DEPARTMENT OF CHEMICAL AND METALLURGICAL ENGINEERING
Heat Transfer Laboratory

The University of Michigan
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Progress Report

on

THE PERFORMANCE OF HIGH-FIN TUBES IN COMBINED RADIATIVE AND NATURAL CONVECTION HEAT TRANSFER

Report No. 57

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Project 1592

WOLVERINE TUBE
Division of
CALUMET & HECLA, INCORPORATED

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NOTATION

A_m = Mean area in sq.ft./ft.

A = Area, sq. ft.

A_f = Area of fins per foot of length, sq.ft./ft.

A_i = Inside heat transfer area per foot of length, sq.ft./ft.

A_o = Outside heat transfer area per foot of length, sq.ft./ft.

A_r = Area of root per foot of length, sq.ft./ft.

D_i = Inside diameter of tube, inches

D_o = Outside diameter of bare tube or diameter over the fins of finned tubes, inches

D_r = Root diameter of tube, inches

F_{ab} = Radiation interchange shape factor from a to b, dimensionless

G_{max} = Mass flow rate at minimum cross section, lb./hr.-sq.ft.

H = Fin height, in.

h_i = Mean inside heat transfer coefficient, Btu/hr.-sq.ft.(internal area) °F

h_o' = Mean outside heat transfer coefficient, Btu/hr-sq.ft.(external area) °F

h_r = Radiative heat transfer coefficient Btu/hr.-sq.ft.(root area) °R

k = Thermal conductivity of gases, Btu/hr.-sq.ft.- °F/ft.

k_m = Thermal conductivity of fin material, Btu/hr.-sq.ft.- °F/ft.

ℓ = Fin height, inches

N = Number of fins per linear inch

Pr = Prandtl Number, $Cp\mu/k$

Q = Heat duty, Btu/hr.

Re = Reynolds Number, $(D_r/G_{max})/\mu$

r_f = Fin resistance, hr.- °F-sq.ft./Btu

r_m = Metal resistance, hr.- °F-sq.ft./Btu

S = Distance between adjacent fins, in.

t = Fin thickness, inch

t_w = Mean water temperature, °F

T = Absolute temperature, °R

U_o = Overall heat transfer coefficient based on outside area, Btu/hr.-sq.ft.- °F

V_t = Velocity of water in tube, ft./sec.

y = Fin thickness, ft.

ΔT_m = Logarithmic temperature difference, °F

μ = Viscosity at bulk stream temperature, lb./ft.-hr.

σ = Radiation constant, 1.712×10^{-9} Btu/hr.-sq.ft.- °R

ϵ_i = Emissivity of surface i, dimensionless

ABSTRACT

The summary of the status of the laboratory investigation currently in progress on the performance of high-fin tubes in combined radiative and convective heat transfer in direct fired heating applications is presented. Insufficient experimental data has been obtained to date on enough banks of finned tubes containing various sizes of fin heights, fin spacings, fin thickness, and root diameter with various tube pitch arrangements to make any predictions or draw any significant conclusions at this time.

OBJECTIVE

The purpose of this investigation is to obtain experimental heat transfer and pressure drop data on high-fin tubes in combined radiative and natural convection heat transfer using hot flue gas from a direct-fired heater. The experimental data is to be analyzed and correlated using regression analysis techniques on the IBM 7090 digital computer.

INTRODUCTION

Wolverine Tube has been providing a number of hot water heater manufacturers finned tubes for use in their commercial units. Table I presents the catalogue number of the tubes purchased from Wolverine Tube by eight hot water heater manufacturing companies.

TABLE I

Catalogue Number of Tubes
Purchased for Hot Water Heater Units

<u>Ace Tank & Heater</u>

61-0516065-01
61-0520065-01
61-0714072-01

<u>Amarillo Welding</u>

63-0710065-01
63-0712065-01

<u>Fleetwood Mfg. Company</u>

61-0710049-01
61-0710065-01

<u>Laars Engineering</u>

61-0614065-01
61-0710065-01
61-0714072-01

<u>Raypak Company</u>

61-0710049-01
61-0714065-01

<u>Rheem Manufacturing</u>

61-0516065-01

<u>Swimming Pool Supply</u>

61-0710065-01
61-0714072-01

<u>Weben Industries</u>

61-0516065-01

An examination of Table I indicates that hot water heater manufacturers purchasing tubes have been purchasing tubes having a wide range of dimensions for the same heat transfer application. In several instances, some of the manufacturers purchase the same tube. Table II indicates which manufacturers purchase which tube. Nine different tubes are purchased.

An examination of the catalogue dimensions corresponding to the tubes presented in Table II indicates that hot water heater manufacturers are purchasing finned tubes having 5 to 7 fins per inch, nominal inside diameters varying from 5/8 in. to 1 1/4 in., root wall thicknesses varying from 0.049 in. to 0.072 in., fin diameters varying from 1.4 in. to 2.2 in., fin heights from 0.35 in. to 0.40 in. All of the tubes have the same mean fin thickness. Consequently, the outside

TABLE II

Wolverine Finned Tubes Purchased by
Various Hot Water Heater Manufacturers

Tube Catalogue No.	Organization
61-0516065-01	Ace Tank & Heater Rheem Manufacturing Weben Industries
61-0520065-01	Ace Tank & Heater
61-0614065-01	Laars Engineering
61-0710049-01	Fleetwood Mfg. Company Raypak Company
61-0710065-01	Fleetwood Mfg. Company Laars Engineering Swimming Pool Supply
61-0714065-01	Raypak Company
61-0714072-01	Ace Tank & Heater Laars Engineering Swimming Pool Supply
63-0710065-01	Amarillo Welding
63-0712065-01	Amarillo Welding

areas and the inside areas in square feet per foot of tube length vary over a wide range, as do also the outside to inside surface area ratios. There appears to be no logical reason why the eight manufacturers involved should purchase finned tubes for the same application that have such a wide variation in dimensional characteristics.

The purpose of this investigation is to explore the effect of a wide range of dimensional characteristics on the performance of high fin tubes in this type of heat transfer application. The results obtained from this investigation should enable one to design an optimum tube for this application.

SELECTION OF EXPERIMENTAL HEAT EXCHANGER

Initially, it was felt that in order to obtain the quality of experimental heat transfer data needed for this investigation, it would be necessary to design a special natural gas fired or liquid petroleum gas (LPG) fired heat exchanger and have it fabricated under contract to some shop. Before proceeding with the development of such a design, catalogues were obtained from all of the significant hot water heater manufacturers. The package units available were carefully evaluated and it was found that several possibilities existed. Discussions with The University of Michigan shop personnel indicated that neither they nor any shop could fabricate a special unit for anywhere near the cost of a commercially built package unit. It was decided that, for economic reasons, a package unit should be used.

An evaluation of LPG vs. natural gas indicated that none of the LPG manufacturers control the composition of their fuel close enough for the needs of the research project. In fact, the variation in composition from load to load is quite significant. This would result in widely fluctuating heating values and associated problems. Because of the quantity of LPG that would be needed in storage and for hourly consumption, a vaporization unit would be required. For fire control reasons, the storage tank and vaporizer would have to be installed at a minimum of 75 feet from the Fluids Building. It turned out that The University would require that this unit be installed across the road in the woods behind the Fluids Building. For these reasons and because the pressure of the vaporized gas at the inlet to the flow regulators and meters adjacent to the experimental unit would fluctuate too widely, LPG was, therefore, discarded as the heating medium.

Discussions with representatives of the Michigan Consolidated Gas Company indicated that they control the heating value of natural gas delivered to the gas main at $\pm$ 2%. Also, the pressure on the gas main serving the North Campus buildings is maintained fairly constant. The pressure reducing regulation system installed by the gas company in the Fluids Building drops the gas main pressure to a carefully regulated pressure inside the building. A further drop in pressure would be required at our experimental unit. This could be satisfactorily controlled without any difficulty. In addition, the gas company offered to loan the project group a Rockwell Model No. 4 gas meter which has a capacity of 2,250 cu. ft. per hour at 1/2 inch water pressure and 5,000 cu. ft. per hour at 1 inch water pressure. The meter is worth approximately $2,800. All factors involved clearly indicated that natural gas should be used as the fuel.

An evaluation of the various types of gas burners was made. The uniformity of the burner gas temperature is a significant variable in the experimental investigation. One type of burner appeared to be far superior to all others being used in this respect. This burner is referred to as a "multi-blade" or "strip" burner. Two manufacturers use this type of burner, Weben Industries, Inc. and Raypak Company, Inc. Discussions with these two manufacturers indicated that only the Raypak Company could provide a large enough unit with enough flexibility for the investigagion. A Raypak Coppertherm Series II unit with modified instrumentation having a maximum input rating of 2,400,000 BTU/hour was purchased and installed on the ground floor of the Fluids Building located at the North Campus of The University of Michigan. Figure 1 shows the installed unit. Figure 2 presents the recovery rate curve for the unit as provided by the manufacturer.

The Fluids Building has a cooling tower water loop of sufficient capacity to handle the water flow rates required for the investigation. Sufficient natural gas capacity is available in the building to supply the burner requirements of the unit.

