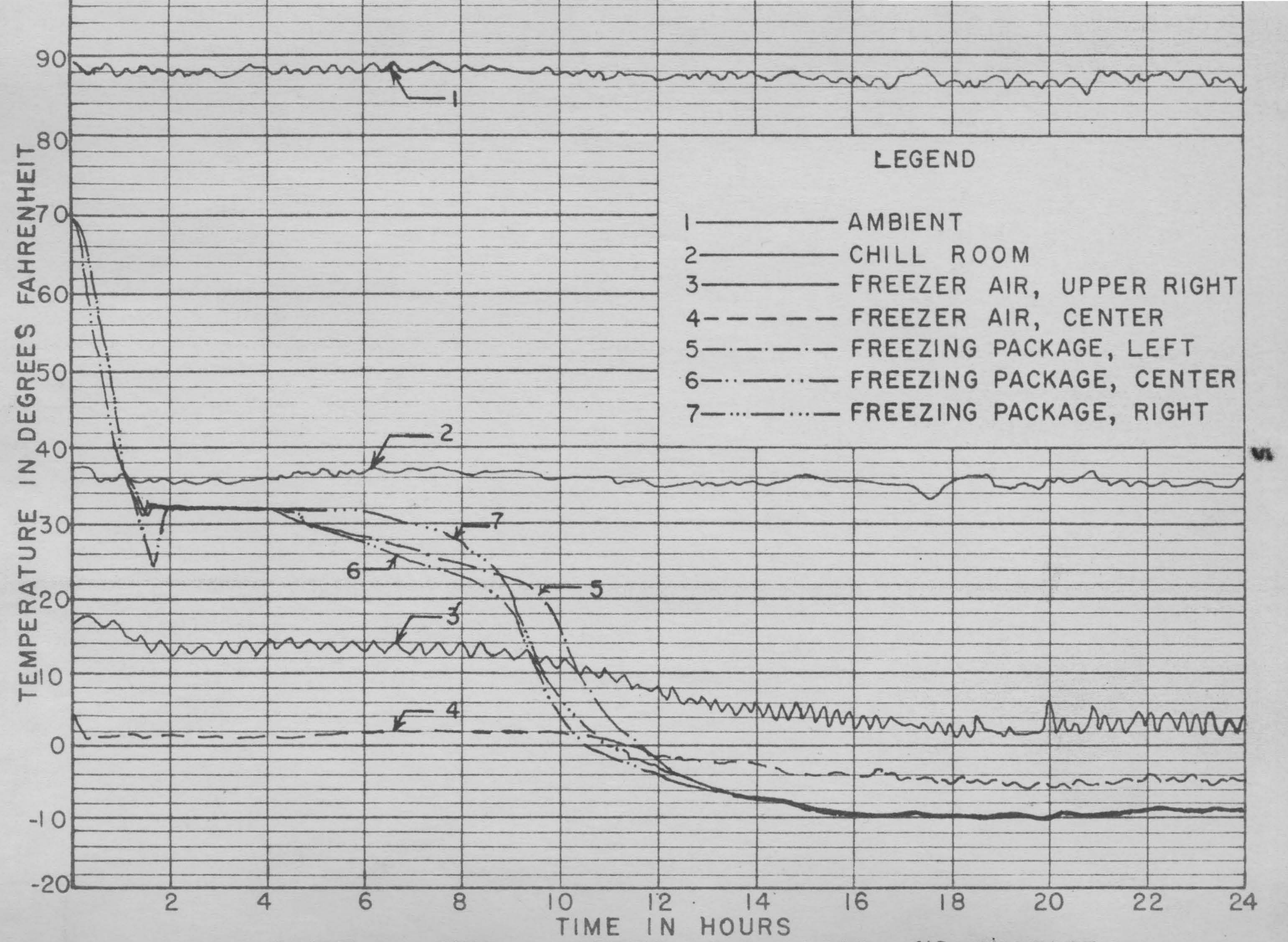


CHART 5 TEST 4 SERIES D-2 LOAD, NO STORAGE

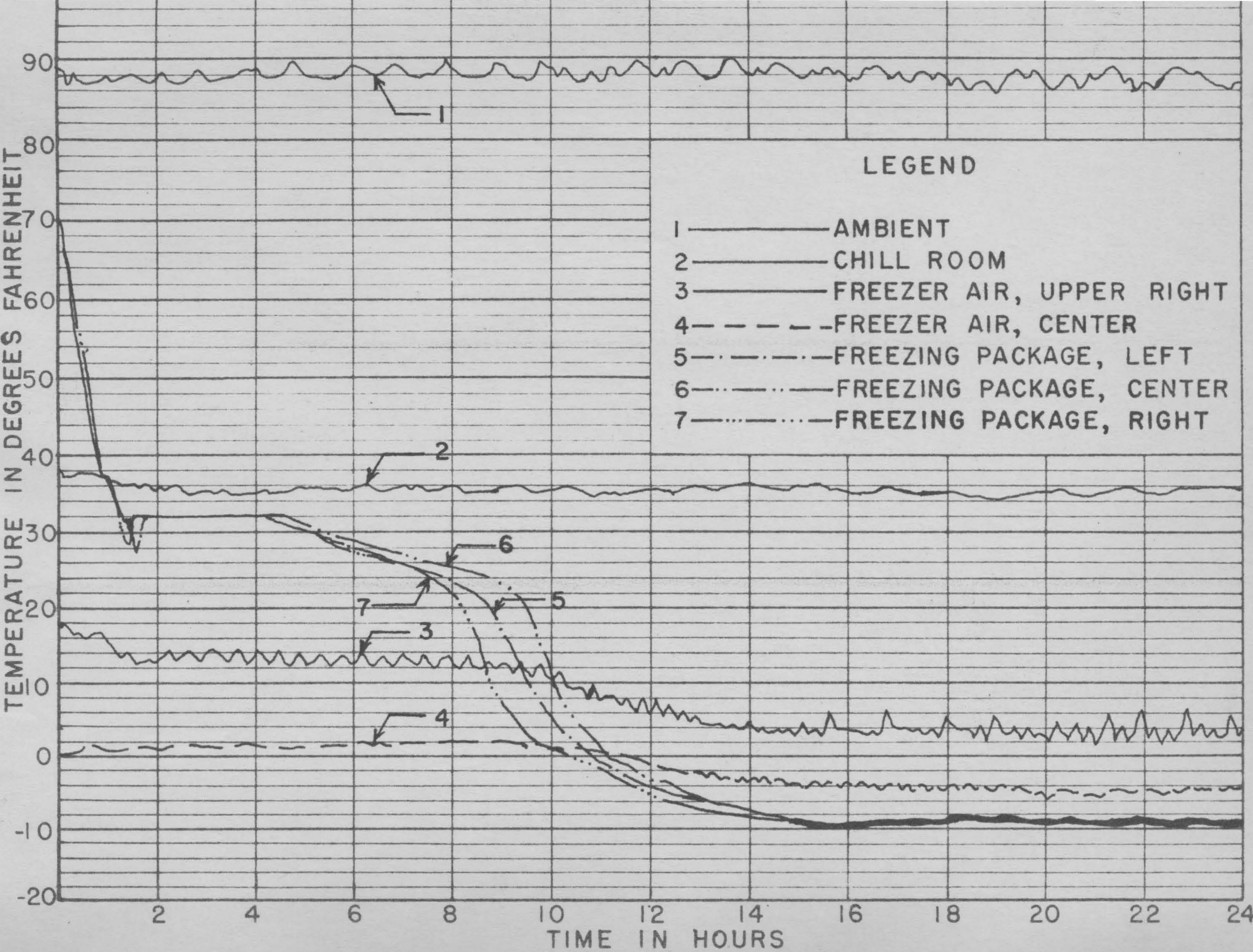


CHART 6 TEST 5 SERIES D-2 LOAD, NO STORAGE

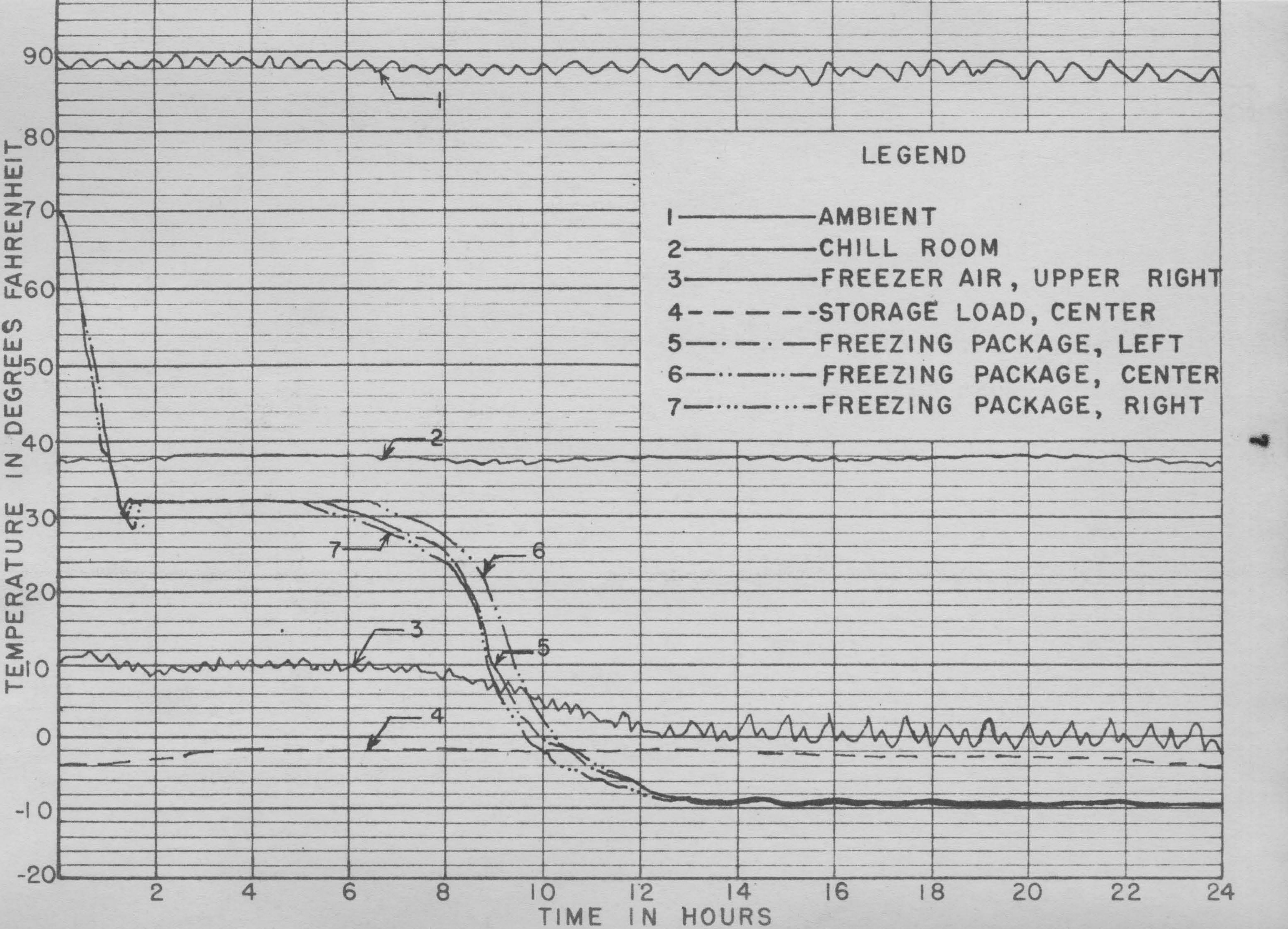


CHART 7 TEST I SERIES D-3 LOAD, ONE-HALF STORAGE

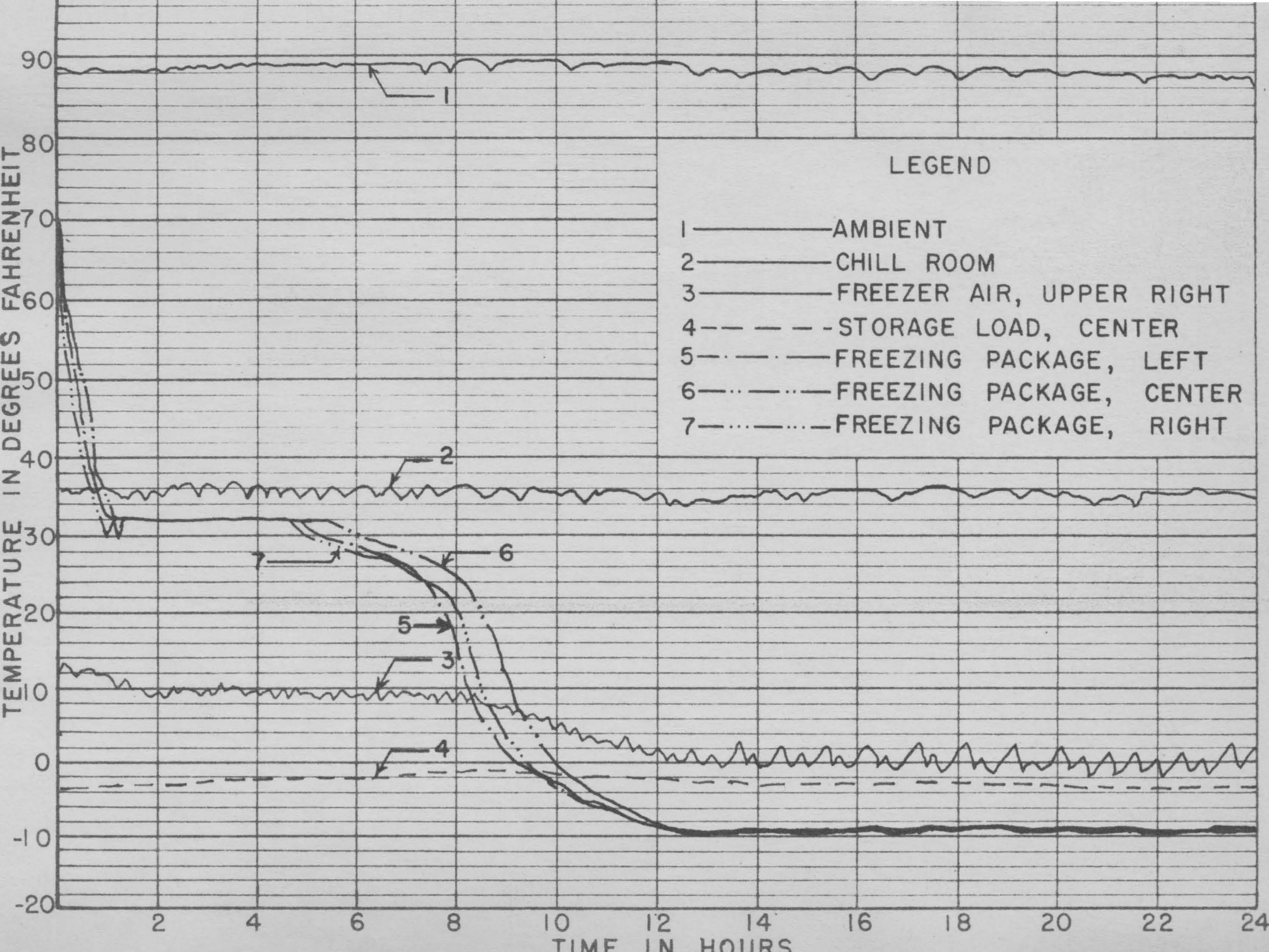


CHART 8 TEST 2 SERIES D-3 LOAD, ONE-HALF STORAGE