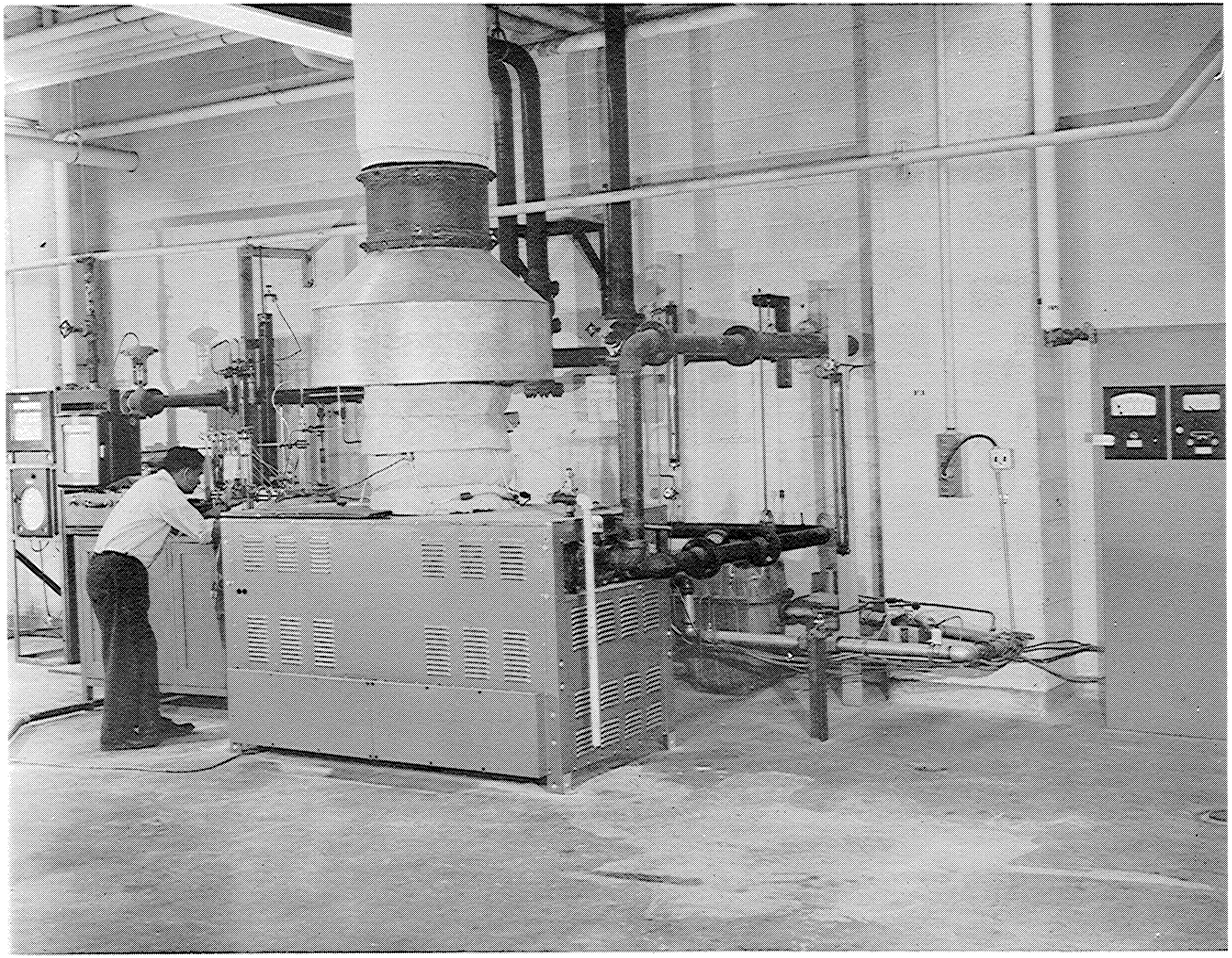

Fig. 1 Laboratory Experimental Installation

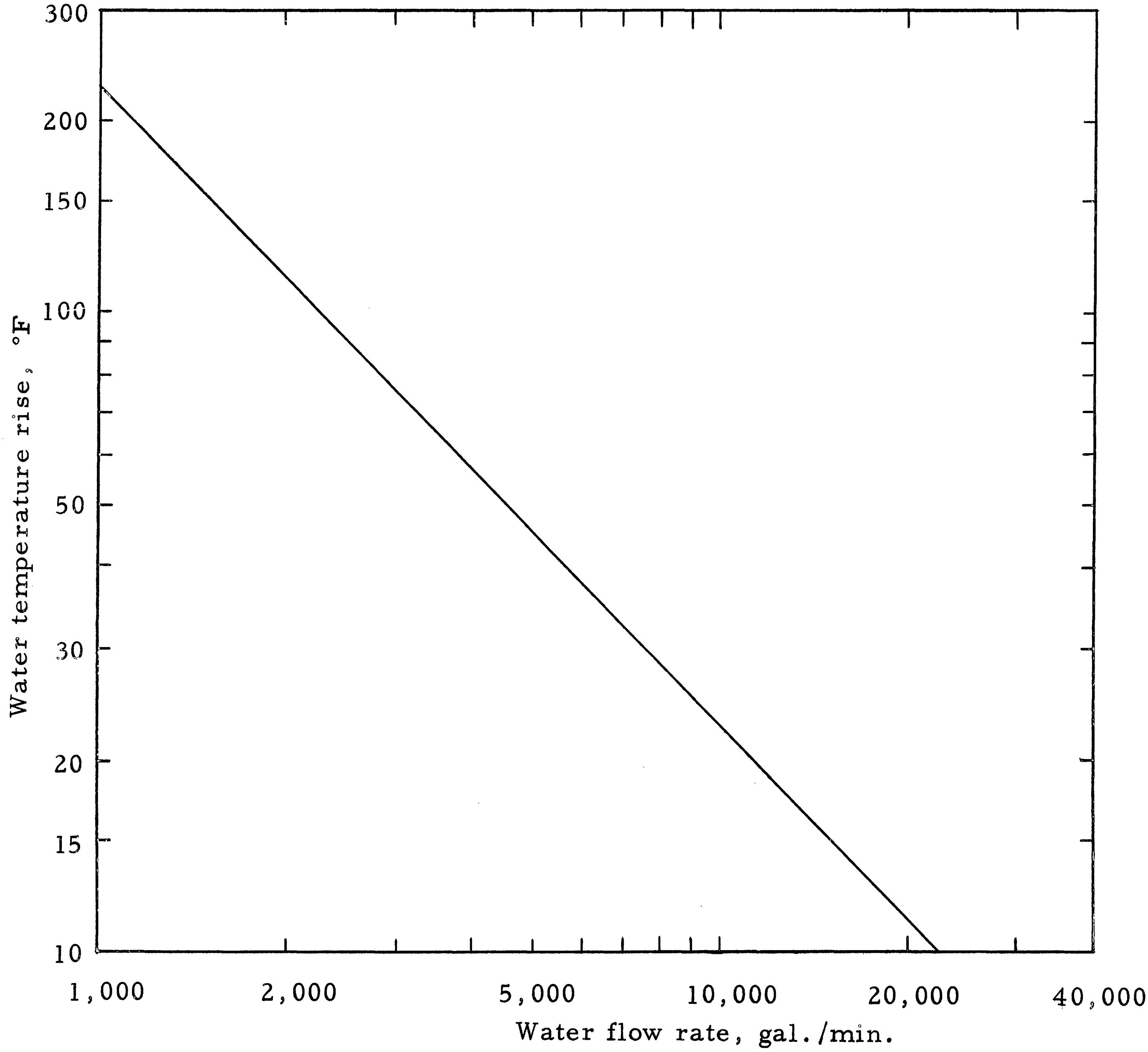

Fig. 2 Manufacturers Rating Curve of Recovery Rate. Input Rate is 2,400,000 Btu/hr.

SURVEY OF THE TECHNICAL LITERATURE

The American Gas Association has made many studies of the use of natural gas for hot water production. Most of these are concerned with domestic heaters with the hot water tank being part of the flue (1, 2, 3, 4) and with burner design for this type of service (5, 6, 7). Studies have also been made of gas fired space heaters (8), and application to simultaneous space heating and hot water production (9).

The American Gas Association has made a very extensive study of condensation from natural gas (10). In this study they investigated the dewpoint of the flue gas as a function of excess air and sulfide content in the fuel. It was found that 10 grains of hydrogen sulfide per 100 cu. ft. of fuel gas raised the dew point of a typical flue gas (7.2% CO_2) from 125 °F to 140 °F. It was also found that condensation improved the rate of heat transfer by approximately 12%.

A survey of the available technical literature on the subject of combined radiation and convective heat transfer to fin-tubes indicates that most of these papers are for fin tubes in radiant sections and convective sections in fired heaters used in oil refineries and gasoline plants. The most recent useful technical paper is that of Schweppe and Torrijos(11). This paper was published in June of 1964 and concerns the increase in heat recovery of refinery heating equipment by using fin-tubes in the convective section of the heater. The procedures presented are based on a master's thesis by Torijos (12) and on methods presented by Lobo and Evans (13) for combustion section radiation; Monrad (14) and Briggs and Young (15) for heat transfer within the convective section; Gardner (16) for efficiency of extended surfaces; and Gunter and Shaw (17) for pressure drop in crossflow.

In 1963, Wimpress (18) of the C. F. Braun and Company published a special report on rating fired heaters. This particular article presents a method which can be used to predict the performance of direct-fired heaters in typical refinery and petrochemical process heating applications. The method for rating the radiant sections is based on an earlier paper by Wilson, Lobo and Hottel (19).

Mieth presented a paper entitled "The Importance of Fin-Tube Application in Energy Conservation" (20) before the Petroleum Mechanical Engineering Conference of ASME in September of 1964. This article presents some test data for finned tubes of various types and bare tubes in waste-heat recovery applications.

The American Standards Association has approved a set of approval requirements for gas water heaters (21, 22, 23, 24). These standards have primarily to do with safety aspects of the design, construction and installation of such units. The American Gas Association's approval program for gas appliances is based upon American Standards, and the American Gas Association sponsors the American Standards Association project in this field.

The technical literature was reviewed to determine what flue gas temperature measurement procedures are acceptable. A symposium on flue gas temperature measurement was held at the American Society of Heating, Refrigeration and Air Conditioning Engineers Semi-Annual Meeting in New Orleans in January 1964. At this symposium, W. B. Kirk, Chief Research Engineer, American Gas Association Laboratories, Inc., presented a comparison of the procedures under the American Standard Association approval requirements for gas appliances (25). K. O. Schlentner, Director of Engineering, Crane Company, presented flue gas temperature measurement procedures followed by the Institute of Boiler and Radiator Manufacturers and the Steel Boiler Institute (26). R. E. Barrett and H. R. Hazard of the Battelle Memorial Institute presented a paper on the problems in flue-gas temperature measurement (27). They examined the codes of the Institute of Boiler and Radiator Manufacturers, American Standards Association-American Gas Association, American Society of Heating and Air Conditioning Engineers (predecessor to ASHRAE), Boiler Manufacturers Association, Steel Boiler Institute, Underwriters Laboratories, and Commercial Standards and found that recommended procedures differ significantly, so that different values of gas temperatures result from the use of these codes. They concluded that accuracy and reproducibility of flue-gas temperature measurements would be improved by codes incorporating the following provisions: (1) insulated flue pipes; (2) measurements made at a uniform location, close to the furnace; (3) smaller thermocouple wires and beads; and (4) multiple-thermocouple grids.

In general, there are two standard techniques, one known as the "5-point thermocouple method" and the other known as the "orifice method." The Institute of Boiler and Radiator Manufacturers and the Steel Boiler Institute Codes accept the American Gas Association 5-point thermocouple method of stack temperature measurement as an alternate to the orifice method. The 5-point thermocouple method results in good correlation with the orifice method. A careful study of the two methods indicated that the 5-point thermocouple method was well suited to our laboratory heater installation without major modification. It appeared quite impractical to use the orifice method. Consequently, the 5-point thermocouple method was adopted and the installation of the required thermocouples made.

In general, only the bottom horizontal row of finned tubes will be greatly affected by radiation when rectangular banks of finned tubes are used. The second horizontal row of finned tubes will be affected when triangular pitch banks of finned tubes are used. Insofar as convective heat transfer alone is concerned, generalized correlations only for triangular pitch banks of finned tubes are available. None exist for square pitch or rectangular pitch finned tube banks. The most useful generalized correlation for triangular pitch banks of finned tubes is that obtained by the Wolverine Tube Project at The University of Michigan. The initial laboratory work was carried out by Dennis J. Ward on his doctoral thesis under a Wolverine Tube Fellowship (28). This work was later extended by the Wolverine Tube Project group and a correlation, including the effect of fin thickness, fin height, and fin spacing, was published in the technical literature, by Briggs and Young (15). The latter reference will be of considerable value in this investigation.

Two books on the general subject of combustion contain some useful information. The books are "Combustion Engineering" by deLorenzi (29) and the "North American Combustion Handbook" (30).

A thorough search was made of the literature in an attempt to find the viscosity, thermal conductivity and heat capacity of the flue gas components for a temperature range of 500 °F to 2,000 °F. The major source was National Bureau of Standards (NBS) Circular 564 (31). This source is complete only for viscosity data. Heat capacity data was obtained for all components and checked against the NBS Circular from McBride's listing of thermodynamic properties (32). Thermal conductivity data was reported at the Fourth World Power Conference (33). This data does not agree with that of Keenan and Keyes (34) for water vapor. The Fourth World Power Conference data did agree with the NBS Circular data in the case of oxygen and carbon dioxide and were therefore used to extend the curves of oxygen and carbon dioxide. Thermal conductivity data is also reported for carbon dioxide by Geringer (35).

LABORATORY APPARATUS

The experimental equipment is basically the Raypak boiler "Coppertherm", Series II Model 2400. This unit has an input rating of 2,400,000 Btu/hr and an output rating of 1,920,000 Btu/hr. As supplied, the water flows through four passes in a bank of 24 tubes (No. 61-0714065-01). Table III lists the physical dimensions and properties of these tubes. The tubes are placed in groups of 12 tubes, two rows deep. Vertical spacing varies along the length of the tubes, horizontal spacing is with touching fins. The center area is blocked off with a fabricated steel strip.

TABLE III

Details of Finned Tubes Supplied with the Water Heater

K_m, thermal conductivity of fin material	= 196 Btu/hr.sq.ft./ °F/ft.
D_o, outside fin diameter, inches	= 1.7914
D_r, root diameter, inches	= 1.000
D_i, inside diameter, inches	= 0.875
S, fin spacing, inches	= 0.1218
Fins/inch	= 7
t, fin thickness, inches	= 0.021
H, fin height	= 0.3957
A_o, outside area, sq.ft./lin.ft.	= 2.316
A_i, inside area, sq.ft./lin.ft.	= 0.229
A_o/A_i	= 10.1
D_o/D_i	= 1.7914
A_f, area of fin, sq.ft./ft.	= 2.093
A_r, area of root, sq.ft./ft.	= 0.223
A_r/A_f	= 0.1065
D_o/D_r	= 1.223
y, fin thickness, ft.	= 0.00175

A line diagram of the experimental layout is shown in Figure 3. Gas is fed to six burners; each is equipped with an automatic modulation valve, which are set to maximum to prevent interference with data collection. The six burners are fed from two manifolds controlled by electrical cut-off valves. These valves are connected such that if (a) any of the four pilot lights are off, (b) if there is no water

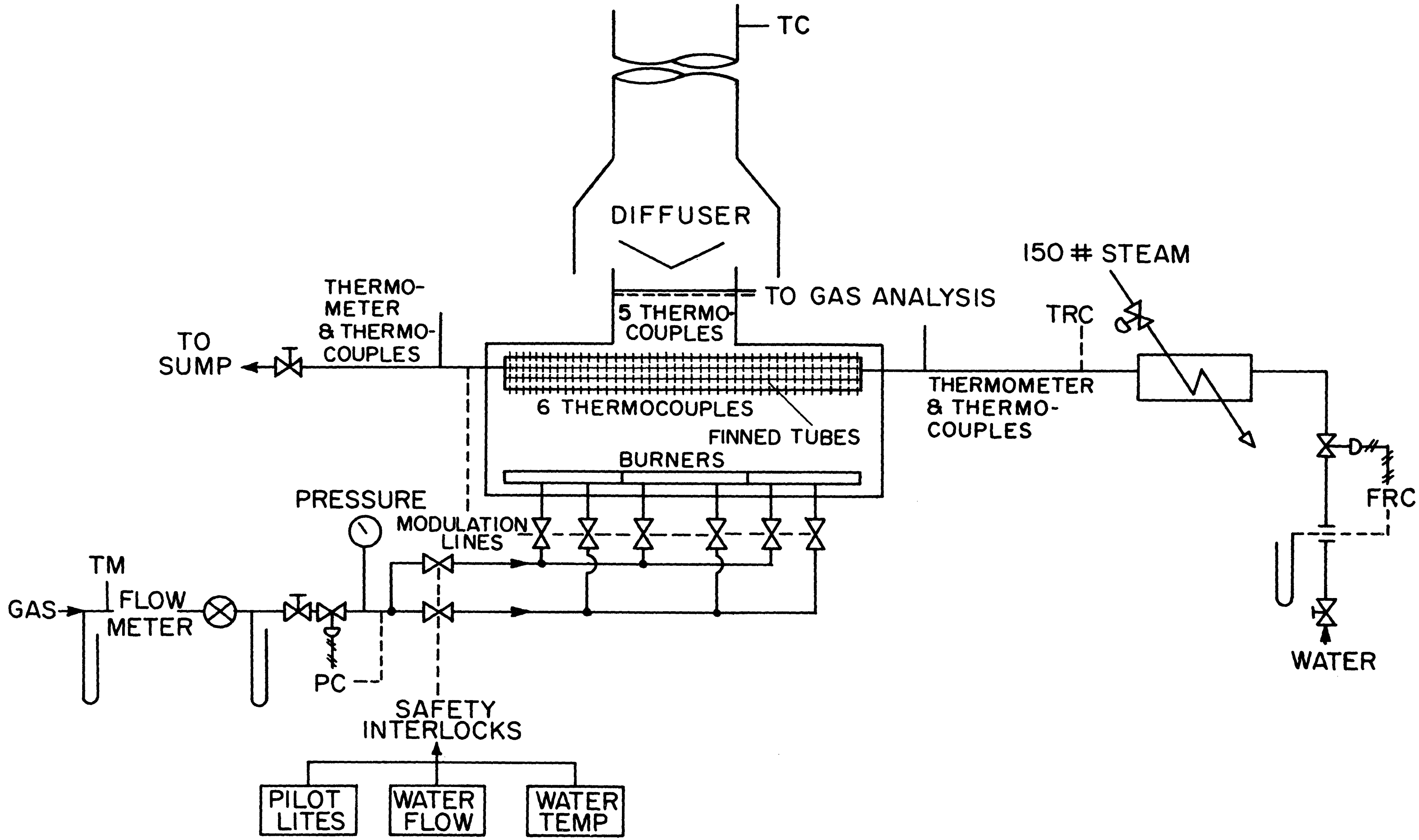

Fig. 3 Line Diagram of Experimental Apparatus

flow, or (c) if the main burner does not produce flame within eight seconds of being turned on, they will automatically close. A further interlock provides that if the temperature of the exit water goes above a specified limit (210 °F), the gas supply will be automatically cut off.

The two gas headers are supplied from the gas main after a meter, a pressure regulator (set at eight inches of water), a master cut-off valve (manual), and a throttling valve. The throttling valve is controlled by a Metrol constant pressure controller so as to maintain constant pressure in the headers. This pressure can be varied from 0.86 to 6 inches of water. A Metrol meter is connected to monitor the header pressure in the range 0 to 4 inches of water.

Water is supplied from a cooling tower provided at the Fluids Building. Valves are provided to completely isolate the apparatus from the cooling tower and provision is made to drain the system of water, which is done after every experimental run. The inlet water flows through an orifice and then a valve controlled by a Minneapolis-Honeywell-Brown recorder controller activated by the pressure drop across the orifice. The controller will control any flow from 40 to 300 gpm while the reading shown on the recorder is correct from 100 to 300 gpm. The orifice has been calibrated with a weigh tank from 5.45 to 200 gpm using a 1.750 specific gravity oil up to 50 inches oil (2.98 in Hg) and a mercury manometer from 2.5 to 6.0 inches of mercury (maximum flow sustainable with pumps available at that time). Since the calibration curve is straight from 0.2 in. of mercury up with a slope of 0.5, it has been extrapolated for the rest of the flow range.

A steam heat exchanger is used to pre-heat the water. The steam flow is controlled by a valve activated by a Minneapolis-Honeywell-Brown recorder controller such that the temperature leaving the pre-heater is constant. Calibrated thermometers placed in water filled wells are used to measure the water temperature both into and out of the experimental apparatus from 46 °F to 116 °F; outside this range copper-constantan thermocouples are used.

The temperatures below the tube bundle are measured with six chromel-alumel thermocouples shielded from radiation from below. The temperature above the bundle is measured with five chromel-alumel thermocouples placed so that linear averaging of temperature will give the correct stack temperature. Samples of the stack gas are aspirated through a perforated length of one-half inch iron pipe spanning the stack at the centerline of the stack.

Figure 4 shows the apparatus from the left side. The controller at the top of the left rack is the water inlet temperature controller recorder. Directly below this is the water inlet flow controller recorder. The recorder in the rack to the right of the one referred to above continuously records the temperature in the stack above the diffuser. Behind the controllers on the floor is the steam heated exchanger used to pre-heat the water from the cooling tower.

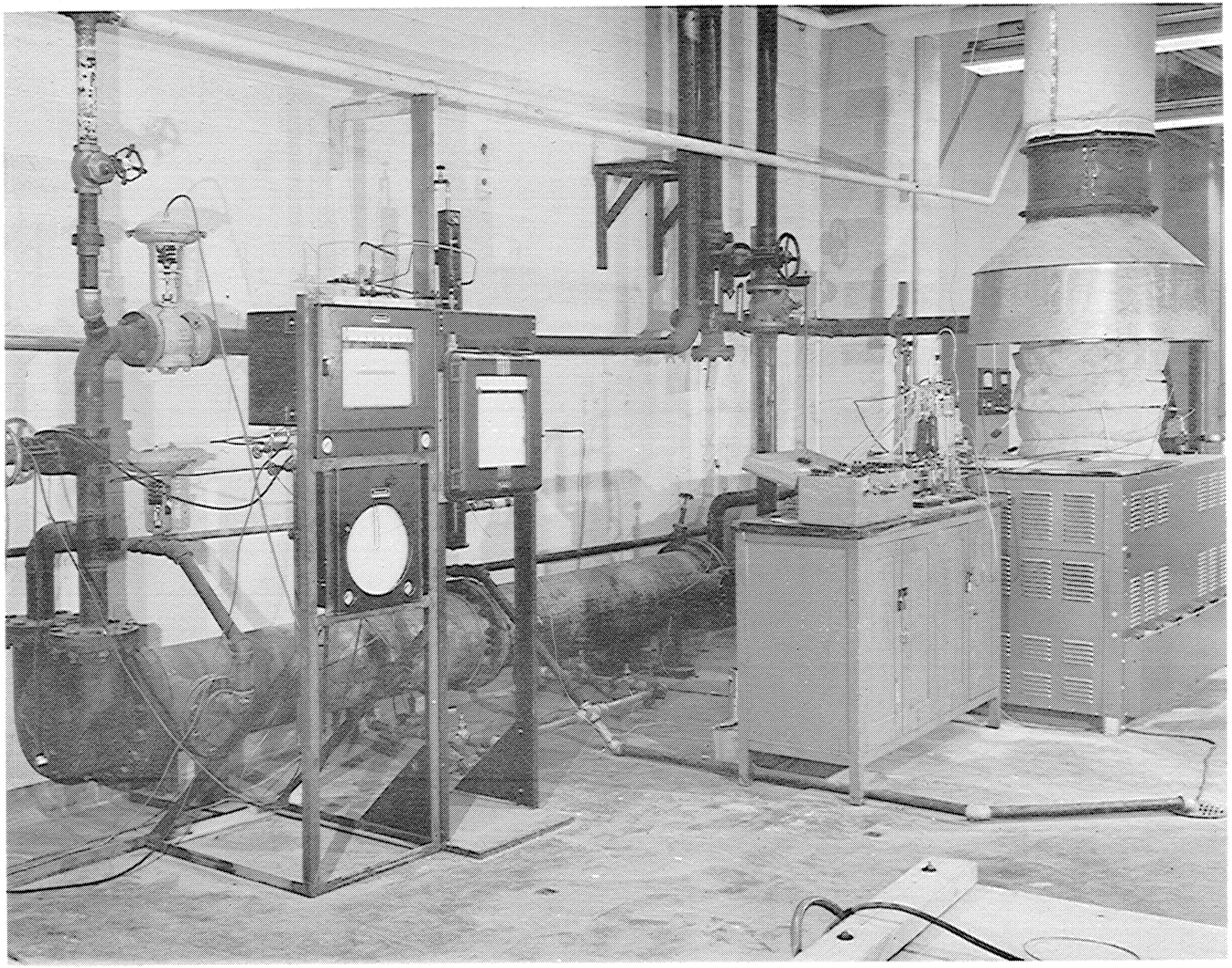

Fig. 4 Laboratory Experimental Installation

On the cabinet top at the left end are the potentiometer and selector switches used to measure the thermocouple temperatures. To the right is the Orsat gas analysis equipment used for analyzing the flue gas. The Rockwell Model 4 gas meter sets on the floor behind the gas fired heat exchanger and can be seen in the right of Figure 1. The Metrol gas flow instrumentation panel can be seen at the extreme right of Figure 1.

OPERATION OF EQUIPMENT

Before starting up the equipment, it was necessary to turn on the cooling water pump. Once this was turned on, the drain valves on the equipment were closed and the main water valves were opened. The water flow and water temperature regulators were set to the desired settings and then turned on. Then the stack temperature recorder was turned on. The steam valve for the water pre-heater was opened and the condensate bled from the steam pipe before the bleed valve was closed.

Before firing the burner, the three permanent pilot lights were checked and lit if not on. Then the power to the safety system was turned on and 30 seconds allowed for the tube to warm up. When the tube warmed up, the relay would drop in and the indicating light would come on. At this point the gas was turned on. After the main burners came on, the gas differential regulator was set to the desired setting and set on automatic control. Once the burners were on, the stack cover was removed. (It cannot be removed earlier or the stack starts a heavy down draft which is very difficult to reverse.)

If the desired water flow rate was below 45,000 lb./hr. (160 gpm), the manifold valves for the manometer using 1.750 SG oil were opened. This manometer is necessary to give sufficient accuracy below 2.6 inches of mercury across the orifice.

Sufficient time must be allowed for the system to reach equilibrium. The usual period was nearly one hour. During this time, the information on Data Sheet No. 1, shown in Table IV, was filled in. It should be noted that this information was assumed constant for the entire run (about four hours). Space is also provided so that these sheets can be partially prepared in advance to serve as a record of future runs, e.g., desired water and gas flow-rates.

It should be noted that although there is a water temperature regulator, there is also need of manual intervention. The water temperature is regulated by control of the steam fed to a pre-heater. This pre-heater can raise the water temperature about 15 °F at a flow rate of 150 gpm of water. The normal temperature desired is between 60 °F and 75 °F. Without the cooling tower turned on, the heater will raise the sump water temperature about 10 °F/hr with maximum gas flow. With the cooling tower on in the winter, the equilibrium temperature of the sump water may go down to 45 °F. Thus, to maintain an inlet temperature of, say 70 °F, one must turn the cooling tower fan on and off at alternate hours, relying on the regulator to eliminate the sinusoidal variation of inlet temperature.

The data for the experimental run is collected on Data Sheet No. 1, Table IV, and Data Sheet No. 2, Table V. Because of the amount of data to be collected, two people are necessary, one to collect the gas samples and analyze them through the Orsat, the other to collect the other readings. This normally requires 30 to 45 minutes per run. A new set of data is started every hour and three sets of data are usually required to prove steady state conditions. Tables IV through VII present a complete sample set of data for experimental run No. 7 on December 10, 1964. Note that although there is a column under stack analysis for infra-red analysis, it is not presently being used.

The shut-down procedure is essentially the reverse of the start-up. The gas regulator is set to manual and the flame is reduced to minimum, then the power to the safety interlocks is shut off and the gas main closed. The regulators are turned to zero and shut off, then the water valves are closed. The system drain valves are opened and left open. Then the steam valve is closed, the water pump shut off and the stack cover replaced.