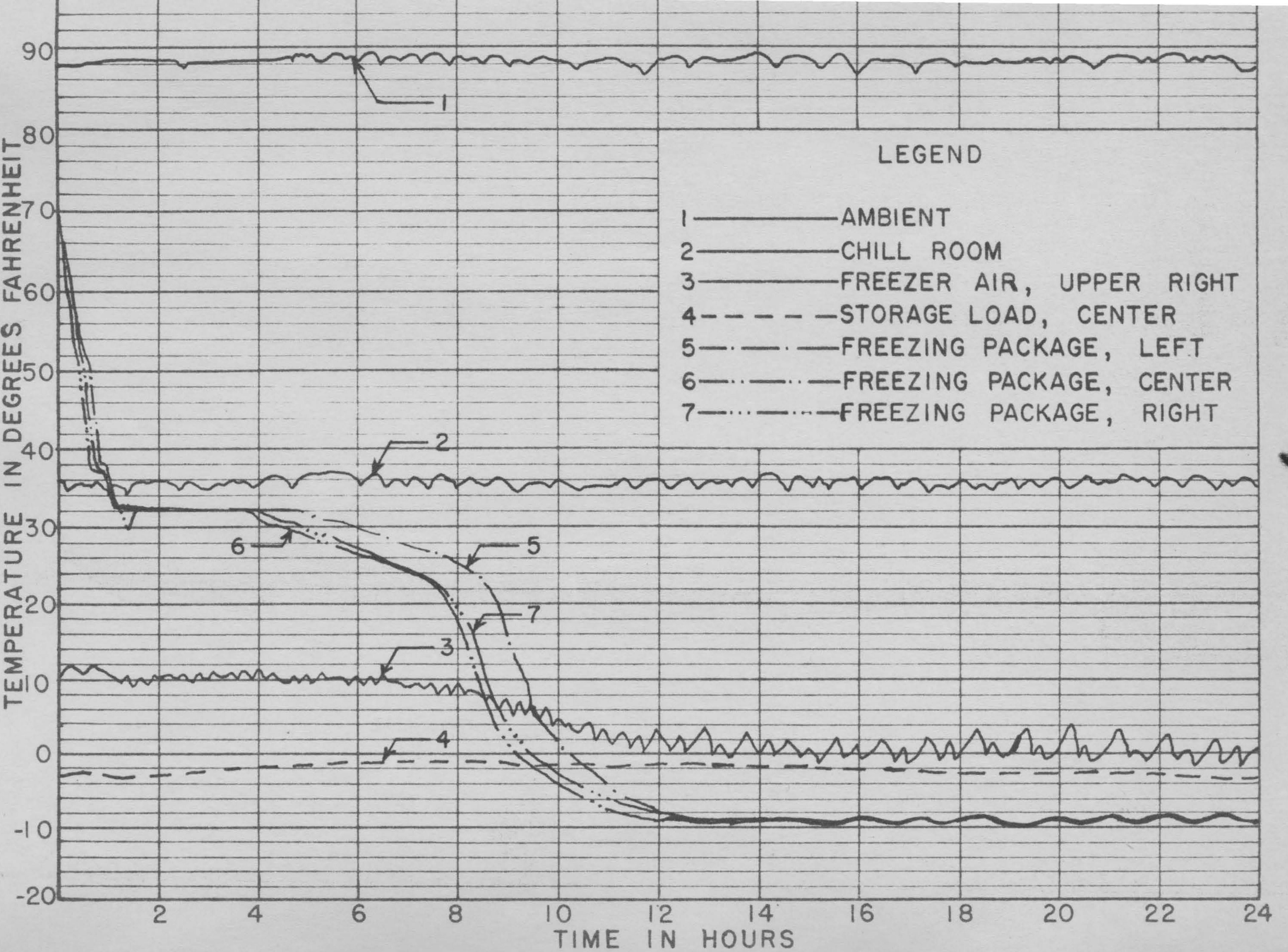


CHART 9 TEST 3 SERIES D-3 LOAD, ONE-HALF STORAGE

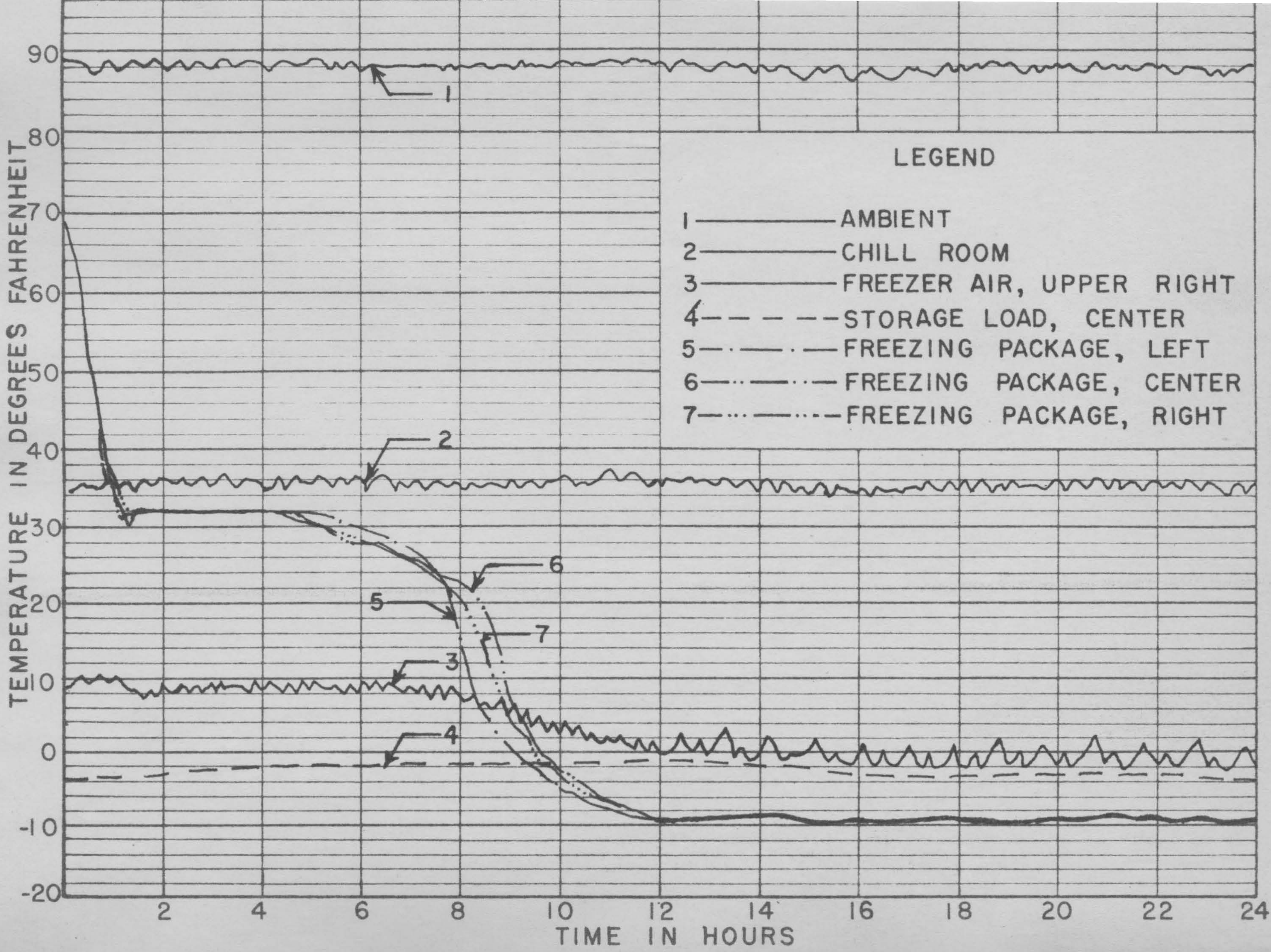


CHART 10 TEST 4 SERIES D-3 LOAD, ONE-HALF STORAGE

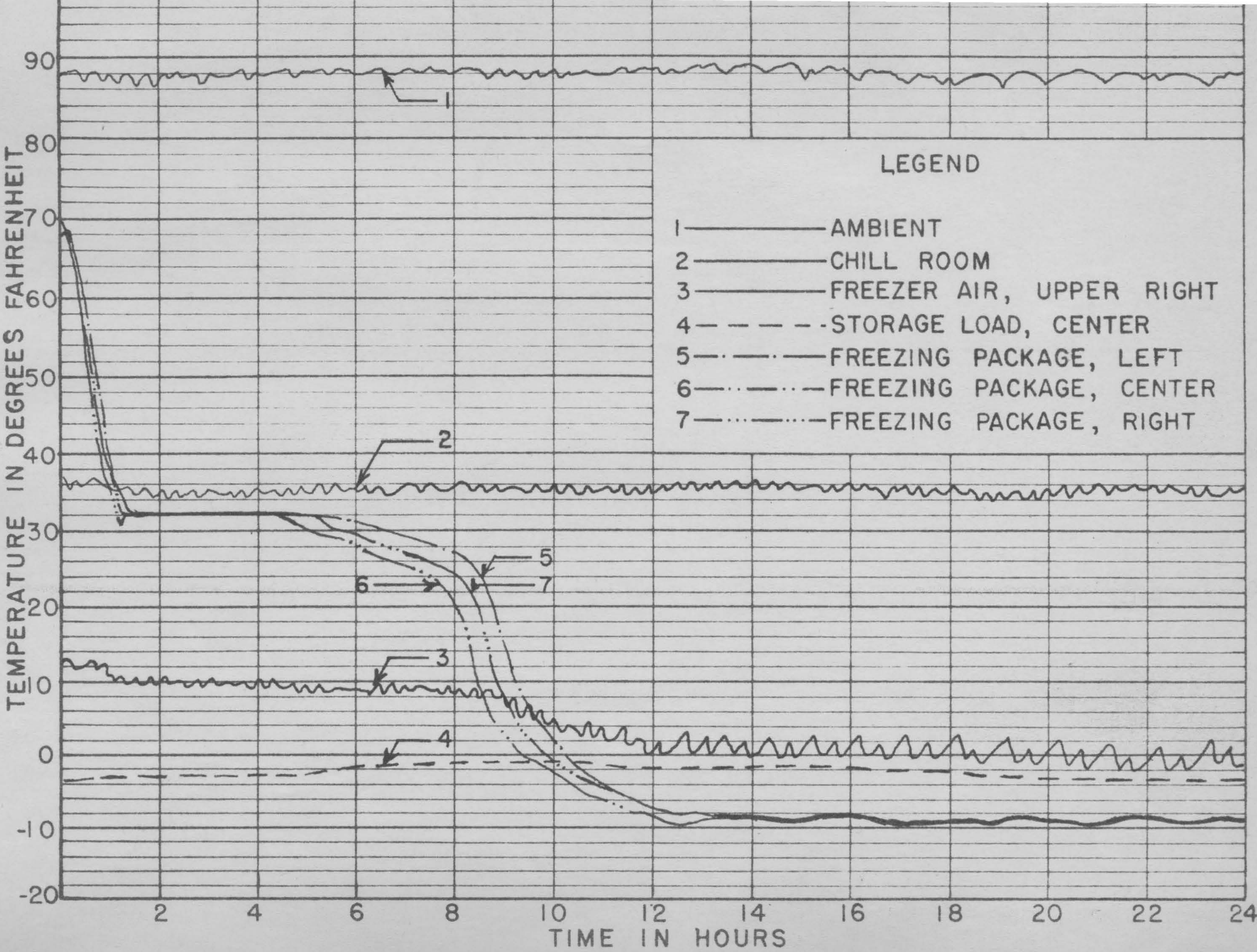


CHART 11 TEST 5 SERIES D-3 LOAD, ONE-HALF STORAGE

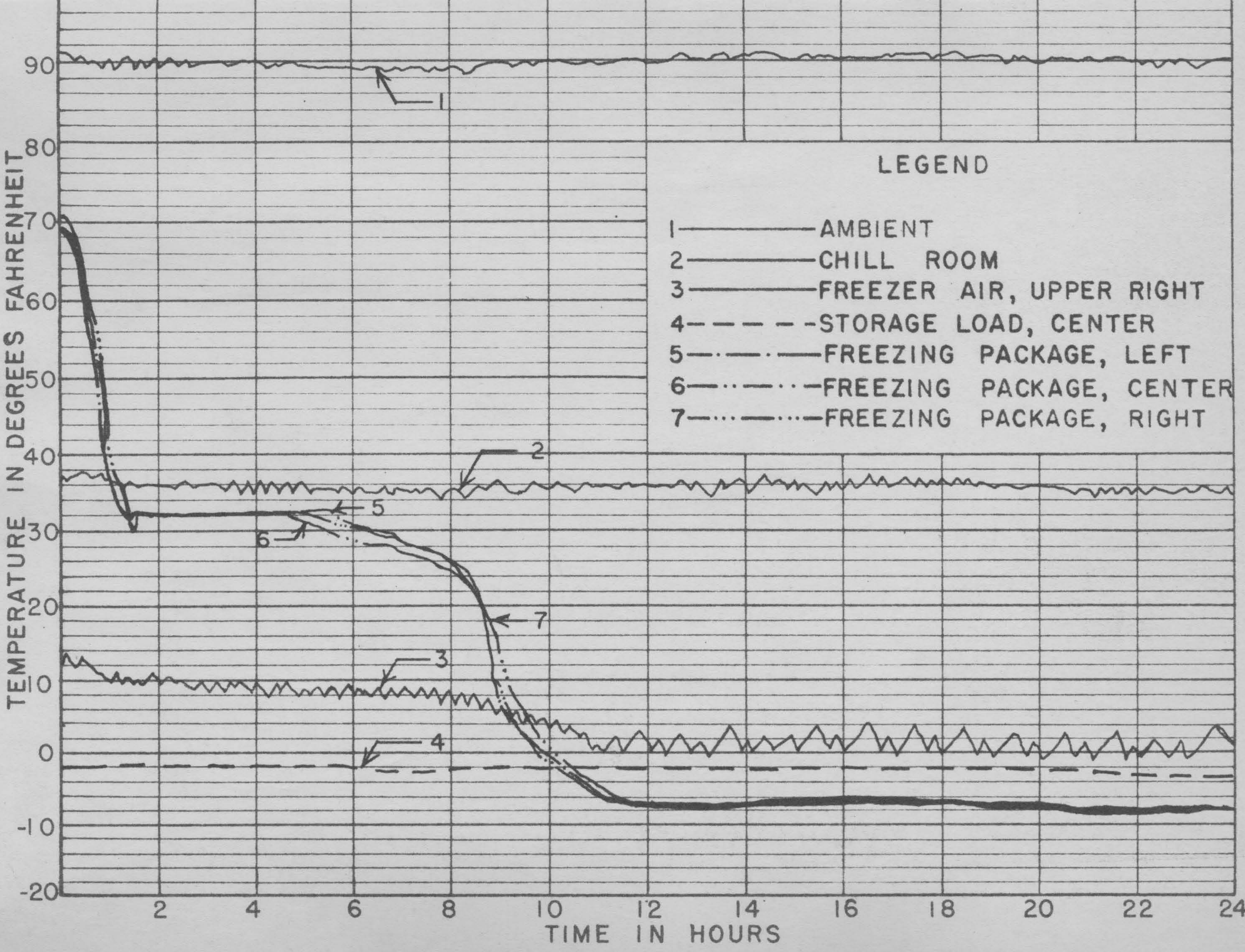


CHART 12 TEST 1 SERIES D-4 LOAD, FULL STORAGE

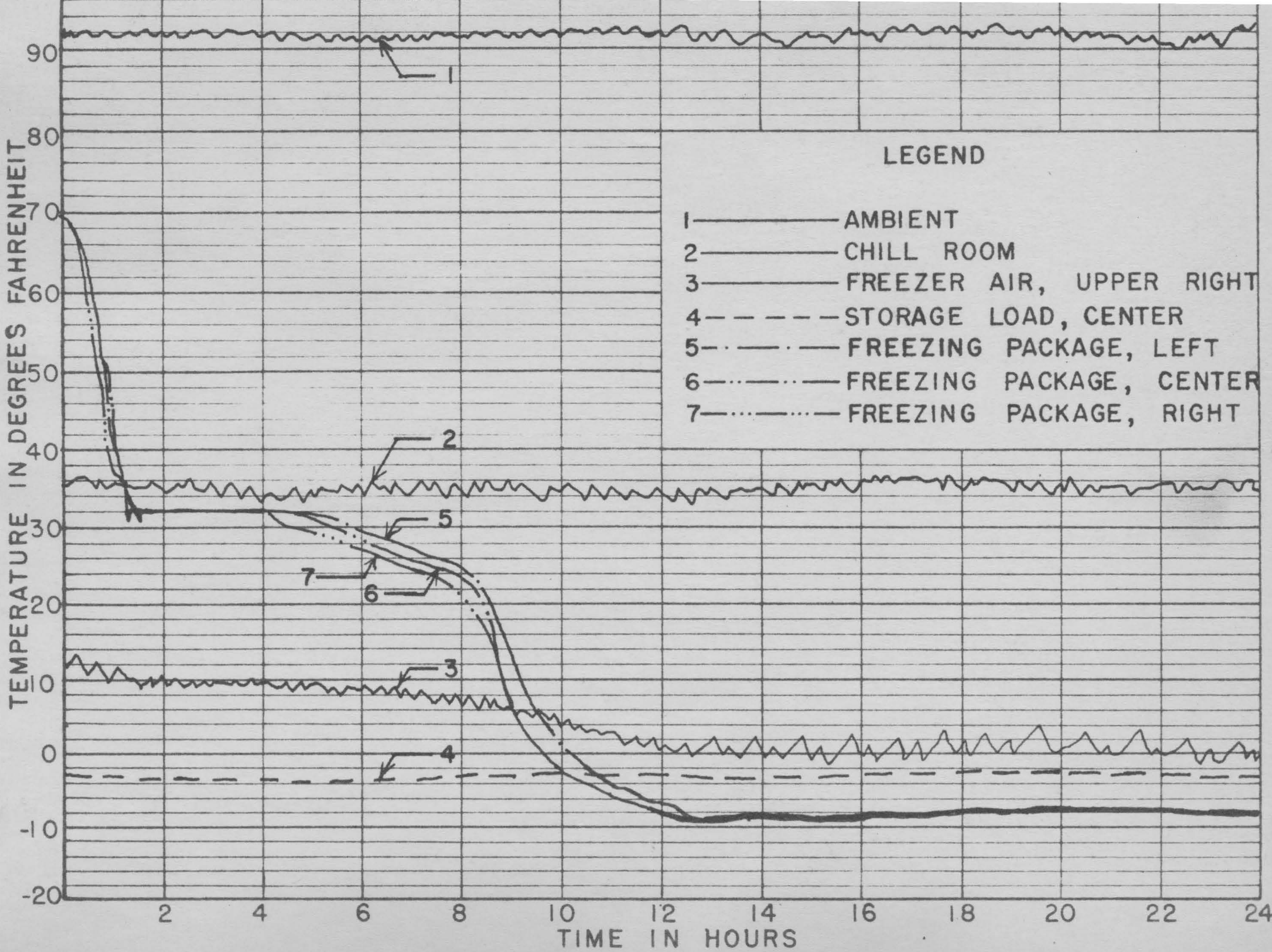


CHART 13 TEST 2 SERIES D-4 LOAD, FULL STORAGE

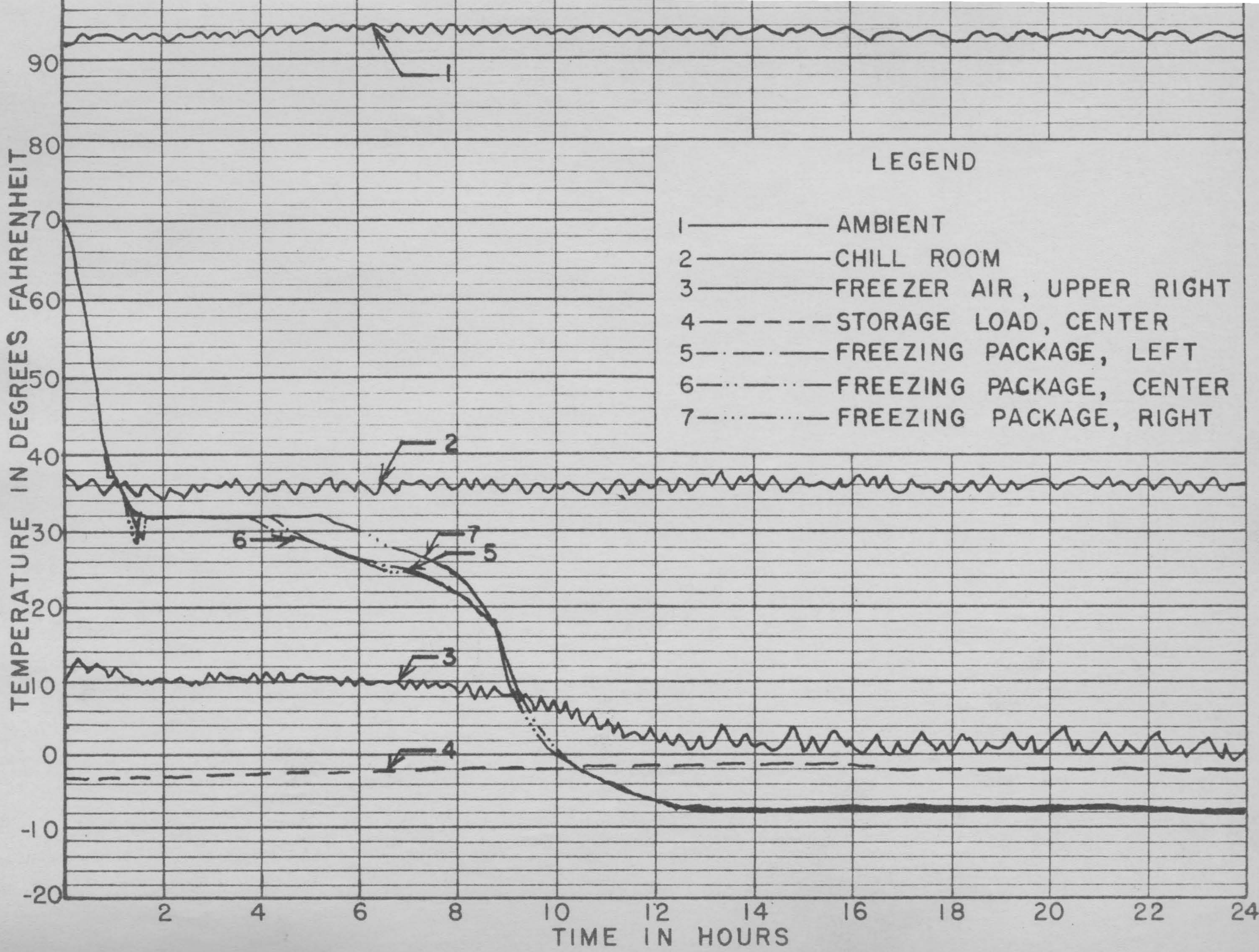


CHART 14 TEST 3 SERIES D-4 LOAD, FULL STORAGE

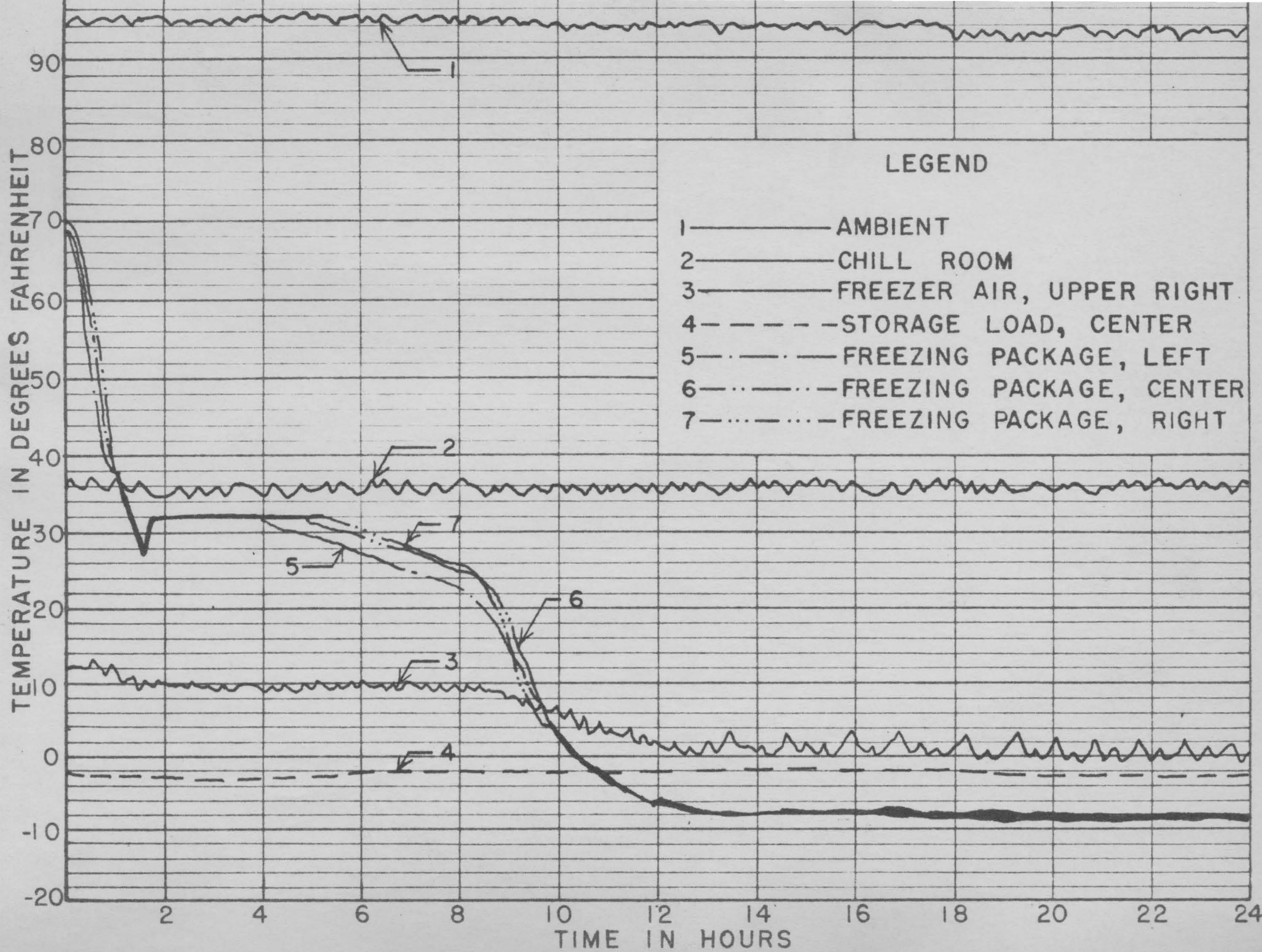


CHART 15 TEST 4 SERIES D-4 LOAD, FULL STORAGE

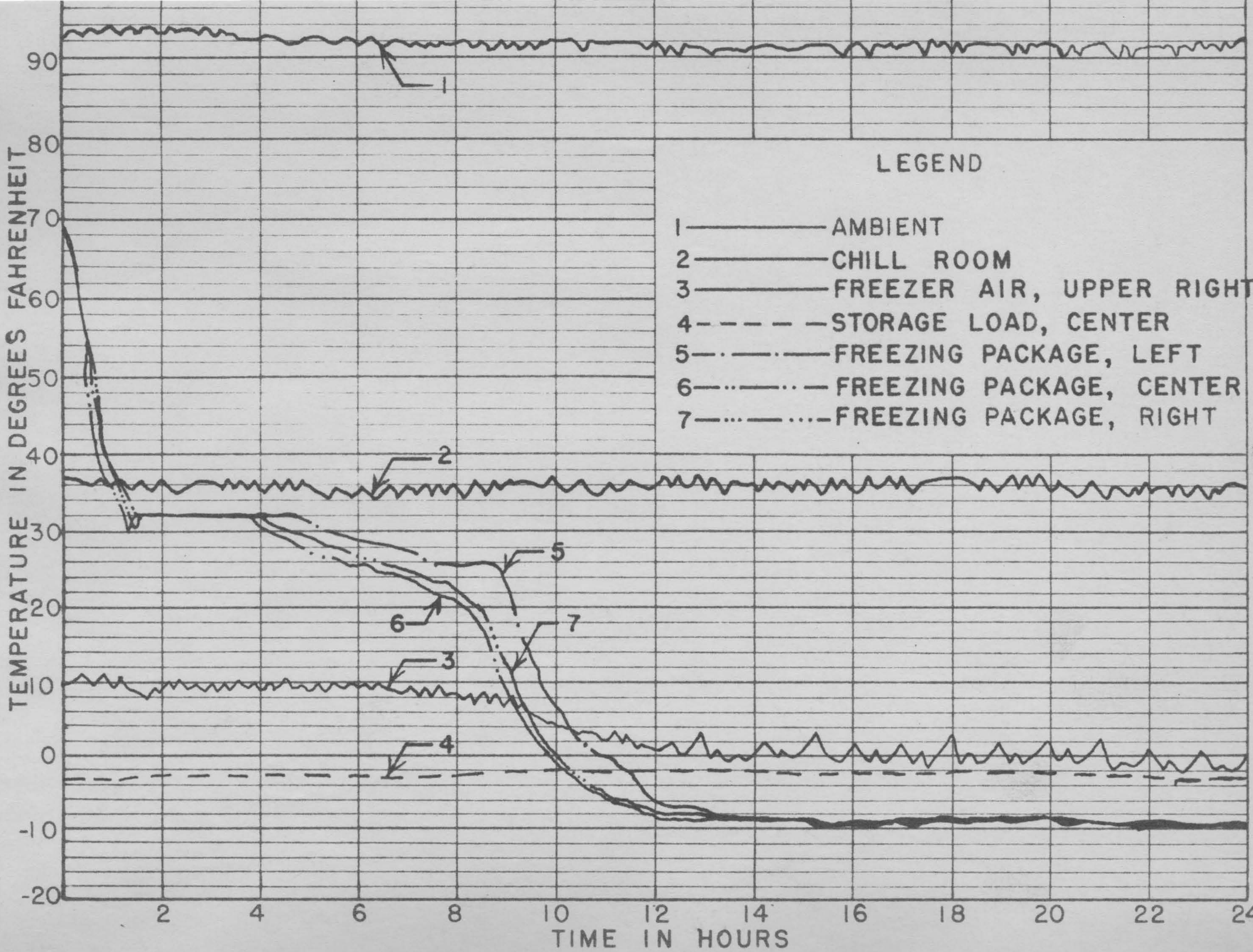


CHART 16 TEST 5 SERIES D-4 LOAD, FULL STORAGE