During the run, samples of the stack and feed gas are taken for the mass spectrograph. A present, samples of the stack gas are taken twice during a set of runs and one sample of the feed gas is taken for the set of runs.

Data Sheet No. 2, Table V, provides a place for recording the amount of condensation evident to the operators during the experimental run.

TABLE IV

Hot Gas to Fin Tube Heat Transfer

Data Sheet No. 1

Run No. 7

Date 12/10/64 Time 12:00 noon ~~AM/PM~~

Operators C. W. Bash and M. V. Kulkarni

Wet Bulb Temperature 59.8 °F

Dry Bulb Temperature 76.6 °F

Humidity 0.007 lb. water/lb. dry air

Barometric Pressure 29.46 in. Hg at 74 °F

Temperature Correction -0.122

Calibration Correction -0.006

Corrected Pressure 29.332 in. Hg at 32 °F

Water Thermometer No.: in 89164

Water Thermometer No.: out 55079

Water Orifice No. Used #3

Desired water flow 100 gpm = ---- in. Hg

Desired gas flow 30 cfm

Regulator Settings

Water flow rate 100 gpm

Water inlet temperature 72 °F

Gas flow differential 2.0 in. water

TABLE V

Data Sheet No. 2

Run No. 7 A

Date 12/10/64

Time 2:30 ~~AM~~/PM

Flow Conditions:

Water T from tower	70	°F	Gas T_{in}	28.8	°C
T into heater	68.6	°F	P_{in}	14.5	in water
T from heater	97.5	°F	$P_{regulated}$	8.0	in water
ΔP across orifice	2.2	in Hg	ΔP (meter)	3.4	in water
	39.3	oil			

100 ft^3 in 3 min 13.55 sec

31.0 cfm

Condensation none light heavy

Stack Temperature 190 °F

Thermocouples

	EMF	T	$(T_i - T_{i-1})$		EMF	T	$(T_i - T_{i-1})$
1	35.75	1580	--	7	5.79	287	--
2	31.05	1374	--	8	8.85	424	--
3	31.40	1389	--	9	6.87	335	--
4	31.7	1402	--	10	6.61	324	--
5	31.7	1402	--	11	5.34	267	--
6	31.55	1396	--	water in			
				water out			

water temp rise = 28.9 °F

Orsat Analysis:

	IR	Level	ΔL
Full		99.30	--
CO_2		93.24	6.06
O_2		83.29	9.95
CO	13.5	83.30	

Mass Spec. Sample 7-A

H/C = 4.02

Excess Air = 82 %

TABLE VI

Data Sheet No. 3

Run No. 7B

Date 12/10/64

Time 3:15 ~~AM~~/PM

Flow Conditions:

Water T from tower	65	°F	Gas T_{in}	29.1	°C
T into heater	67.8	°F	P_{in}	14.5	in water
T from heater	96.9	°F	$P_{regulated}$	8	in water
ΔP across orifice	2.3	in Hg	ΔP (meter)	3.35	in water
	39.6	in oil			

100 ft^3 in 3 min 13.3 sec

31.0 cfm

Condensation None <u>light</u> heavy

Stack Temperature 200 °F

<u>Thermocouples</u>

	EMF	T	$(T_i - T_{i-1})$		EMF	T	$(T_i - T_{i-1})$
1	36.0	1591	+11	7	5.86	290	+3
2	31.3	1385	+11	8	8.9	426	+2
3	31.6	1398	+ 9	9	6.91	337	+2
4	31.9	1411	+ 9	10	6.75	330	+6
5	31.05	1374	-28	11	5.33	266	-1
6	31.8	1407	+11	water in			--
				water out			--

water temp rise = 29.1 °F

Orsat Analysis:

	IR	Level	ΔL
Full		99.17	--
CO_2		93.12	6.05
O_2		83.43	9.70
CO	24.5	83.41	0.02

H/C = 4.26

Excess Air = 77.5 %

TABLE VII

Data Sheet No. 4

Run No. 7 C

Date 12/10/64

Time 4:15 ~~AM~~/PM

Flow Conditions:

Water T from tower	62.2	°F	Gas T_{in}	29.3	°C
T into heater	67.5	°F	P_{in}	14.5	in water
T from heater	96.3	°F	$P_{regulated}$	8.0	in water
ΔP across orifice	2.4	in Hg	ΔP (meter)	3.4	in water
	40.2	in oil	100 ft^3 in	3 min 13.5 sec	
				31.0	cfm

Condensation none light heavy

Stack Temperature 190 °F

Thermocouples

	EMF	T	$(T_i - T_{i-1})$		EMF	T	$(T_i - T_{i-1})$
1	35.75	1580	-11	7	5.80	287	-3
2	31.10	1366	-19	8	8.86	424	-2
3	31.45	1392	- 6	9	6.87	335	-2
4	31.70	1402	- 9	10	6.67	326	-4
5	30.90	1368	- 6	11	5.26	263	-3
6	31.45	1392	-19				

water in --

water out --

thermocouple temp rise

thermometer temp rise 28.8

Mass Spec. Samples 7 C Stack

7 Feed

Stack Gas Analysis:

	IR	Level	ΔL
Full	--	99.9	
CO_2	--	93.9	6.00
O_2	--	83.8	10.10
CO	--	83.7	0.10

H/C = 4.04

Excess Air = 82.5 %

STACK GAS AND FUEL GAS ANALYSIS

Samples of the stack gas were collected at both ends of a perforated section of 1/2 in. pipe spanning the stack. The two lines were connected to an aspirator to continually purge the lines. Provision was made for collecting samples of stack gas for Orsat analysis and mass spectrometer analysis.

Although a very old and often studied problem, analysis of stack gases is still not an exact science whenever the gas contains large amounts of water vapor. Since both analyses take place below the dew point of the flue gas, it is necessary to remove most of the water vapor before analysis is made. This introduces problems because of the solubility of carbon dioxide and carbon monoxide in the water removed.

The first method of analysis attempted was the use of the basic Orsat apparatus for analysis of carbon dioxide, oxygen and carbon monoxide. The residue is assumed to be nitrogen. Knowing these four quantities and the composition of the feed gas, the ratio of volume of air to volume of natural gas at the burner can be calculated. Very poor reproducibility was obtained and it was impossible to get results that would correlate with the known feed gas composition as supplied by Michigan Consolidated Gas Company branch in Ann Arbor.

It was next decided to use a mass spectrometer to determine the stack gas composition. At the same time, samples of the fuel gas would also be analyzed. Figure 5 shows the mass spectrometer apparatus used in making these analyses. The mass spectrometer takes a small sample of gas and breaks the molecules into smaller charged ions. These ions are passed through an electromagnetic field that curves the path of the ion in a manner proportional to the mass-to-charge ratio. At the end of the path a current measuring device counts the number of ions with a specific mass-to-charge ratio. By varying the electrical field, one can continuously scan a range of molecular weights. The amount of current measured is automatically plotted by a galvanometer. A typical plot of a fuel gas analysis is presented in Figure 6.

It is known that the way a molecule breaks up is a characteristic of the molecule and independent of other species present. Also, the height of a given peak is directly proportional to the partial pressure of the compound in the sample. Since no two compounds break up in the same way, a set of simultaneous equations for the contribution of each component to the height of each peak can be solved for the partial pressure of each component present. This calculation is done on the IBM 7090 computer using a program written in the Michigan Algorithmic Decoder (MAD) language.

At present, there is not sufficiently good agreement between the mass spectrometer results and the gas company method of analysis. The feed gas