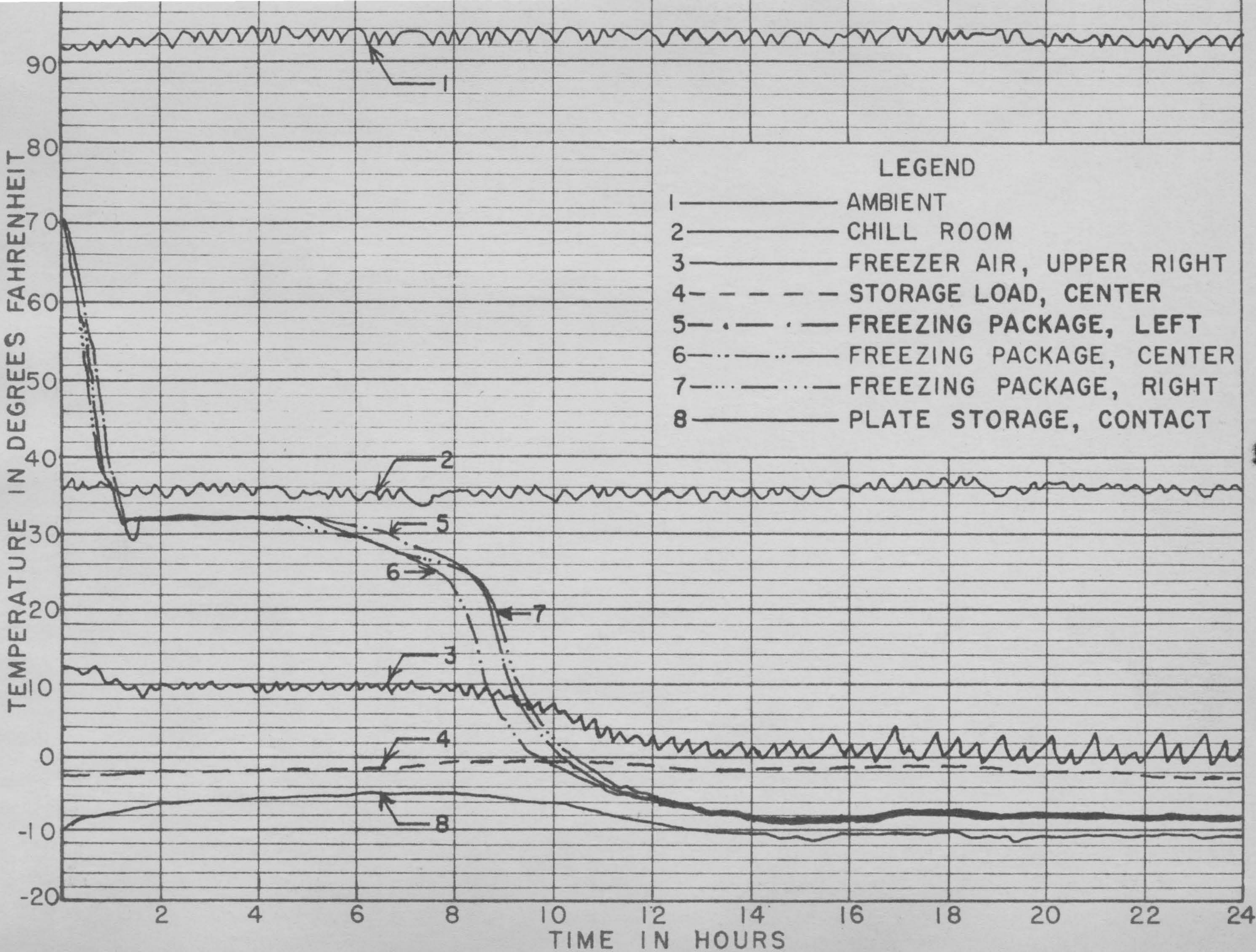


CHART 17 TEST 1 SERIES D-5 LOAD, FULL STORAGE(ONE LAYER EACH PLATE)

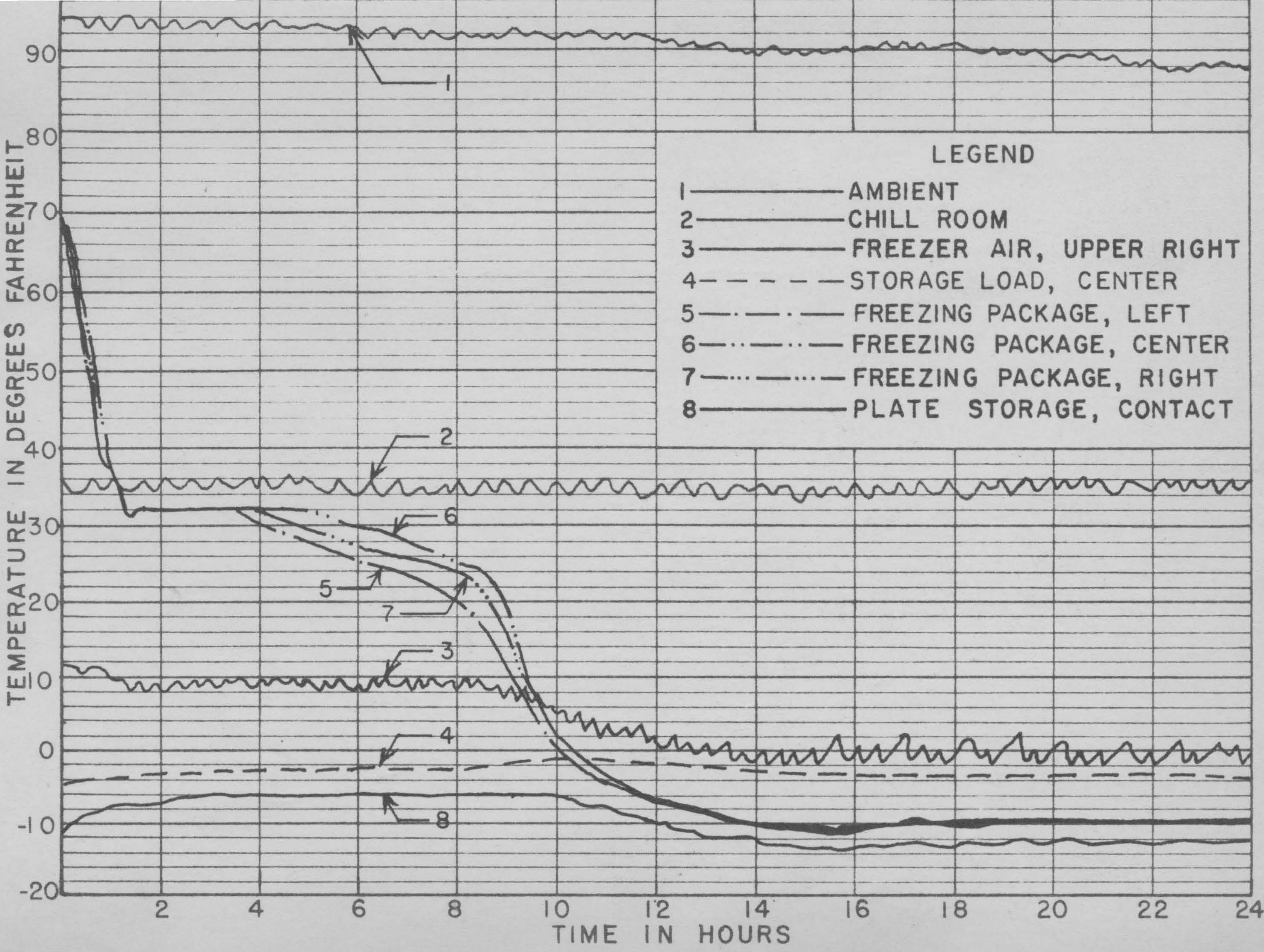


CHART 18 TEST 2 SERIES D-5 LOAD, FULL STORAGE (ONE LAYER EACH PLATE)

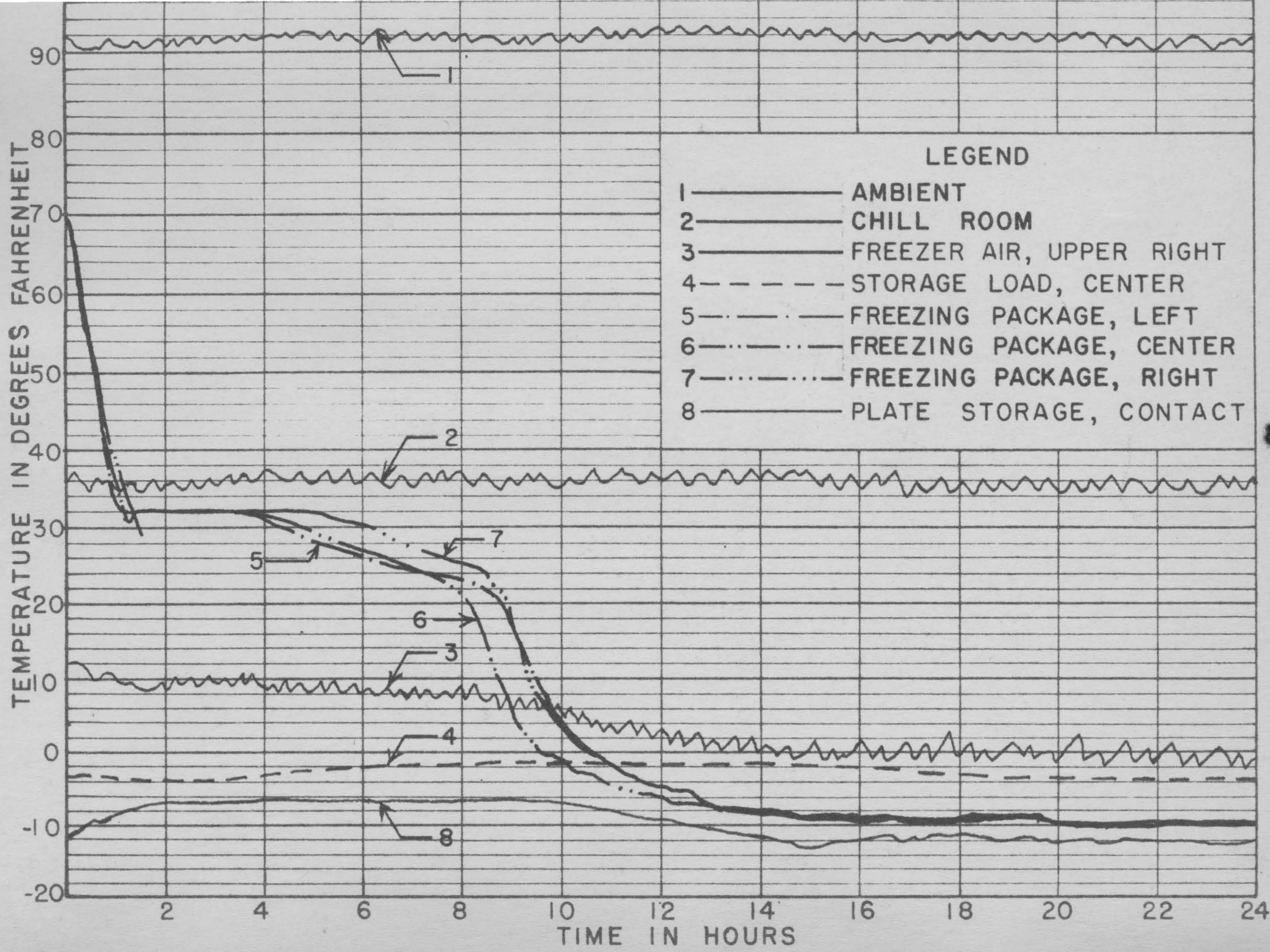


CHART 19 TEST 3 SERIES D-5 LOAD, FULL STORAGE (ONE LAYER EACH PLATE)

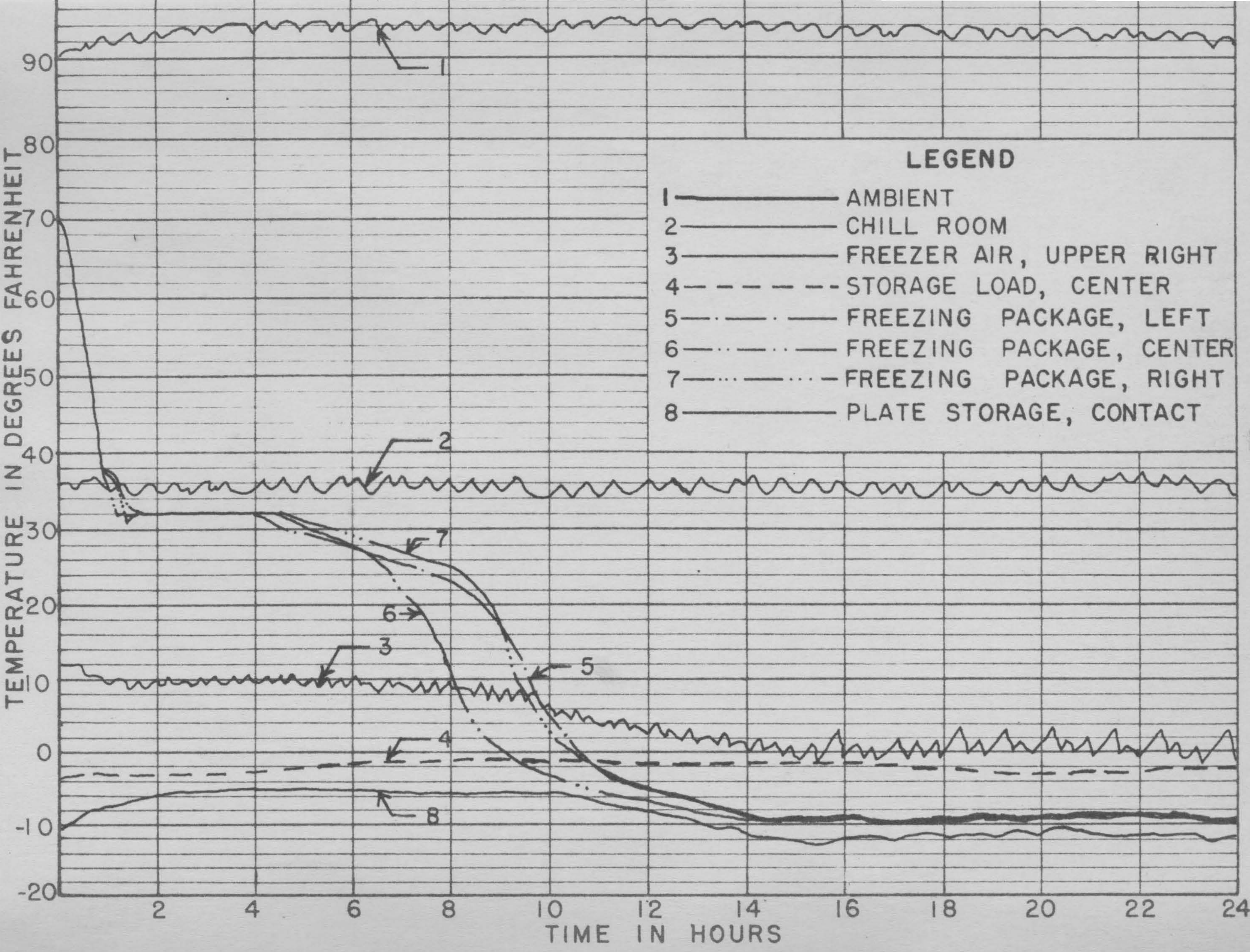


CHART 20 TEST 4 SERIES D-5 LOAD, FULL STORAGE (ONE LAYER EACH PLATE)

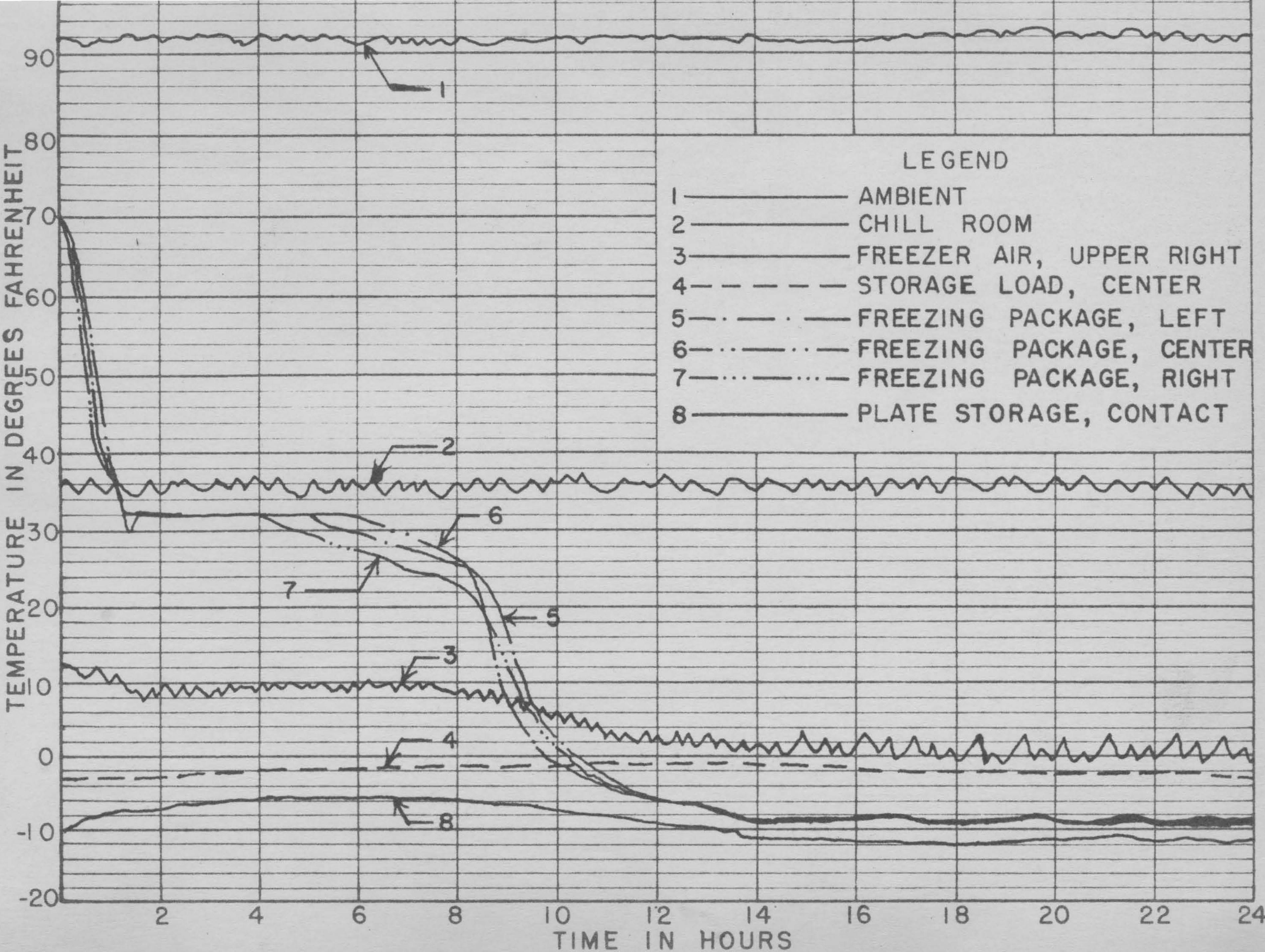


CHART 21 TEST 5 SERIES D-5 LOAD, FULL STORAGE (ONE LAYER EACH PLATE)

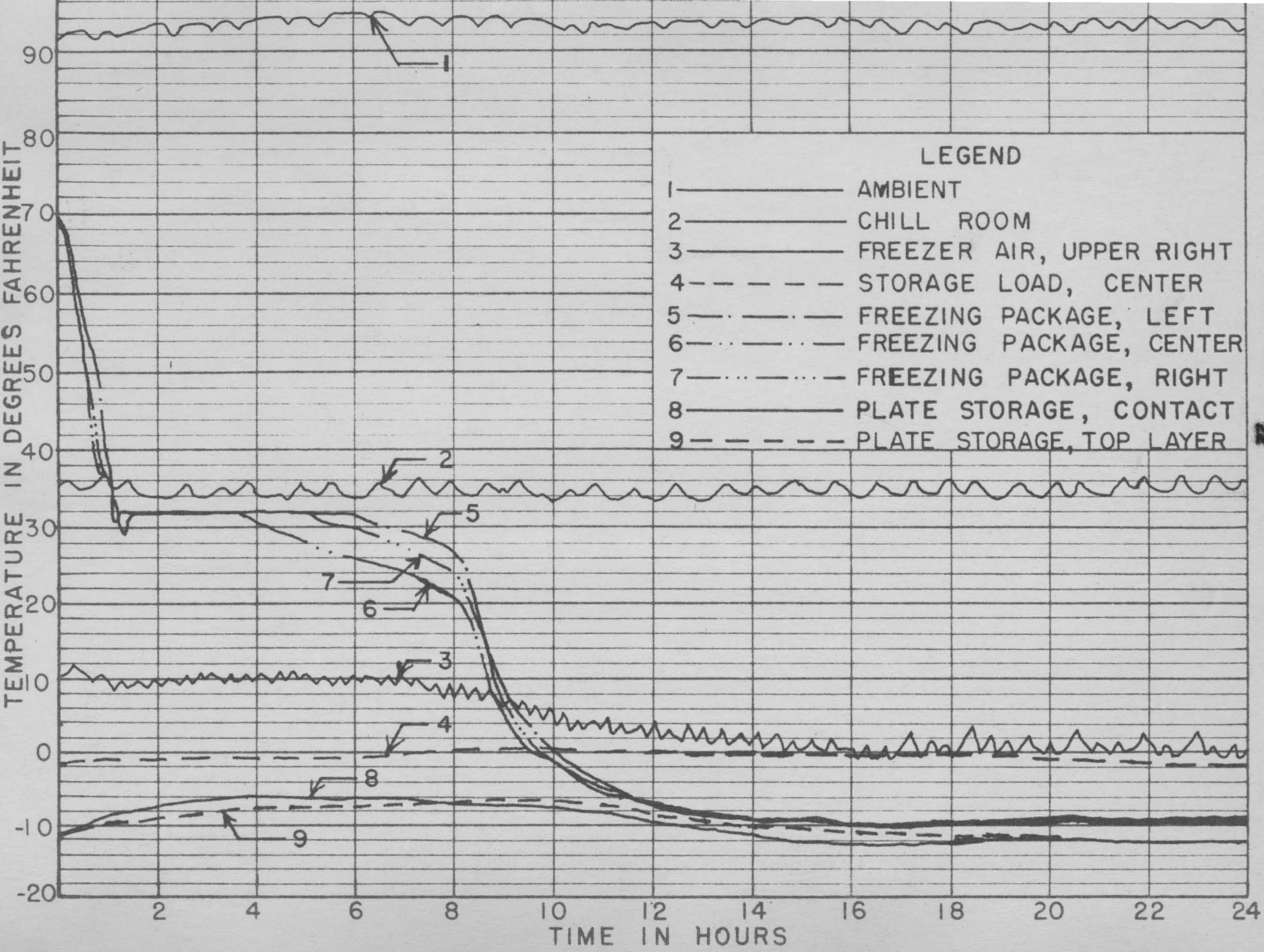


CHART 22 TEST I SERIES D-6 LOAD, FULL STORAGE (MAX. PLATE)

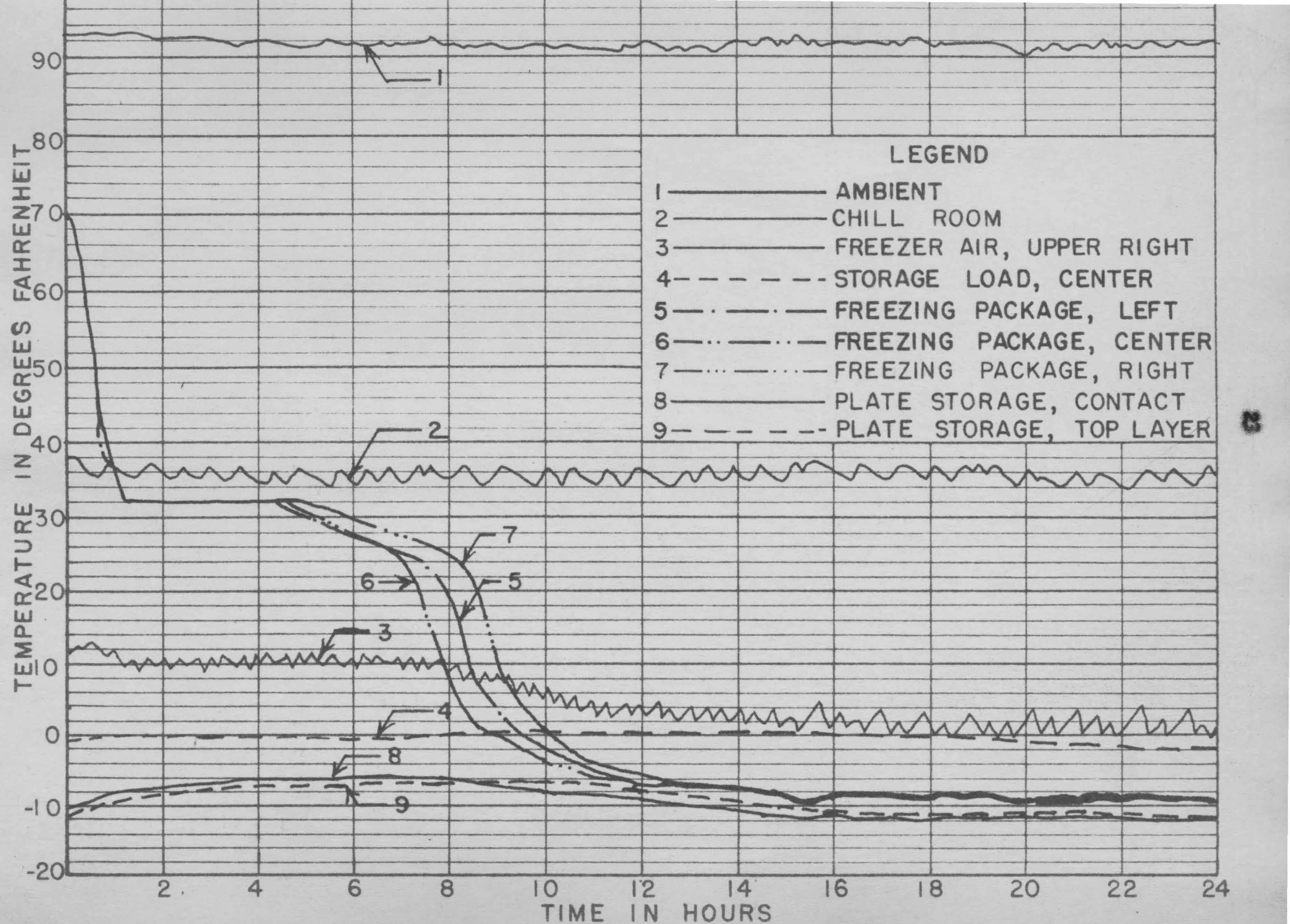


CHART 23 TEST 2 SERIES D-6 LOAD, FULL STORAGE (MAX. PLATE)