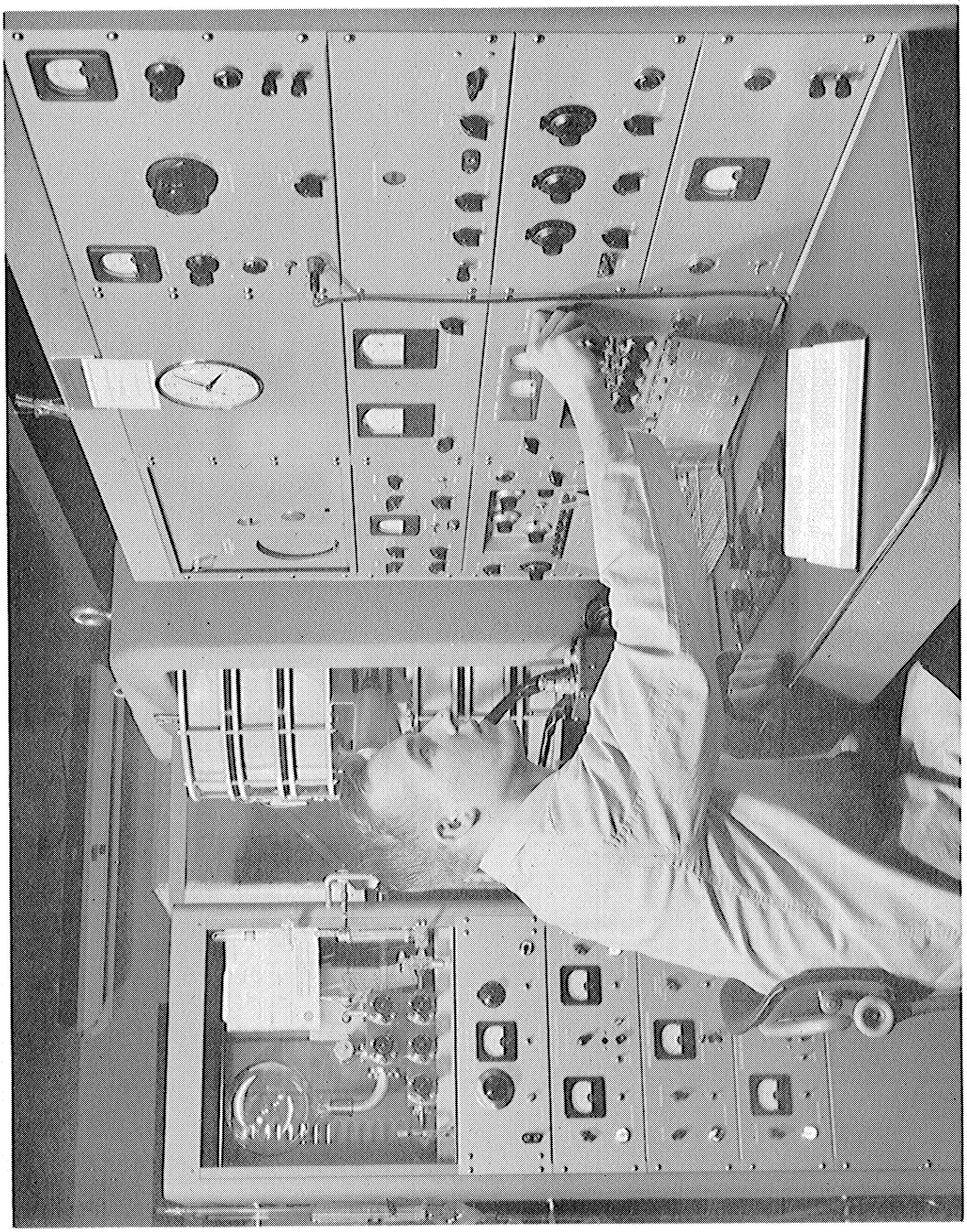

Fig. 5 Mass Spectrometer Apparatus

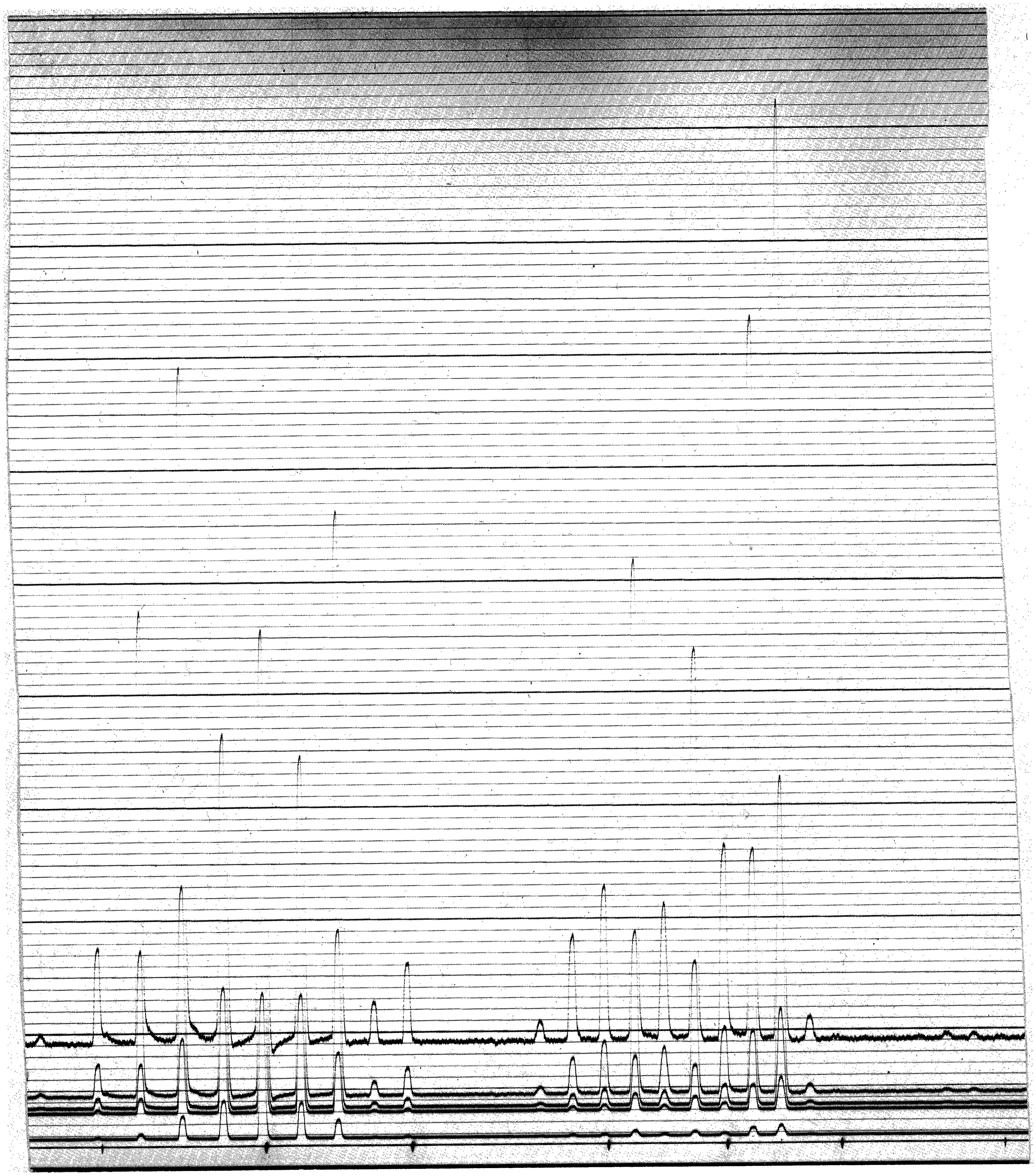

Fig. 6 Typical Mass Spectrometer Plot

composition according to the mass spectrometer is always lower in methane and higher in propane than that indicated by the gas company. Table VIII presents a comparison of the mass spectrometer results with the composition provided by the gas company.

TABLE VIII

Comparison of Fuel Gas Compositions

Component	Gas Co.	Mass Spectrometer Run Number		
		14060	14102	14101
Methane	93.50	87.18	84.71	86.41
Ethane	3.85	6.92	8.78	7.75
Propane	.77	1.34	1.80	1.80
i-Butane	.08	.17	.16	.12
n-Butane	.01	0	.05	.04
Nitrogen	.36	4.38	4.24	3.71
Carbon Dioxide	1.31	0	.26	.36
Others	.12	0	0	0
Gross Heating Value	1033	1099.4	1069.1	1058.4
Net Heating Value	941	993.3	966.4	956.8
Mole of Carbon/Mole of Fuel	1.0518	1.0572	1.0851	1.0774
Mole of Hydrogen/Mole of Fuel	4.0416	4.0266	4.0802	4.0660
H/C, atomic ratio	3.843	3.809	3.760	3.774

It has also been found that the mass spectrometer results do not agree exactly with the Orsat analysis of the flue gas, but the discrepancy is within reasonable error. A comparison of two typical analyses is shown in Table IX.

An explanation of how the mass spectrometer results are calculated is presented in the Appendix.

TABLE IX

Comparison of Flue Gas Analyses

	Run 7 C		Run 9 B	
	Orsat	Mass Spec.	Orsat	Mass Spec.
Carbon Dioxide	6.0	6.08	7.89	7.34
Oxygen	10.1	9.79	6.90	7.25
Carbon Monoxide	0	0	0	0
Nitrogen & Argon	83.7	83.08	84.80	85.37
H/C	4.04	4.097	3.92	4.27
% Excess Air	82.5	79.59	44.3	47.96

ANALYSIS OF DATA

Heat Balance

The performance of the water heater is evaluated on the basis of a heat balance. The calculation involves the following:

1) The total heat input, calculated from the fuel gas input flow rate and its heating value.

2) The total heat transfer to water, calculated from the total flow rate and rise in temperature of the water. Care was taken to calibrate the water orifice and the thermometers to minimize the source of experimental error.

3) The convective heat transfer. (The total heat transfer is the sum of radiative heat transfer and convective heat transfer.) The convective heat transfer is calculated from the temperature drop of stack gas across the water tube bundle as measured by thermocouples and the gas flow rate computed from the stack gas analysis and fuel gas rate.

4) Radiative heat transfer, the difference between the total heat transfer and convective heat transfer is taken as being due to radiation.

5) Total heat loss and efficiency of the water heater is then calculated by usual procedures.

Radiation

The radiative heat transfer for a typical analysis amounts to about 35 to 45 percent. A survey of the recent literature published on the design and performance of fired heaters (11, 18) indicates that the radiation heat transfer should be less than this amount.

The basis for radiant heat transfer is the Stefan-Boltzman equation. A black body at absolute temperature T radiates energy at a rate of q_b by the following relationship:

$$q_b = \sigma T^4 \tag{1}$$

and for radiant heat transfer between two real surfaces at a temperature T_a and T_b, the relation becomes:

$$q_r = A \cdot F_{ab} (\epsilon_a T_a^4 - \epsilon_b T_b^4) \qquad (2)$$

where

σ	=	1.712×10^{-9} Btu/hr.ft^2 °R
A	=	Area of surface a, sq. ft.
F_{ab}	=	Interchange shape factor from a to b
ϵ_a, ϵ_b	=	Emissivities of surfaces a and b
T_a, T_b	=	Absolute temperatures of surfaces a and b, °R

The luminous radiant heat transfer rate calculated from the above equation using different shape factors for our particular water heater is given in Figure 7.

A small amount of heat transfer also takes place due to non-luminous radiation from the gas in the convection zone following the method presented by Hottel (36). For finned tubes the gas radiation coefficient is based on root diameter of the tube. The area used is the area of bare tube. A tube surface emissivity of 0.9 and a constant partial pressure of CO_2 and H_2O and a tube-to-tube distance of 1.5 to 2 diameters were assumed. The apparent heat transfer coefficient for non-luminous radiation from the flue gas to the tube is given by Equation 3.

$$h_r = \frac{(\sigma) \dfrac{(1 + e_t)}{2} (e_g T_g^4 - e_t T_b^4)}{(T_g - T_t)} \qquad (3)$$

where the average gas temperature T_g is defined as the average inlet gas temperature plus the log mean temperature difference from gas to inlet water, the average tube wall temperature T_t is approximately the average inlet water temperature plus 50 °F and:

e_t = emissivity of tube = 0.9

e_g = emissivity of the flue gas at the temperature in question at a partial pressure of (water + CO_2) of .19 atm (18)

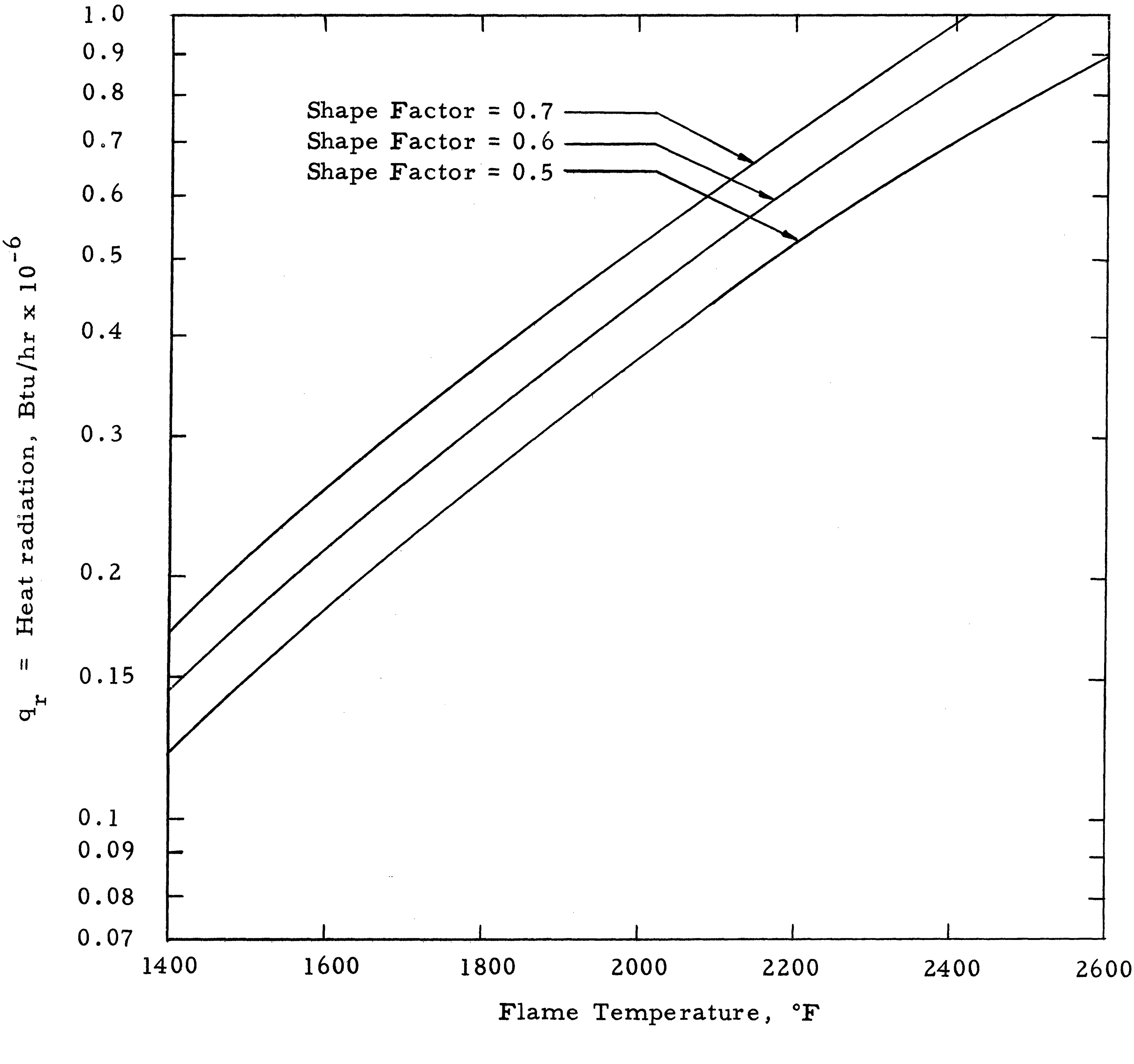

Fig. 7 Rate of Luminous Radiation Heat Transfer

The solution of Equation 3 for 80% excess air is presented in Figure 8. This figure indicates that non-luminous gas radiation should not be neglected.

Example Analysis

The summary of a typical analysis (data collected for Experimental Run No. 7) is presented in Table X. A review of the results presented in Table X indicates that the radiative heat transfer is excessively high compared to theoretical predictions. This raises the question as to whether or not the thermocouple temperature measurements of the hot flue gasses are correct. Subsequently, a theoretical analysis of the radiation losses from the thermocouples to the cold tubes was made. This analysis showed that the indicated temperatures below the bundle were low by approximately 200 °F. On the basis of these theoretical calculations, the heat balances were revised by raising the measured hot flue gas temperatures as measured by thermocouples located below the tube bundle by 200 °F.