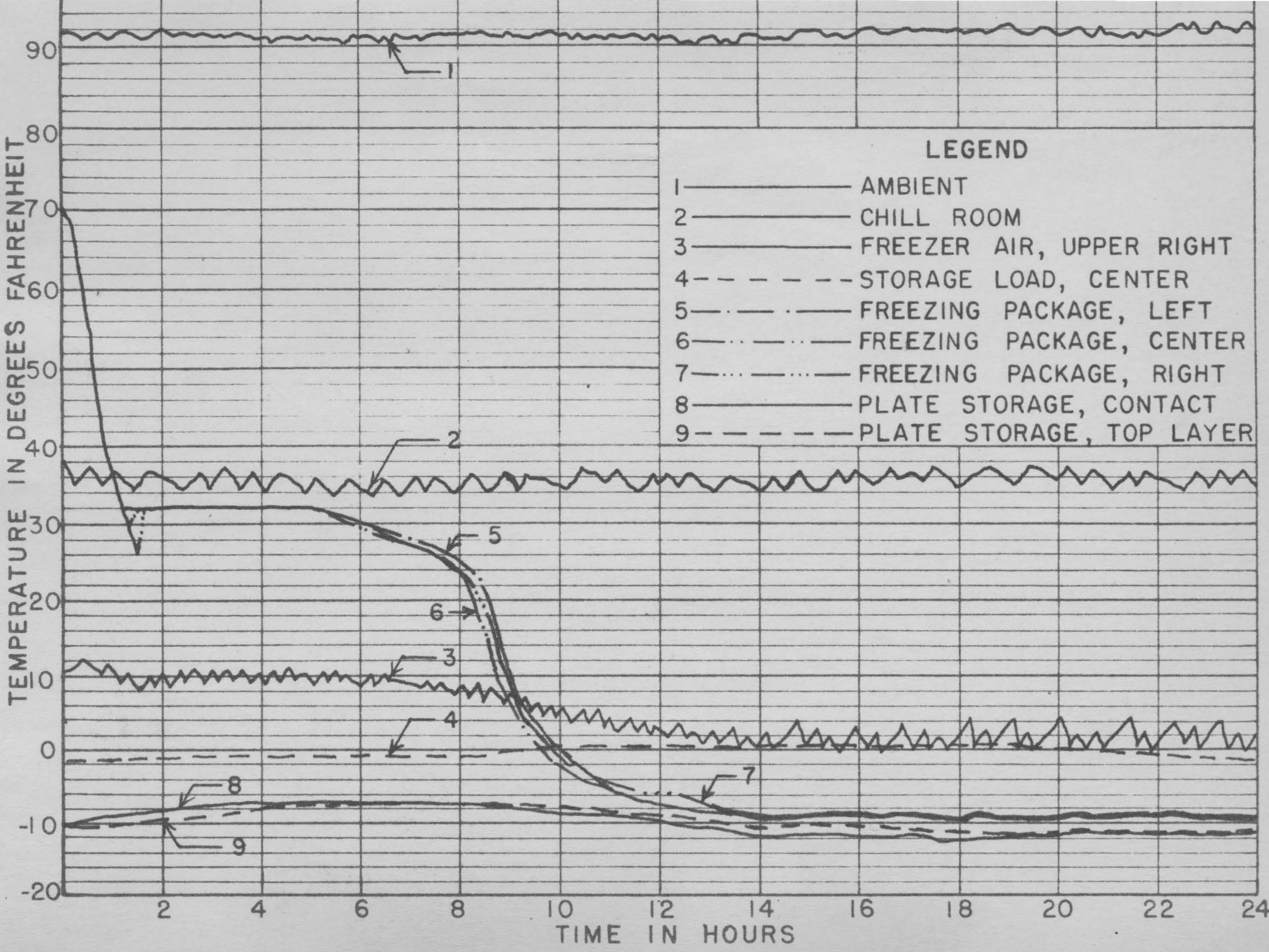


CHART 24 TEST 3 SERIES D-6 LOAD, FULL STORAGE (MAX. PLATE)

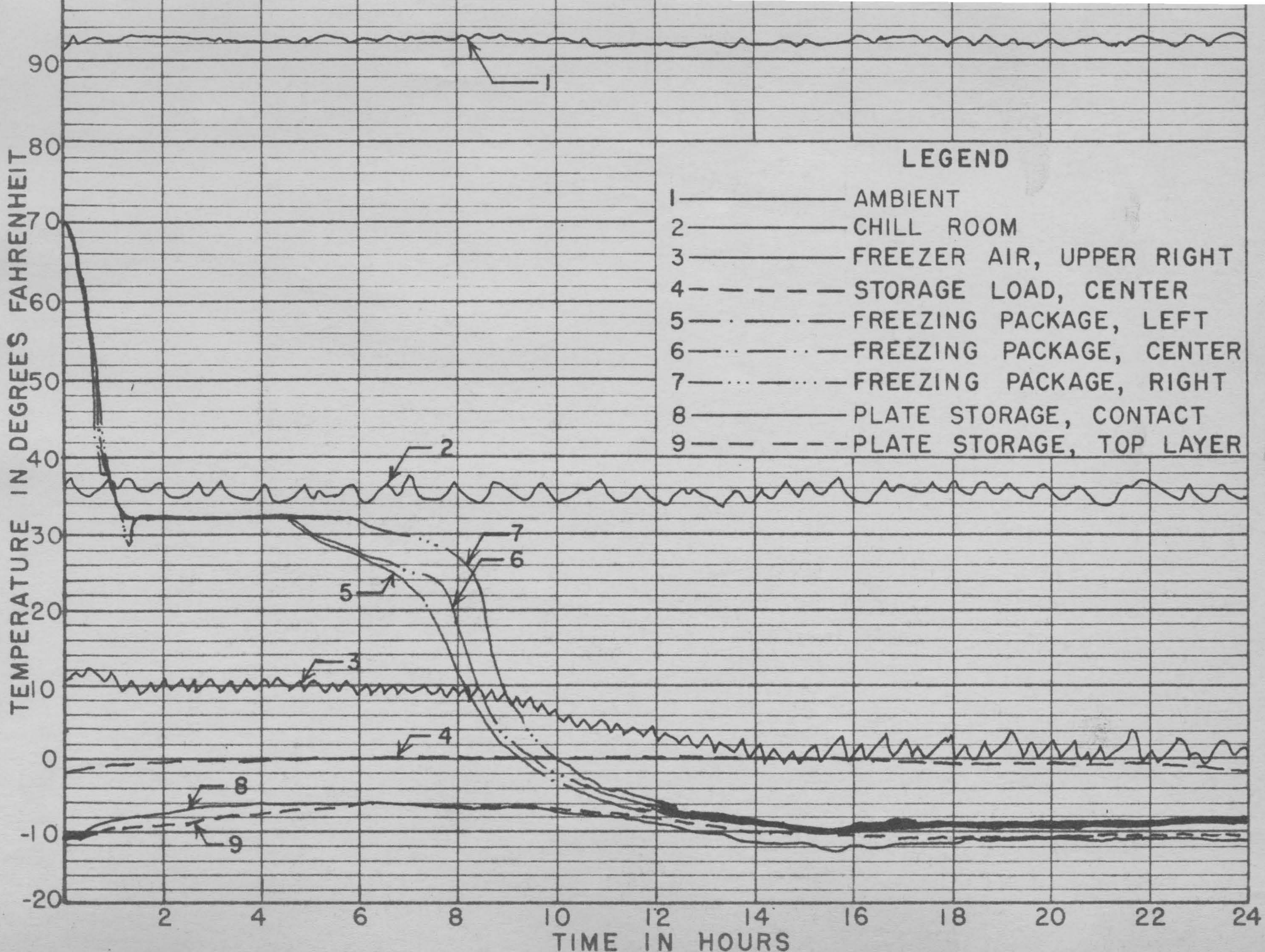


CHART 25 TEST 4 SERIES D-6 LOAD, FULL STORAGE (MAX. PLATE)

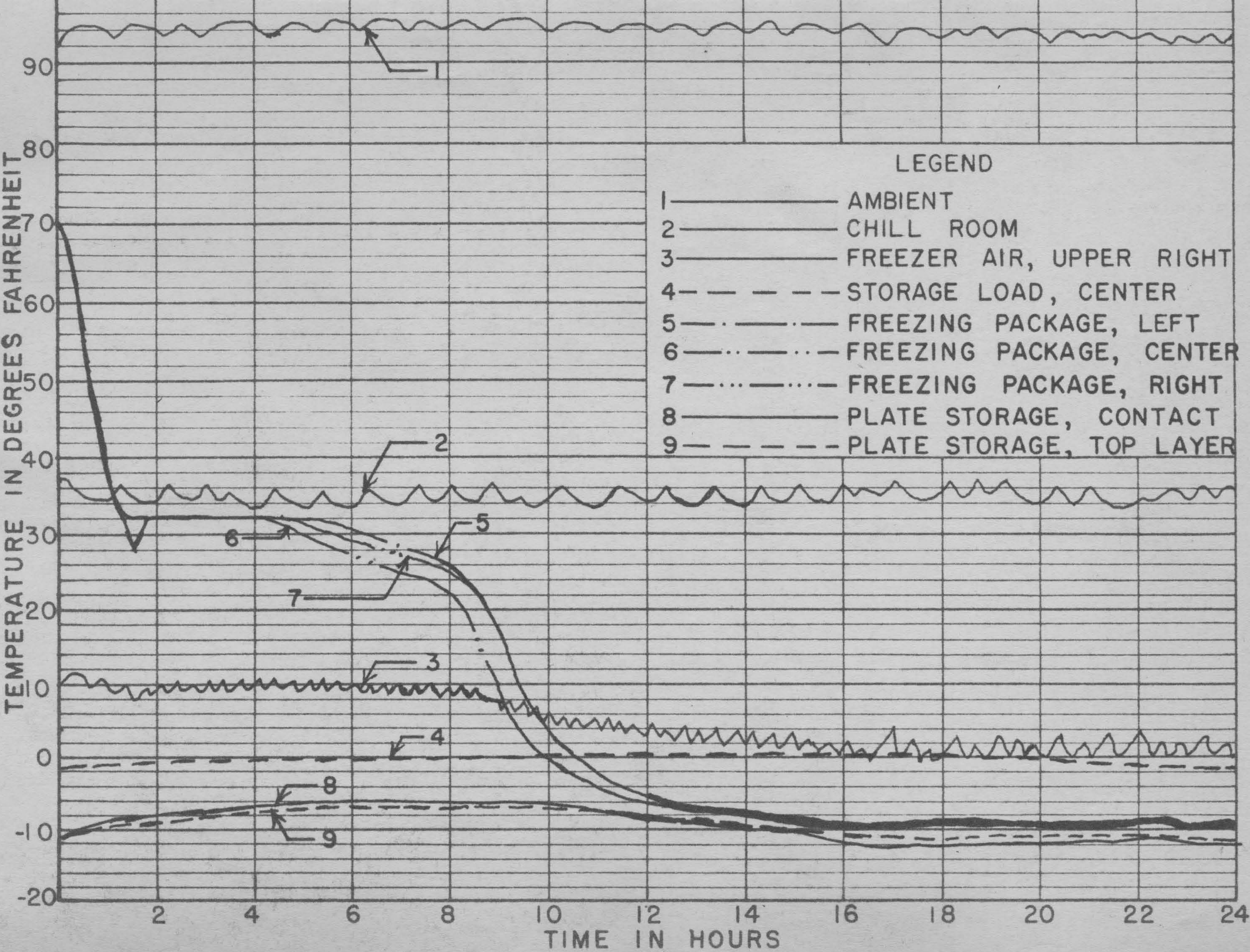


CHART 26 TEST 5 SERIES D-6 LOAD, FULL STORAGE(MAX PLATE)

APPENDIX B

HEAT LOAD CALCULATIONS

HEAT LOAD CALCULATIONS FOR ZERO STORAGE COMPARTMENT OF WALK-IN TYPE FARM REFRIGERATOR

Basic Conditions

1. Mean maximum monthly temperature 90 F
2. Average chill room temperature 35 F
3. Average freezer compartment temperature 0 F
4. Refrigerant temperature minus 10 F
5. Product Load

   a. Long-time storage (frozen food)

   2000 pounds per year (estimated)

   b. Processing load each day not to exceed the cooling of 30 pounds of vegetables from 60 F to 10 F in 12 hours[1] or the cooling of 60 pounds of beef from 40 F to 0 F in 18 hours, whichever is greater. Products frozen would have approximately the following average properties:

	Specific Heat above freezing	Freezing Temp.	Latent Heat	Specific Heat below freezing
Vegetables	0.92	30 F	128	0.47
Beef	0.70	28 F	100	0.38

---

1 There appears to be considerable controversy over the rate at which produce should be frozen. The author is not attempting to set this time at 12 hours, which was used in the design calculations, but merely agreeing with (11) who insists that the produce be frozen in 12 to 14 hours if only to insure that the produce in the freezer overnight can be removed to make way for fresh produce to be processed the following day.

6. Miscellaneous Load

a. Air Changes

For holding	10 per 24 hours
For processing	20 per 24 hours

b. Lights and men working (estimated)

For holding	20 Btu per hour
For processing	80 Btu per hour

7. Dimensions used for Freezer Compartment

a. Outside

Height	8 ft 3 in
Length	7 ft 11 1/2 in
Width	4 ft 1/2 in

b. Inside

Height	6 ft 3 in
Length	5 ft 6 in
Width	2 ft 3 in

c. Freezer door

Height	6 ft
Width	2 ft

d. Volume is approximately 75 cu ft

8. Surface areas and resistivities

	Area	"R"
Floor	32 sq ft	49.66
Ceiling	32 sq ft	48.93
Outside walls	131	49.90
Partition wall (less door)	35 sq ft	33.53
Door	12 sq ft	19.06

Conducted Heat Load for 90 F Ambient Temperature and 0 F Storage Temperature

$$H = \frac{at_d}{R}$$

Where a = area of surface in question

t_d = temperature difference

R = Resistivity

$$H(\text{Floor}) = \frac{32 \times 90 \times 24}{49.66} = 1385 \text{ Btu per 24 hours}$$

$$H(\text{Ceiling}) = \frac{32 \times 90 \times 24}{48.93} = 1410 \text{ Btu per 24 hours}$$

$$H(\text{Outside walls}) = \frac{131 \times 90 \times 24}{49.90} = 5670 \text{ Btu per 24 hours}$$

$$H(\text{Partition}) = \frac{35 \times 35 \times 24}{33.53} = 880 \text{ Btu per 24 hours}$$

$$H(\text{Door}) = \frac{12 \times 35 \times 24}{19.06} = 530 \text{ Btu per 24 hours}$$

Total conducted load--9875 Btu per 24 hours

Added Heat Load Due to 2 ft by 3 ft by 10 1/2 in Panel Substituted for Equal 12 in Insulated Wall Area

Resistivity "R" for Panel (insulated area)

Outside surface -	0.60
1/2 in plyboard and vapor seal -	1.33
9 1/2 in mineral wool insulation -	36.80
1/2 in plyboard (inside) -	0.70
Inside surface -	0.60
Total "R"	40.03

Resistivity "R" for Panel Frame (1 5/8 in by 9 1/2 in timbers)

Outside surface -	0.60

10 1/2 in plyboard and frame -	12.50
Inside surface -	0.60
Total "R"	13.70
Resistivity "R" for 12 in insulated wall -	49.90