TABLE X

Analysis of Data Collected for Experimental Run No. 7

	Run 7		
	A	B	C
Fuel gas flow rate S. cu.ft./mi.	30.24	30.20	30.06
Water flow rate, 10^4 lbs./hr.	4.85	4.85	4.8
Stack gas flow rate, lbs./hr.	2631	2658	2620
Water temperature rise, °F	28.9	29.1	28.8
Stack gas temperatures (Ave.), °F			
Below the water tube bundle	1424	1428	1420
Above the water tube bundle	327.4	329.8	327
Total heat input, 10^{-6} x Btu/hr.	1.71	1.702	1.698
Total heat transfer to water, 10^{-6} x Btu/hr.	1.401	1.410	1.382
Convective heat transfer, 10^{-6} x Btu/hr.	0.842	0.824	0.802
Radiative heat transfer (by difference) 10^{-6} x Btu/hr.	0.559	0.586	0.581
Total heat lost, 10^{-6} x Btu/hr.	0.309	0.29	0.316
Sensible heat lost, 10^{-6} x Btu/hr.	0.168	0.173	0.168
Radiation heat lost, 10^{-6} x Btu/hr.	0.141	0.117	0.148
Efficiency, per cent	82.1	83.1	81.51
Heat Transfer by Radiation, %	39.7	41.8	41.8
Excess air, %	82	77.5	82.5

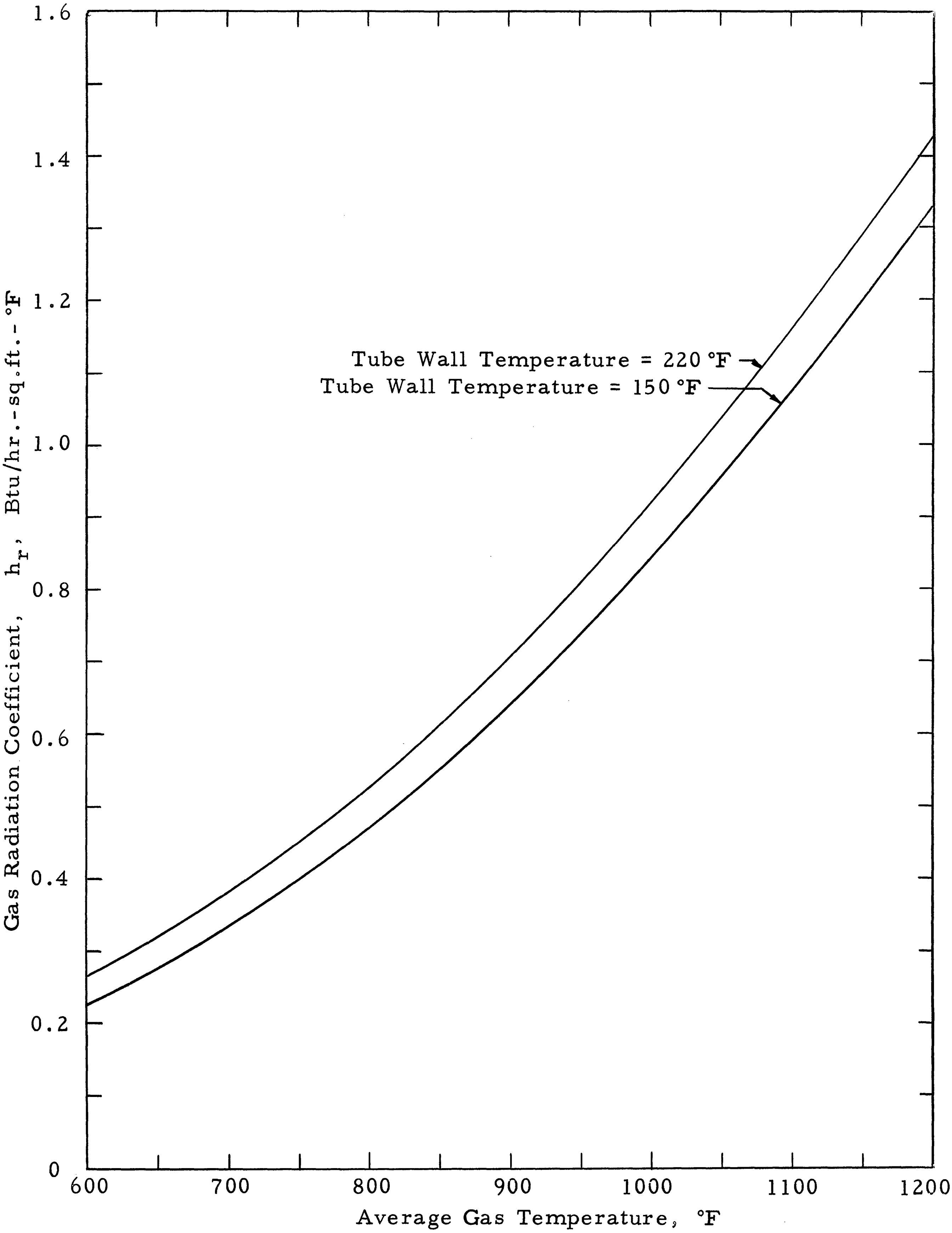

Fig. 8 Non-luminous Gas Radiation Heat Transfer Coefficient

The revised heat balance on the basis of the estimated hot gas temperatures for Run No. 7 are presented in Table XI.

TABLE XI

Revised Heat Balance for Run No. 7

	A	B	C
Total heat input, 10^{-6} x Btu/hr.	1.71	1.702	1.698
Total heat transfer, 10^{-6} x Btu/hr.	1.401	1.410	1.382
Convection heat transfer, 10^{-6} x Btu/hr., Q_c	0.976	0.983	0.972
Radiation heat transfer, 10^{-6} x Btu/hr., Q_r	0.425	0.427	0.410
Per cent heat transfer by convection	69.6	69.7	70.3
Per cent heat transfer by radiation	30.4	30.3	29.7

A comparison of the results given in Table XI with Table X indicates that the radiative heat transfer has been reduced by approximately one-third.

The overall heat transfer coefficient is given by Equation 4.

$$Q_c = U_o A_o \Delta t_m \qquad (4)$$

or re-arranging:

$$U_o = \frac{Q_c}{A_o \Delta t_m} \qquad (5)$$

where:

Q_c = convective heat transfer, Btu/hr

U_o = overall heat transfer coefficient, Btu/hr.-ft.^{2}- °F

A_o = heat transfer area of fin tube, sq.ft.

Δt_m = log mean temperature difference, °F

A summary of overall heat transfer coefficients for the revised data taken in Run No. 7 are as follows:

	A	B	C
Q_c, Btu/hr.	0.976×10^6	0.983×10^6	0.972×10^6
A_o, sq.ft.	144.75	144.75	144.75
Δt_m, °F	714	717	715
U_o, Btu/hr-sq ft-°F	9.45	9.48	9.40

The individual gas film heat transfer coefficient can be calculated after determining the individual resistances that make up the total resistance, $1/U_o$.

Water Film Coefficient

The water film coefficient is given by (36):

$$h_i = 150\,(1 + .011\, t_w)\,\frac{(V_t)^{0.8}}{(D_i)^{0.2}} \qquad (6)$$

where:

h_i = water side film coefficient, Btu/hr-sq ft-°F

t_w = mean water temperature, °F

V_t = velocity of water inside the tube, ft/sec

D_i = inside diameter of tube, inches

A summary of h_i for Run No. 7 is as follows:

	A	B	C
mean water temperature, t_w, °F	83.05	82.35	81.9
water flow rate, cu ft/sec	0.216	0.216	0.214
area of flow, sq ft	0.0125	0.0125	0.0125
V_t, ft/sec	17.3	17.3	17.1
D_i, inches	.875	.875	.875
h_i, Btu/hr-sq ft-°F	2,890	2,880	2,840

Gas Film Coefficient

The gas film coefficient is determined from Equation 7.

$$\frac{1}{U_o} = \frac{1}{h_o'} + r_f + r_w\left(\frac{A_o}{A_m}\right) + \frac{1}{h_i}\left(\frac{A_o}{A_i}\right) \tag{7}$$

where

A_m = mean area in sq. ft./ft.

U_o = overall heat transfer coefficient, Btu/hr-sq ft-°F

h_o' = actual gas film coefficient, Btu/hr-sq ft-°F

r_f = resistance of fin

r_w = resistance of root wall

h_i = water side film coefficient (Equation 6)

$\frac{A_o}{A_i}$ = ratio of outside surface to inside surface

The fin resistance is given by Equation 8.

$$r_f = \left(\frac{1}{h_o'}\right)\left[\frac{m^2/3\ \sqrt{D_o/D_r}}{1 + \frac{A_r}{A_f}\left(1 + \frac{m^2}{3}\sqrt{\frac{D_o}{D_r}}\right)}\right] \tag{8}$$

where

$$m = H\sqrt{\frac{2}{(k_m)\ (y)\ \ \frac{1}{h_o'} + r_o'}} \tag{9}$$

Assuming different h_o' values, r_f, the fin resistance can be calculated for the particular fin. The fin resistance calculates out to be a constant value of 0.00025 for h_o' of 1.0 to 100.

The metal wall resistance is given by:

$$r_w \left(\frac{A_o}{A_m} \right) = \frac{X \cdot A_o}{k \cdot A_m}$$

where

k_m = thermal conductivity = 196 Btu/hr-sq ft-°F

A_o = 2.316 sq ft/ft

D_r = 1.0 inch

D_i = .875 inch

X = thickness of wall = .063 inch

$$A_m = \pi \left(\frac{D_r - D_i}{\ell n \, (D_r / D_i)} \right) (\ell) = \pi \left(\frac{1 - .875}{\ell n \, \frac{1}{.875}} \right) = .2455 \text{ sq ft/ft}$$

$$\left(\frac{A_o}{A_m} \right) r_w = \left(\frac{.063}{196 \times 12} \right) \left(\frac{2.316}{.2455} \right) = .0002525$$

The calculated values of h_o' for the revised data for Run 7 using Equation 7 are presented in Table XII.

TABLE XII

Calculated Coefficients and Resistances

	A	B	C
U_o	9.45	9.48	9.40
r_f	0.00255	0.00255	0.00255
$r_w\left(\frac{A_o}{A_m}\right)$	0.00025	0.00025	0.00025
$h_i\left(\frac{A_i}{A_o}\right)$	286	285	281
h_o'	10.0	10.15	10.0

DISCUSSION OF RESULTS

Considerable difficulties were encountered in obtaining reliable physical property data for the components of the flue gas in the temperature range of 500 °F to 2000 °F.

Figures 13 through 24* present the physical property data as they are presently being used by the project staff. The thermal conductivity of carbon dioxide has been extrapolated. In extrapolation, an attempt was made to make the curve parallel to Geringer's data (35). Geringer's data appears to be consistently lower than the NBS data (31). The thermal conductivity of water vapor has been extrapolated using the equation of Keenan and Keyes (34). Thermal conductivity data for oxygen apparently is available only to 1000 °F. However, since the available data is nearly a straight line, it has been extrapolated to 2000 °F.

The estimation of the physical properties of a mixture of gases is somewhat of a problem. There is a dispute in the technical literature as to the proper relationships that should be used. Heat capacity of the flue gas was assumed to be a linear function of the weight fraction and was calculated using Equation 10.

$$C_{p_m} = C_{p_1} W_1 + C_{p_2} W_2 + \ldots C_{p_n} W_n \tag{10}$$

where C_p is heat capacity in Btu/lb- °F

W is the weight fraction

subscript m implies mixture

subscripts 1 through n are components 1 through n

Viscosity of the flue gas is estimated by Equation 11 according to Maxwell (37) and Perry (38).

$$\mu_m = \frac{\sum_{i=1}^{n} \mu_i x_i \sqrt{M_i}}{\sum^{n} x_i \sqrt{M_i}} \tag{11}$$

---

* See Appendix, pages 56-67.

where

μ_i = the viscosity in lb/ft hr for the i th component

x_i = the mole fraction of component i

M_i = the molecular weight of component i

i = 1, 2, ..., n numbers identifying the components

subscript m implies a mixture

The prediction of thermal conductivity of a gas mixture requires the use of an extensive set of equations. These are reported by Reid and Sherwood (39) and for the sake of brevity are not reported here.

Table XIII summarizes the various references used for determination of the physical properties of the flue gas components.

TABLE XIII

References Used for Physical Properties of Flue Gas

Flue Gas Component	Heat Capacity (Ref. No.)	Viscosity (Ref. No.)	Thermal Conductivity (Ref. No.)
N_2	31, 32	31	31, 33
O_2	31, 32	31	31, 33
CO_2	31, 32	31	31, 33, 35
H_2O	31, 32	31	31, 33, 34

Although Table XIII summarizes the sources of the physical property information used for preparation of the physical property curves, some question still exists as to whether all of this information is reliable enough for analytical purposes for all components in this laboratory investigation.

Theoretical calculations of the temperature of the flue gas leaving the burner as a function of the per cent excess air were made. Figure 9 presents the curve. Examination of this figure indicates that with 20% excess air the theoretical gas temperature could be expected to be 2780 °F. With 80% excess air the temperature would be 2080 °F. Table X indicates that the measured hot gas temperature for Run No. 7 with 80% excess air was 1425 °F. The calculated

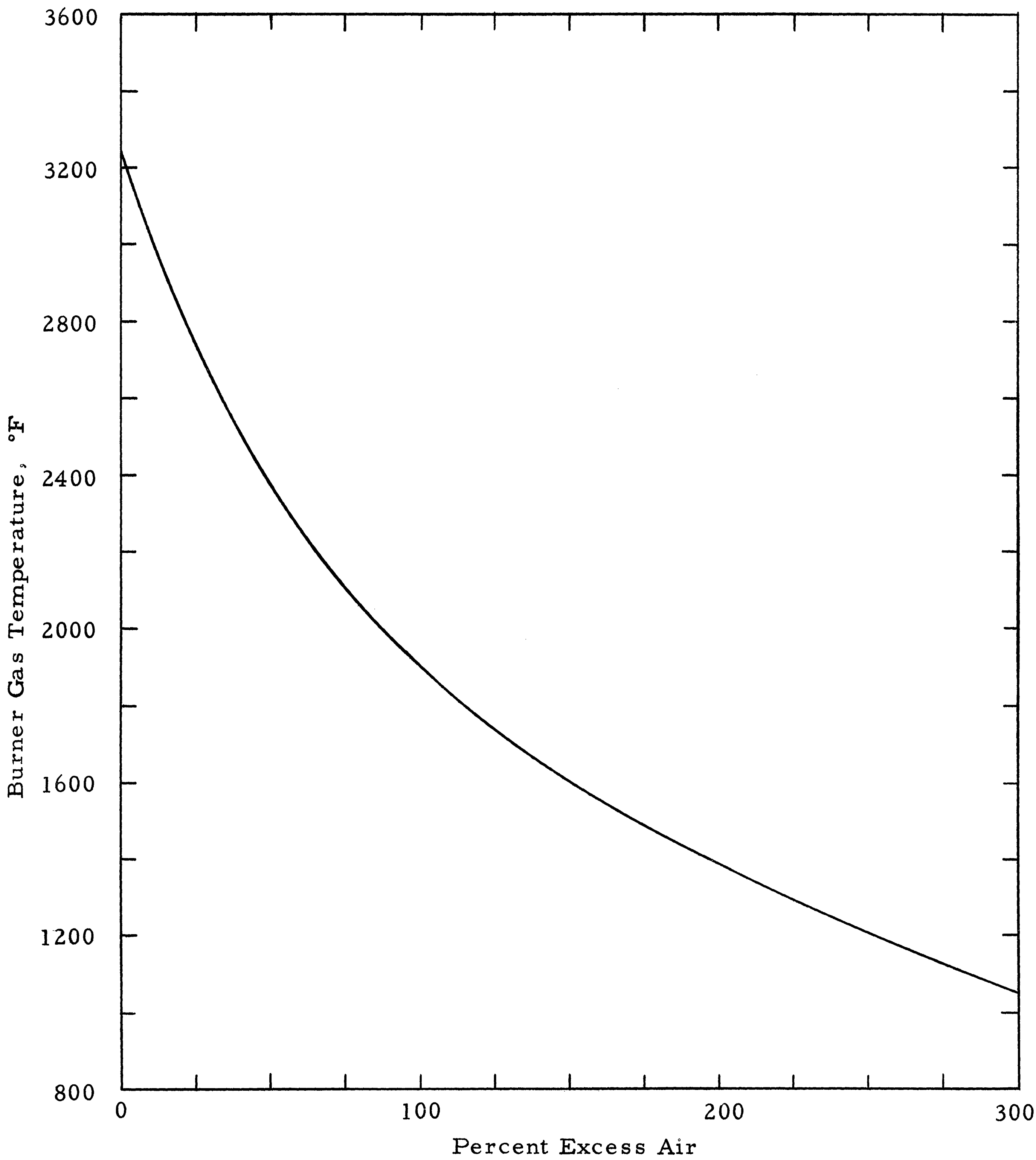

Fig. 9 Theoretical Temperature of Gas Leaving Burners

theoretical gas temperature assumes 100% combustion efficiency and no radiation affects. The calculations for Run No. 7 indicate that if the 82% efficiency and the direct radiation encountered are taken into account, then the measured gas temperature should have been approximately 1600 °F or higher.

The dew point of the hot flue gases is a factor in the performance of the experimental unit. During all experimental test runs, except for one, condensation of water on the finned tubes was encountered. The condensation affects the heat transfer performance. Theoretical calculations of the dew point of the flue gas assuming no water vapor in the combustion inlet air were made. Figure 10 presents the dew-point curve. The curve indicates a dew point of 132 °F for 20% excess air and a dew point of 119 °F for 80% excess air. This problem is aggravated on humid days. Condensation increases the heat transfer rate but corrodes the finned tubes.

As noted earlier, no general correlation for predicting the heat transfer coefficient for banks of finned tubes on a rectangular pitch exists. For comparison purposes, the correlation of Briggs and Young (15) for equilateral triangular pitch banks of finned tubes was used. The theoretical hot gas temperature was read from Figure 9 with an average water temperature of 100 °F assumed and gas film coefficients calculated. Figure 11 presents the predicted heat transfer coefficients as a function of natural gas feed rate with 20% and 80% excess air.

Table X indicates that the fuel gas rate was 30 cubic feet per minute for Run No. 7. Figure 11 would predict a coefficient of 7.4 for 80% excess air for triangular pitch banks of this finned tube with the same pitch. Table XII indicates that the calculated gas film heat transfer coefficient from the experimental data for this run was 10. This value is 35% higher than that predicted by the triangular pitch correlation. The actual experimental gas film coefficient for experimental Run No. 7 should be less than 7.4.

The experimental laboratory furnace was also instrumented for measuring the pressure drop of the flue gas flowing across the bank of finned tubes. To date no satisfactory reproducible measurements have been made. It is thought that the great change in density of the hot flue gas as it cools down in flowing across the finned tubes is upsetting the measurements. Various locations of the presure taps are being tried out in an attempt to obtain more realistic and reproducible results. A micro-manometer is being installed in an attempt to obtain more accurate results. However, it is believed that the fluctuations in gas pressure may possibly rule out the use of a micro-manometer.

The experimental data obtained to date indicate that the manufacturer of the experimental unit is correct in claiming that the unit operates at 80% efficiency.

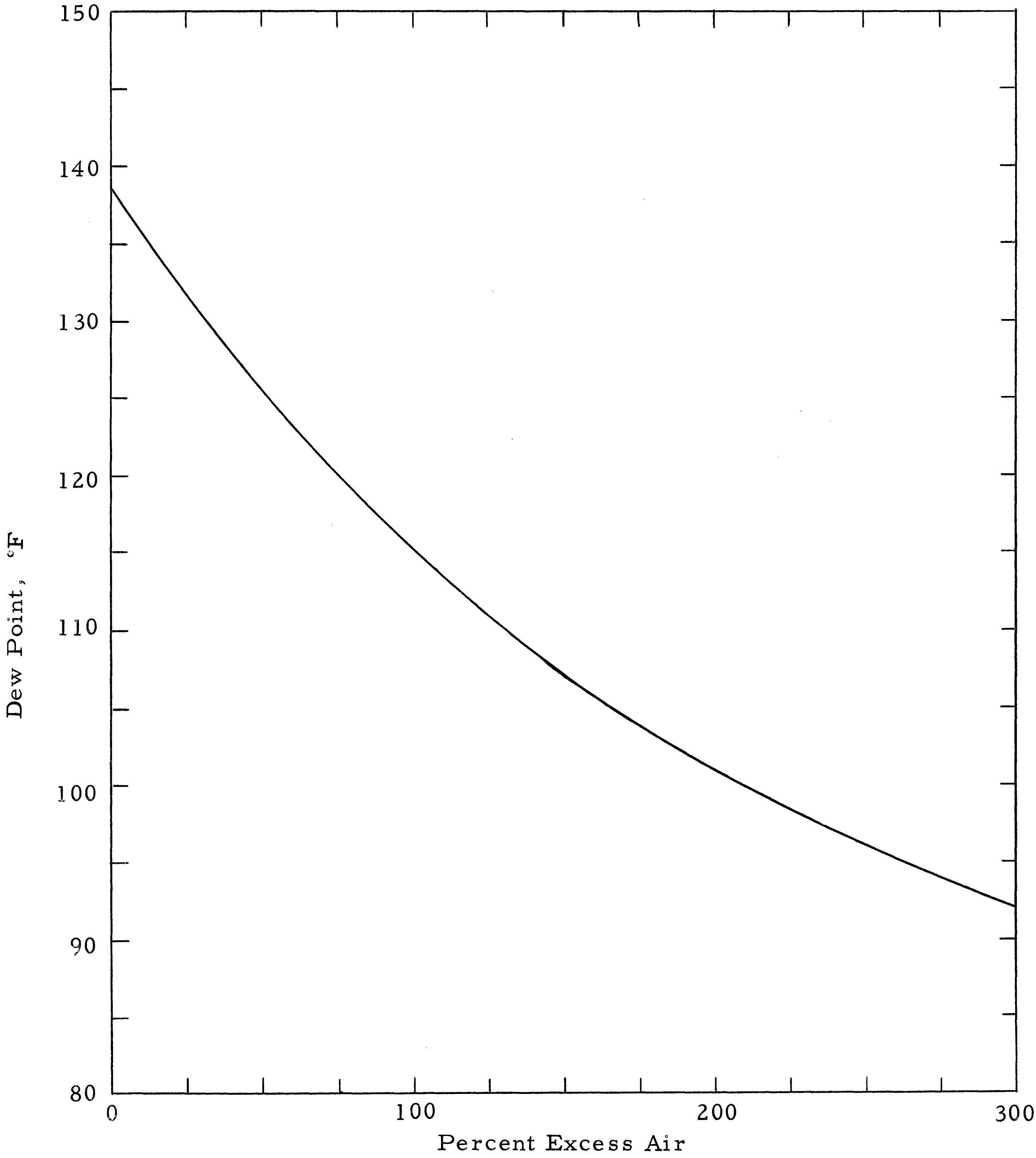

Fig. 10 Dewpoint of Flue Gas

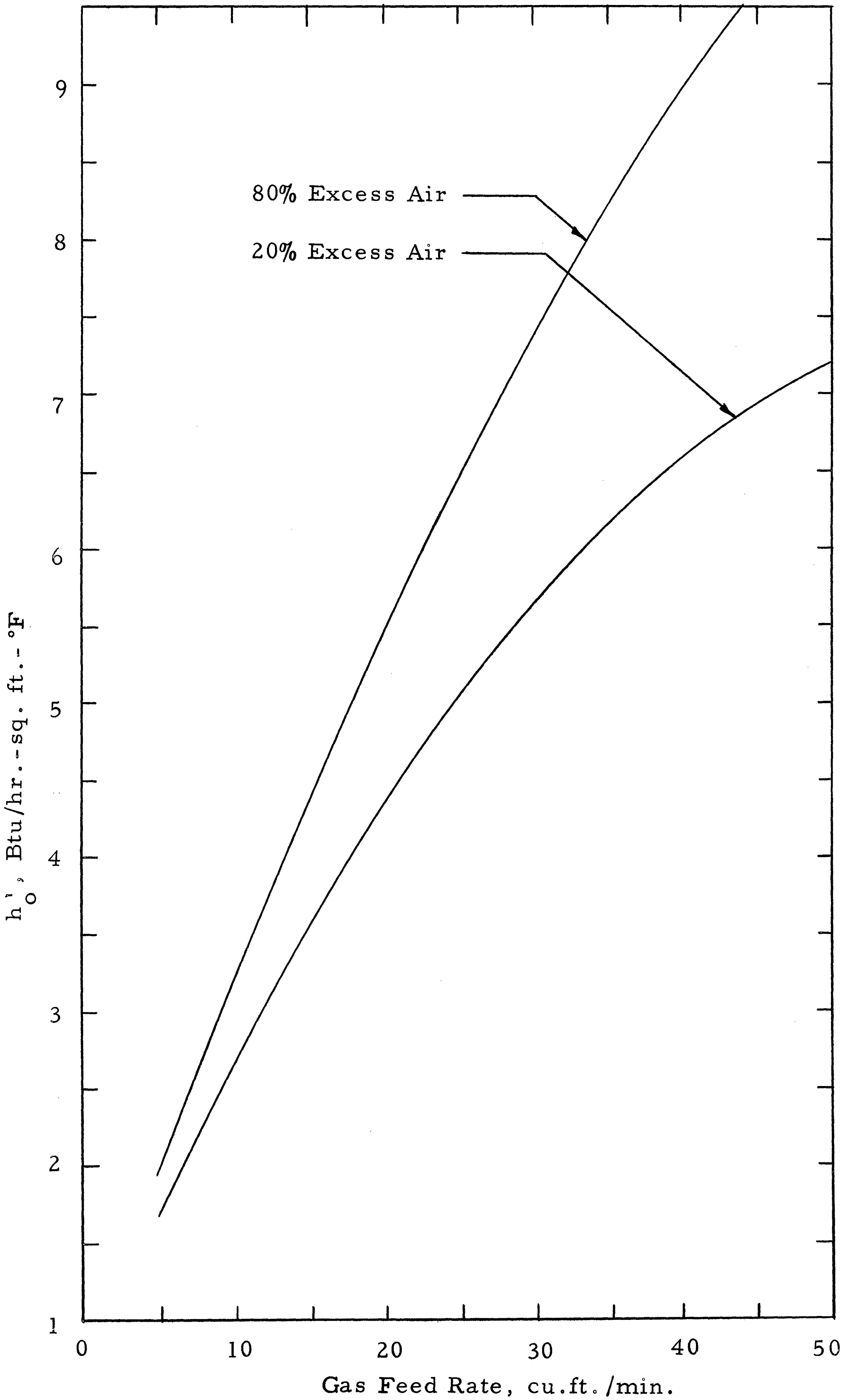

Fig. 11 Gas Film Coefficient for Triangular Pitch Tubes

In summary, the following situations were causing some difficulties at the time of the preparation of this progress report:

(1) Disagreement with the gas company on the actual composition of the natural gas used as fuel.