Areas

12 in insulated wall	6.0 sq ft
10 1/2 in insulated panel	4.7 sq ft
10 1/2 in frame of panel	1.3 sq ft

Losses

12 in insulated wall	260 Btu per 24 hours
10 1/2 in insulated panel	252 Btu per 24 hours
10 1/2 in frame of panel	205 Btu per 24 hours
Total panel losses	475 Btu per 24 hours
Added load due to panel	197 Btu per 24 hours

<u>Infiltration Heat Load for Zero Storage Compartment for 35 F Air Entering</u>

Losses = Volume x air changes x Btu per cu ft

For holding = 75 x 10 x 1.2 = 900 Btu per 24 hours

For processing = 75 x 20 x 1.2 = 1800 Btu per 24 hours

<u>Miscellaneous Loads (Lights and Men Working)</u>

Holding = 480 Btu per 24 hours (estimated)

Processing = 1920 Btu per 24 hours (estimated)

<u>Product Load</u>

Vegetables (cool from 60 F to 10 F)	165 Btu per pound
Beef (cool from 40 F to 0 F)	119 Btu per pound

Hourly Load for Recommended Capacity

30 pounds vegetables = $\frac{30 \times 165}{12}$ = 410 Btu per hour

60 pounds beef = $\frac{60 \times 119}{18}$ = 396 Btu per hour

Total Load (holding)

Conducted	9875 Btu per 24 hours
Gain by Panel	197 Btu per 24 hours
Infiltration	900 Btu per 24 hours
Miscellaneous	480 Btu per 24 hours
Total	11452 Btu per 24 hours
or	477 Btu per hour

Total Maximum Load (Processing Vegetables)

Conducted	9875 Btu per 24 hours
Gain by Panel	197 Btu per 24 hours
Infiltration	1800 Btu per 24 hours
Miscellaneous	1920 Btu per 24 hours
Sub-total	13792 Btu per 24 hours
or	573 Btu per hour
Product	410 Btu per hour
Total (Processing)	982 Btu per hour

Total Design Load (Based on Continuous Operation With 10 Per cent Added As A Safety Factor)

Holding	- 477 + 48	= 525 Btu per hour
Product	- 410 + 41	= 451 Btu per hour
Processing	- 982 + 98	= 1080 Btu per hour
After freezing	- 573 + 57	= 629 Btu per hour

APPENDIX C

TABLES OF FREEZING RATES, OPERATING TIME, POWER CONCUMPTION, AND TEMPERATURE VARIATION OBTAINED IN TESTS ON THE ZERO STORAGE COMPARTMENT OF A WALK-IN TYPE REFRIGERATOR

TABLE I

FREEZING RATES SERIES D-2 NO STORAGE LOAD

Test No.	Thermocouple	Starting Temp. (° F)	Time to Reach (° F) Hours and Minutes			
			32°	30°	25°	10°
1	3	69	1-28	5-00	7-00	9-20
1	5	69	1-30	5-30	7-29	9-42
1	7	69	1-31	5-50	7-55	9-52
2	3	69	1-18	8-00	9-20	10-59
2	5	67	1-13	6-17	8-22	10-40
2	7	68	1-38	4-52	7-05	9-04
3	3	69	1-44	5-00	8-24	10-25
3	5	69	1-20	5-10	8-24	10-25
3	7	69	1-44	5-50	9-00	10-25
4	3	69	1-27	4-37	7-55	10-18
4	5	69	1-18	4-38	7-00	9-32
4	7	69	1-22	6-48	8-17	9-37
5	3	70	1-14	5-00	7-17	9-31
5	5	70	1-14	5-07	8-20	10-02
5	7	70	1-12	5-00	7-17	8-45
		Averages	1-25	5-31	7-56	9-55

TABLE II

FREEZING RATES SERIES D-3 277-POUND STORAGE LOAD

Test No.	Thermocouple	Starting Temp. (° F)	Time to Reach (° F) Hours and Minutes 32°	30°	25°	10°
1	3	70	1-45	6-19	8-10	8-59
1	5	70	1-45	7-00	8-35	9-25
1	7	70	1-45	5-50	7-45	7-40
2	3	70	1-05	5-13	7-10	8-14
2	5	69	1-05	6-02	7-55	9-06
2	7	69	0-50	4-50	7-05	8-30
3	3	69	1-05	5-57	7-10	9-20
3	5	69	1-05	4-17	6-45	8-24
3	7	69	1-05	4-52	6-45	8-35
4	3	69	1-05	5-45	7-22	8-08
4	5	69	1-00	5-10	7-15	8-52
4	7	69	1-14	5-10	7-15	8-35
5	3	68	1-14	6-30	8-28	9-14
5	5	70	1-28	5-00	7-15	8-15
5	7	70	1-08	5-34	7-48	8-50
		Averages	1-15	5-34	7-31	8-40

TABLE III

FREEZING RATES SERIES D-4 554-POUND STORAGE LOAD

Test No.	Thermocouple	Starting Temp. (° F)	Time to Reach (° F) Hours and Minutes 32°	30°	25°	10°
1	3	69	1-17	6-17	8-14	8-50
1	5	71	1-17	4-28	7-57	8-50
1	7	69	1-22	6-17	8-08	9-08
2	3	69	1-15	5-50	8-00	9-07
2	5	69	1-28	5-16	7-30	8-47
2	7	69	1-28	4-30	6-52	8-47
3	3	70	1-20	4-32	7-10	9-06
3	5	70	1-32	4-15	6-37	9-06
3	7	70	1-15	6-00	7-52	9-00
4	3	69	1-18	4-37	6-55	9-20
4	5	69	1-18	5-40	8-00	9-28
4	7	70	1-18	6-05	8-15	9-10
5	3	69	1-29	5-18	8-55	9-39
5	5	69	1-13	4-05	6-20	8-55
5	7	69	1-20	4-34	7-02	9-08
		Averages	1-21	5-11	7-35	9-05

TABLE IV

FREEZING RATES SERIES D-5 (554-POUND STORAGE LOAD WITH 174 POUNDS ON PLATES)

Test No.	Thermocouple	Starting Temp. (° F)	Time to Reach (° F) Hours and Minutes 32°	30°	25°	10°
1	3	70.5	1-13	6-38	8-20	9-05
1	5	70.5	1-08	5-45	7-41	8-37
1	7	70.5	1-08	5-28	8-20	9-17
2	3	69.0	1-14	4-00	6-05	9-04
2	5	68.0	1-14	5-50	8-05	9-22
2	7	69.0	1-14	4-45	7-30	9-18
3	3	69.5	1-05	4-16	6-30	9-30
3	5	69.5	1-05	4-45	6-52	8-43
3	7	69.5	1-05	6-07	8-20	9-18
4	3	70.0	1-40	4-32	7-25	9-35
4	5	70.0	1-07	5-00	6-40	8-02
4	7	70.0	1-20	5-32	8-15	9-18
5	3	70.0	1-09	5-40	8-15	9-21
5	5	70.0	1-09	6-40	6-55	8-51
5	7	70.0	1-09	4-40	8-20	9-06
		Averages	1-12	5-19	7-34	9-06

TABLE V

FREEZING RATES SERIES D-6 (554-POUND STORAGE LOAD WITH 290 POUNDS ON PLATES)

Test No.	Thermocouple	Starting Temp. (° F)	Time to Reach (° F) Hours and Minutes 32°	30°	25°	10°
1	3	69	1-04	6-30	8-15	8-45
1	5	69	1-10	4-10	6-40	8-38
1	7	69	1-10	5-30	7-48	8-52
2	3	70	1-09	5-00	7-13	8-25
2	5	70	1-09	5-16	6-55	7-48
2	7	70	1-09	5-52	8-00	9-00
3	3	70	1-10	6-02	8-03	9-00
3	5	70	1-10	5-45	7-50	8-43
3	7	70	1-10	6-02	7-50	8-55
4	3	70	1-09	4-55	6-43	8-08
4	5	70	1-09	5-00	7-05	8-21
4	7	70	1-09	5-40	8-16	8-52
5	3	70	1-18	6-15	8-07	9-24
5	5	70	1-18	4-52	7-20	9-00
5	7	70	1-18	5-32	8-00	9-24
		Averages	1-11	5-30	7-37	8-45

TABLE VI

GROUP #3 FREEZING RATES

Test No.	Thermocouples	Starting Temp. ($^\circ$ F)	Time to Reach ($^\circ$ F) Hours and Minutes			
			32°	30°	25°	10°
1A	1	59 ½	1 33	8-02	12-33	14-41
1A	3	59 ½	1 30	8-10	11-46	13-30
1A	9	59 ½	1 43	14-29	15-33	18-05
1A	11	59 ½	2 14	19-53	20-30	22-38
2A	1	62	1 53	10-33	14-00	16-09
2A	3	62	1 44	8-18	12-13	13-53
2A	9	62	2 17	15-41	16-46	19-40
2A	11	62	2 51	21-49	22-28	24-29
3A	1	60	2 06	12-25	16-10	18-18
3A	3	60	1 55	9-15	12-51	14-27
3A	9	60	1 17	5-26	8-19	10-30
3A	11	60	2 16	14-23	17-01	18-23
4A	1	60	2 07	11-57	16-25	18-30
4A	3	60	1 51	8-15	13-35	16-00
4A	9	60	0 59	4-27	6-35	8-35
4A	11	60	2 11	14-44	17-15	19-00

The above freezing rates were taken from page 78 of (7). Packages containing thermocouples 1 and 3 in tests 1 and 2 were flat packages. All packages in tests 3 and 4 were flat packages. A different type of evaporator plate was used by Cristel (7) in these load tests. Also, note the starting temperature of the water was 10 F lower than for the load tests of this experiment, otherwise, conditions are approximately the same.

TABLE VII

SUMMARY OF OPERATING TIME AND POWER CONSUMPTION

Test No.	Total Time	Operating Time	Per cent Running Time	Power Consumed Kwh	Cycles
(D-2) 1	1440	1309	90.7	6.74	12
2		1271	88.5	6.55	14
3		1408	97.7	7.49	6
4		1373	95.5	7.23	11
5		1350	93.8	7.20	9
Averages		1342.2		7.042	10.4
(D-3) 1	1440	1214	85.0	6.56	17
2		1182	82.2	6.31	16
3		1175	81.8	6.23	14
4		1190	82.7	6.38	12
5		1194	82.8	6.44	13
Averages		1185.0		6.384	14.4
(D-4) 1	1440	1171	81.4	6.20	13
2		1210	83.1	6.35	11
3		1222	84.9	6.45	12
4		1252	87.0	6.62	11
5		1202	83.7	6.38	12
Averages		1212.4		6.400	11.8
(D-5) 1	1440	1286	89.3	6.78	7
2		1259	87.4	6.62	9
3		1258	87.3	6.61	9
4		1279	88.7	6.69	8
5		1247	86.7	6.56	9
Averages		1265.8		6.652	8.4
(D-6) 1	1440	1302	90.7	6.67	7
2		1256	87.2	6.50	9
3		1191	82.8	6.40	10
4		1265	87.7	6.58	9
5		1304	90.7	6.68	8
Averages		1263.6		6.562	8.6

TABLE VIII

TEMPERATURE VARIATIONS IN DEGREES FAHRENHEIT OF THE CENTER AIR, SHELF STORAGE AND PLATE STORAGE OF THE ZERO STORAGE COMPARTMENT

Test No.	Start of Test			After 12 Hours			End of Test		
	Air	Shelf Storage	Plate Storage	Air	Shelf Storage	Plate Storage	Air	Shelf Storage	Plate Storage
(D-2) 1	12			3			-1		
2	13			2			-1		
3	16			9			3		
4	17			8			4		
5	18			6			4		
(D-3) 1	10	-4		2	-2		-2	-4	
2	12	-4		2	-2		2	-3	
3	11	-3		3	-1		0	-3	
4	9	-4		0	-1		-2	-4	
5	13	-4		1	-2		-1	-3	
(D-4) 1	12	-2		1	-2		1	-3	
2	12	-3		1	-3		-1	-3	
3	11	-3		2	-2		0	-2	
4	12	-2		2	-2		0	-3	
5	10	-3		1	-2		-1	-3	
(D-5) 1	12	-3	-10	2	-1	- 9	1	-3	-11
2	12	-4	-11	1	-2	-10	-1	-4	-12
3	12	-3	-12	3	-2	- 9	-1	-4	-12
4	12	-4	-11	3	-2	- 8	-1	-2	-12
5	12	-3	-10	3	-1	- 9	-1	-3	-11
(D-6) 1	10	-2	-11	4	0	- 8	0	-2	-12
2	12	-1	-10	3	0	- 9	1	-2	-12
3	11	-2	-10	3	1	-10	2	-1	-12
4	11	-2	-11	3	0	- 9	1	-2	-11
5	10	-2	-11	3	0	- 8	0	-2	-12

TABLE IX

SUMMARY OF FREEZING RATES IN FREEZING LOAD TESTS

SERIES Number	Cooling From 70 F to 32 F		Cooling From 70 F to 30 F		Cooling From 70 F to 25 F		Cooling From 70 F to 10 F	
	Average Time Hours & Minutes	S. D. Minutes	Average Time Hours & Minutes	S. D. Minutes	Average Time Hours & Minutes	S. D. Minutes	Average Time Hours & Minutes	S. D. Minutes
D-2	1-25	10.8	5-31	55.5	7-56	44.1	9-55	38.6
D-3	1-15	17.7	5-34	32.4	7-31	35.1	8-40	29.8
D-4	1-21	5.2	5-11	46.1	7-35	44.0	9-05	15.1
D-5	1-12	8.8	5-19	50.7	7-34	47.2	9-06	24.1
D-6	1-11	4.6	5-29	37.4	7-37	33.6	8-45	26.2
*Comparison	1-47	23.2	9-40	40.3	13-10	171.0	15-7	202.0

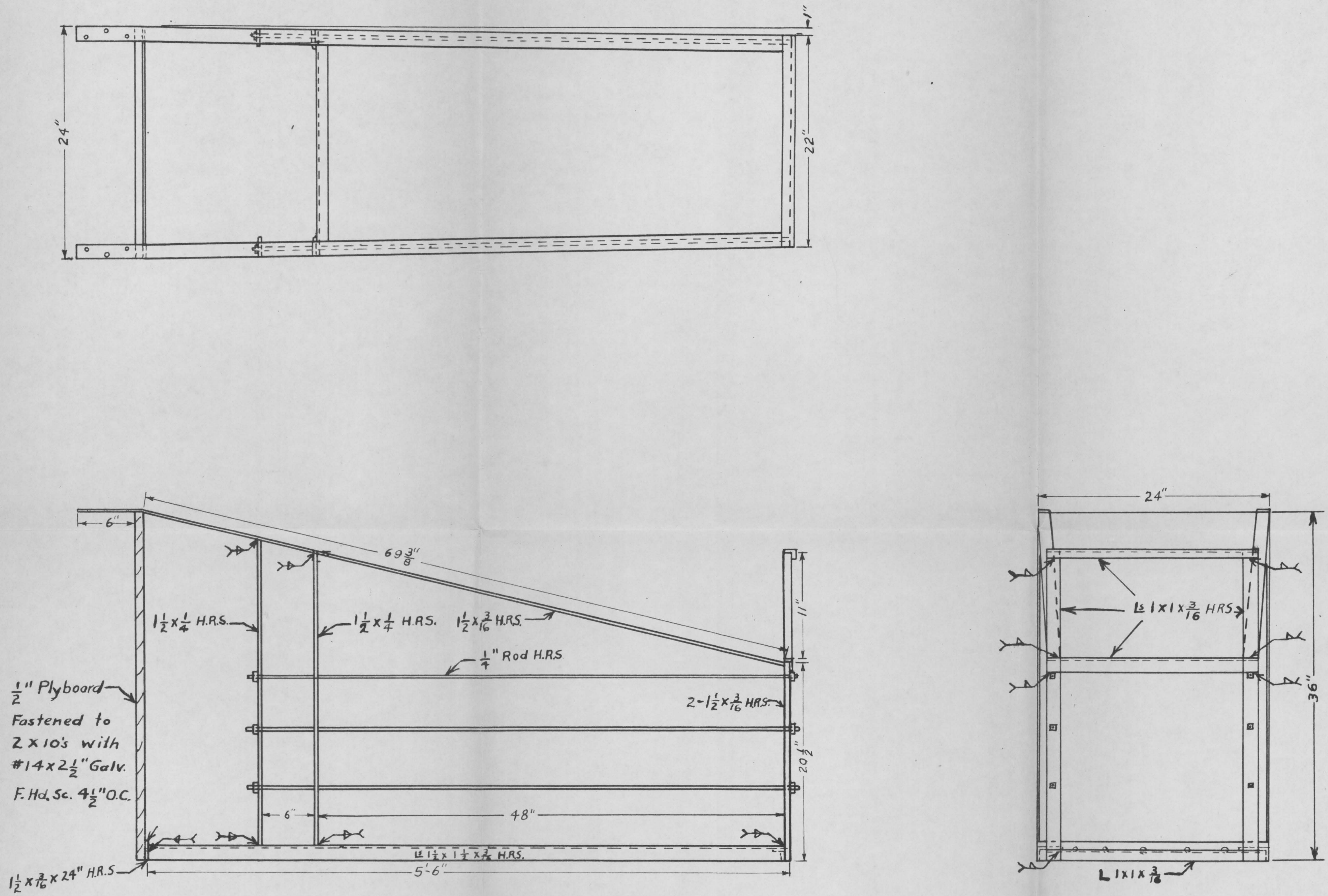
Calculations of Standard Deviations were made using the formula (8).

$$S.D. = \sqrt{\frac{\sum x^2 - \frac{S^2}{N}}{N-1}}$$

x = time of test in minutes

S = Sum of the times for the tests in the series.

* These values were taken from Table VI which gives the freezing rates for similar load tests conducted by Cristel (7).



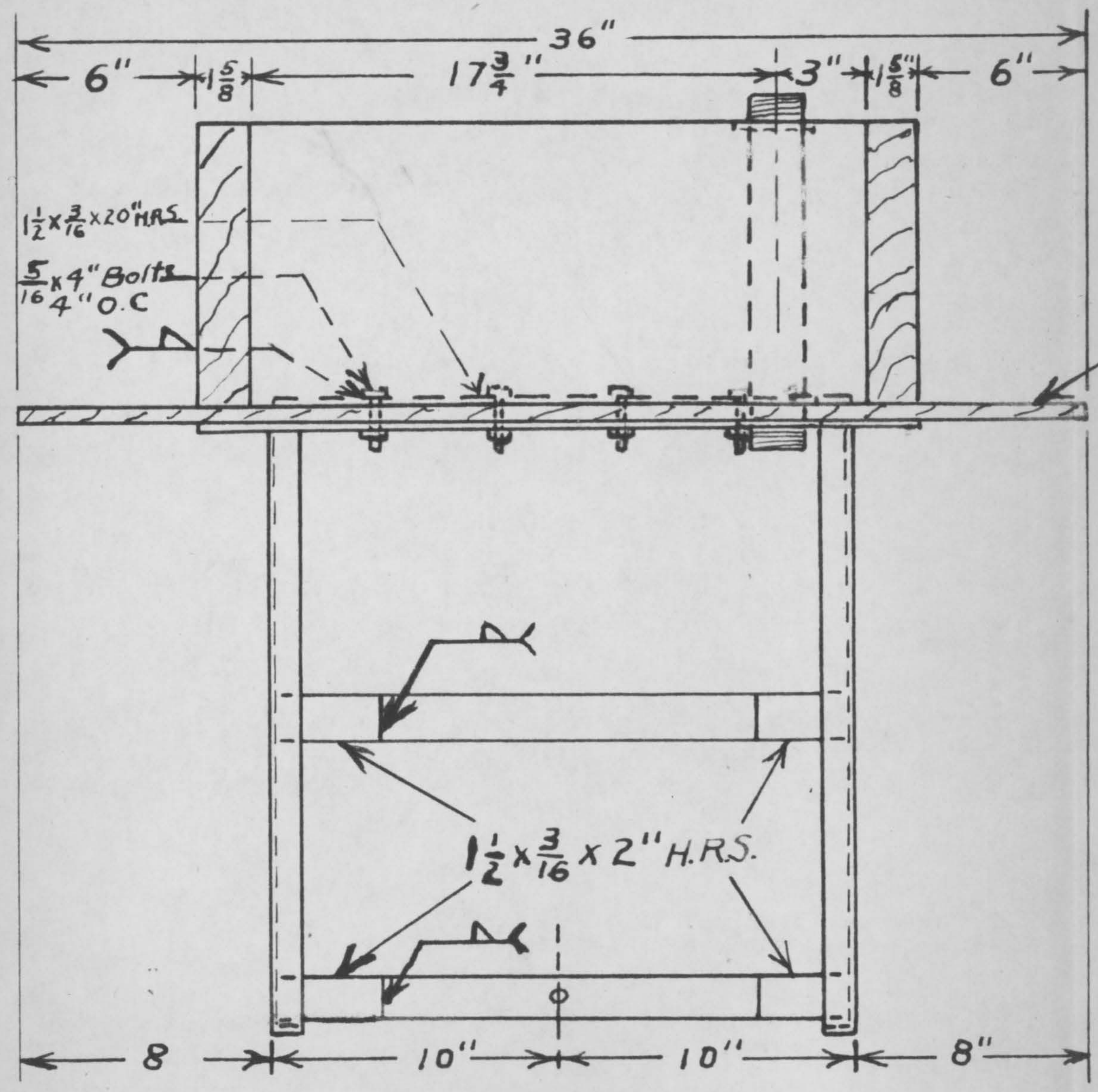
APPENDIX D

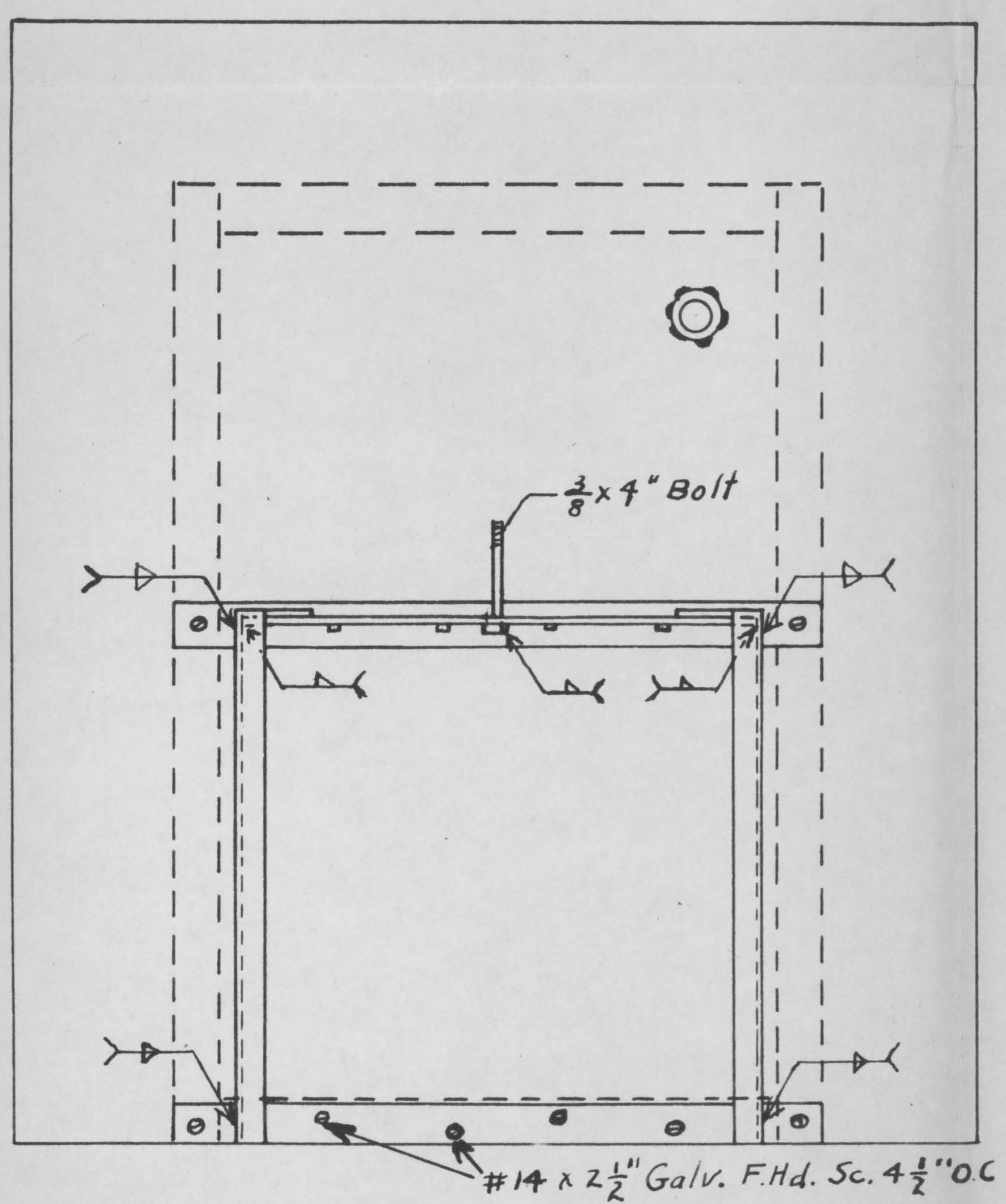
INTEGRAL REFRIGERATION UNIT

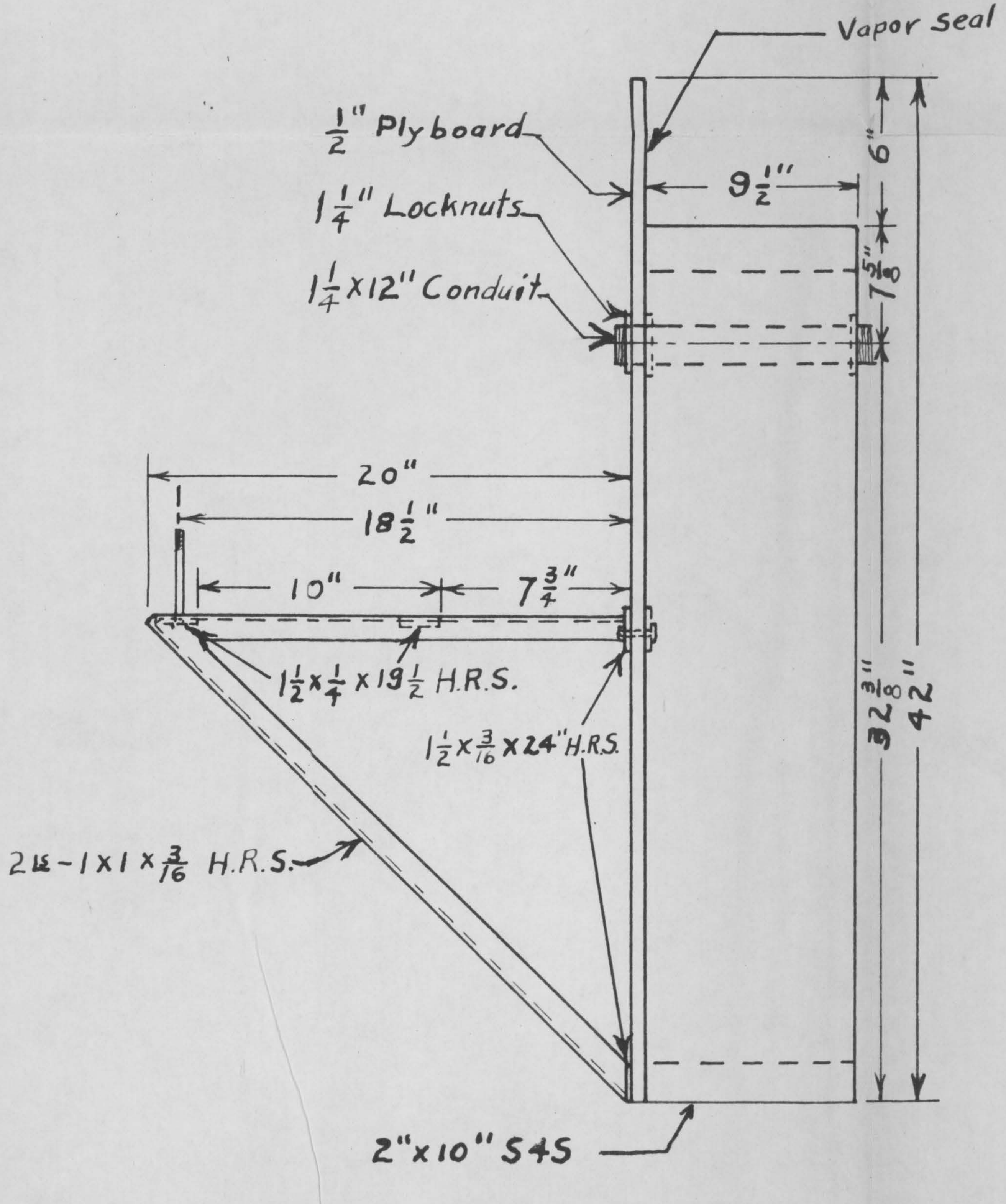
EVAPORATOR FRAME

Sheet 1 of 2 Scale: 1 1/2" = 1'0"

Drawn By: W.C. Wheeler







INTEGRAL REFRIGERATION UNIT

CONDENSER UNIT FRAME
AND
PANEL

Sheet 2 of 2 Scale: 2" = 1'0"

Drawn By: W.C. Wheeler