(2) The effect of the above disagreement on the analysis of the experimental data.

(3) Some disagreement, but not serious, between the Orsat analysis and mass spectrometer analysis of the "stack" or "flue" gas.

(4) Difficulty in accurately measuring the temperature of the hot flue gas (burner gas) below the finned tube bundle without radiation affects from the burner flames and re-radiation from the walls.

(5) The analysis of the experimental data obtained to date indicates that the radiation heat transfer is in excess of that predicted from theory.

(6) The analysis of the experimental data obtained to date also indicate that the calculated convective heat transfer coefficients may be too high.

(7) Some question exists as to whether the physical property data currently being used for some of the flue gas components are reliable at the elevated temperatures. (Some of the data are calculated from molecular model theory, others are semi-empirical.)

(8) Condensation of water vapor on fins below the dew point of the flue gas is upsetting the heat transfer since it is impossible to accurately collect and weigh the condensate formed.

(9) The per cent excess air measured in the experiments to date are believed to be considerably higher than those normally encountered in hot water heater installations.

(10) Difficulty is being encountered in obtaining satisfactory pressure drop measurements of the hot flue gas across the finned tubes.

The above ten items were being worked on at the time of the preparation of this progress report.

MODIFICATIONS TO THE EXPERIMENTAL APPARATUS AND FUTURE WORK

The use of multiple metal radiation shields around the thermocouples should reduce the radiation error by reducing the net rate of radiation from the thermocouple. Figure 12 shows a sketch of the radiation shields that will be used. These shields are patterned after a design suggested by Moffatt (40) and King (41).

In order to separate the heat transfer into the effect of each row, it was necessary to redesign the water flow from a four-pass design to a one-pass design. This necessitated fabrication of new water headers and tube sheets. Table XIV shows a list of the tubes that are on order from Wolverine Tube for experimental investigation. Performance of these tubes will be investigated in both triangular and square pitch arrangement until sufficient experimental data is available to rule out one of these pitch arrangements. The effect of the number of horizontal rows will be investigated by collecting experimental data on each horizontal row, then cutting out one horizontal row of tubes at a time and plugging the tube holes in the tube sheets with brass plugs.

TABLE XIV

Tubes on Order for Investigation

Tube No.	Rows	Tubes/Row	Bundle No.
61-0714065	2	6/pass (4 passes)	1
61-0714065	3	13	2
61-0710065	4	16	3
61-0720065	not determined		11
61-0516065	3	14	4
61-0916065 (if available) or 61-0920065	3	13	7
61-0520065	3	11	5
61-0714072	3	13	6
61-0910049	4	16	8
Largest available ID and fin height			
61-0905035	4	26	10

Thermocouples will be inserted through the exit water header cover plate into the tubes so that the temperature rise for the water in each row of tubes can be measured.

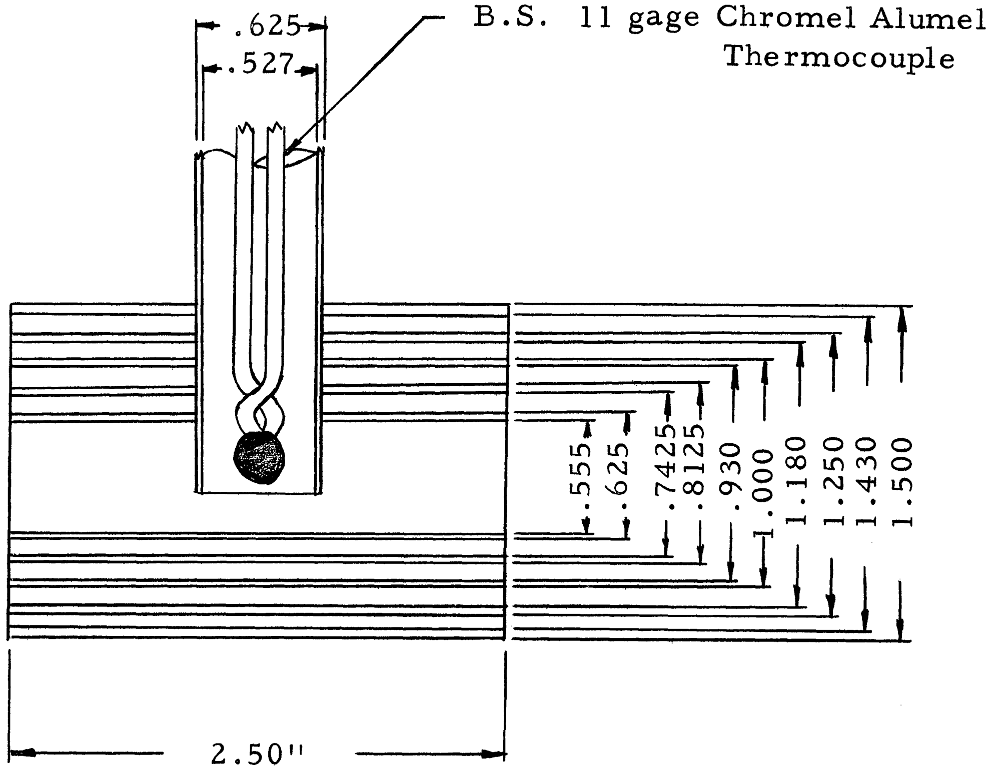

Fig. 12 Sketch of Thermocouple Shield Assembly

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APPENDIX

CALCULATION OF MASS SPECTROMETER RESULTS

The mass spectrometer always breaks a molecule up in a manner that is distinct for a given molecule and the peak height is always proportional to the partial pressure of the compound in the sample. The usual method of analysis is to define a sensitivity for a given compound as the number of divisions (Figure 6) produced per micron of partial pressure at a particular mass-to-charge ratio (M/E). All other peaks for that compound will be indicated as the ratio of the height of that peak to the height of the standard peak. With these definitions, one can readily show that Equation 12 describes the height of an unknown peak.

$$S_1 X_1 R_{1j} + S_2 X_2 R_{2j} + \ldots\ldots S_n X_n R_{nj} = P_j \tag{12}$$

where

S_i = sensitivity of component i

X_i = partial pressure of component i

P_j = peak number j in sample

R_{ij} = ratio of the peak of compound i at (M/E) number j to the standard peak of compound i

i = 1, 2, ..., n

j = 1, 2, ..., n

In order to solve for the value of X_i, we need n independent equations where n is the number of components in the mixture.

These n equations are as shown in Equations 13a to 13n.

$$S_1 X_1 R_{11} + S_2 X_2 R_{21} + \ldots S_n X_n R_{n1} = P_1 \tag{13a}$$

$$S_1 X_1 R_{12} + S_2 X_2 R_{22} + \ldots S_n X_n R_{n2} = P_2 \tag{13b}$$

$$\vdots \qquad \vdots \qquad \vdots \qquad \vdots$$

$$S_1 X_1 R_{1n} + S_2 X_2 R_{2n} + \ldots S_n X_n R_{nn} = P_n \tag{13n}$$

These equations are summarized in matrix form by Equation 14.

$$(R)\ (S)\ (X) = (P) \tag{14}$$

where:

$$(R) = \begin{matrix} R_{11} & R_{21} & \cdots & R_{n1} \\ R_{12} & R_{22} & \cdots & R_{n2} \\ \vdots & \vdots & & \vdots \\ R_{1n} & R_{2n} & \cdots & R_{nn} \end{matrix} \tag{15}$$

$$(S) = (S_1, S_2, \ldots, S_n) \tag{16}$$

$$(X) = (X_1, X_2, \ldots, X_n) \tag{17}$$

$$(P) = (P_1, P_2, \ldots, P_n) \tag{18}$$

The sensitivity vector, (S), and the partial pressure vector, (P), may be combined to define a new variable (Y) as shown in Equation 19.

$$(Y) = (S)\ (X) = (S_1X_1, S_2X_2, \ldots, S_nX_n) \tag{19}$$

Substituting Equation 19 into Equations 13a through 13n gives Equations 20a through 20n.

$$R_{11}Y_1 + R_{21}Y_2 + \ldots R_{n1}Y_n = P_1 \tag{20a}$$

$$R_{12}Y_1 + R_{22}Y_2 + \ldots R_{n2}Y_n = P_2 \tag{20b}$$

$$\vdots \qquad \vdots \qquad \vdots \qquad \vdots$$

$$R_{1n}Y_1 + R_{2n}Y_2 + \ldots R_{nn}Y_n = P_n \tag{20n}$$

This set of equations can be recognized as a simple set of linear algebraic equations. Equations 20a through 20n are summarized in matrix notation in Equation 21.

$$(R)\ (Y) = (P) \tag{21}$$

The theory of matrices (42) allows these equations to be solved very quickly on a digital computer as a large number of the non-diagonal elements of the matrix (R) are zero.

A computer program has been written in the MAD language to solve these equations for any set of unknowns consisting of less than ten compounds. Provision is included to calculate the (R) matrix from a set of standards and then to use these standards to solve for the partial pressure and mole fraction of the compounds present in a set of unknowns. Tables XV, XVI, and XVII show a sample computer output for Runs 7A and 7C.

TABLE XV

Typical Digital Computer Output for Mass Spectrometer Analysis of Natural Gas

Sample No. 14101 for Run 7, Feed on December 10, 1964

M/E Compound	Sensitivity	Peak Height	Partial Pressure	Percent Composition
15.0 Methane	89.0359	4900.00	54.783	86.409
28.0 Nitrogen	132.8002	1278.00	2.353	3.712
30.0 Ethane	30.8868	155.70	4.915	7.753
44.0 Carbon Dioxide	159.1601	83.20	.229	.362
29.0 Propane	163.3267	312.00	1.012	1.597
43.0 n-Butane	207.2247	64.20	.027	.043
42.0 i-Butane	77.1060	17.30	.079	.124
			63.399	100.000
		Sample Pressure	.000	

Heating Value is 1054.711 Btu/cu.ft. at 60 °F

TABLE XVI

Typical Digital Computer Output for
Mass Spectrometer Analysis of
Stack Gas for Run No. 7A

Sample No. 14106 for Run 7A, on December 1, 1964

M/E Compound	Sensitivity	Peak Height	Partial Pressure	Percent Composition
28.0 Monoxide, Carbon	128.1727	4830.00	.000	.000 *
14.0 Nitrogen	19.6831	806.00	40.949	83.089
32.0 Oxygen	98.7805	499.00	5.050	10.248
44.0 Dioxide, Carbon	144.2210	398.00	2.760	5.600
18.0 Water Vapor	93.2000	2.30	.025	.050
40.0 Argon	163.8000	81.80	.499	1.013
			49.283	100.000
		Sample Pressure	49.230	

* Negative answers are considered zero

Combustion Analysis
H/C = 4.457
Excess Air = 86.560

TABLE XVII

Typical Digital Computer Output for
Mass Spectrometer Analysis of
Stack Gas for Run No. 7C

Sample No. 14107 for Run 7C, on December 1, 1964

M/E Compound	Sensitivity	Peak Height	Partial Pressure	Percent Composition
28.0 Monoxide, Carbon	128.1727	4780.00	.000	.000 *
14.0 Nitrogen	19.6831	810.00	41.152	83.083
32.0 Oxygen	98.7805	479.00	4.848	9.788
44.0 Dioxide, Carbon	144.2210	434.00	3.009	6.075
18.0 Water Vapor	93.2000	2.00	.021	.043
40.0 Argon	163.8000	82.00	.501	1.011
			49.531	100.000
		Sample Pressure	49.230	

* Negative answers are considered zero

Combustion Analysis
H/C = 4.097
Excess Air = 79.588

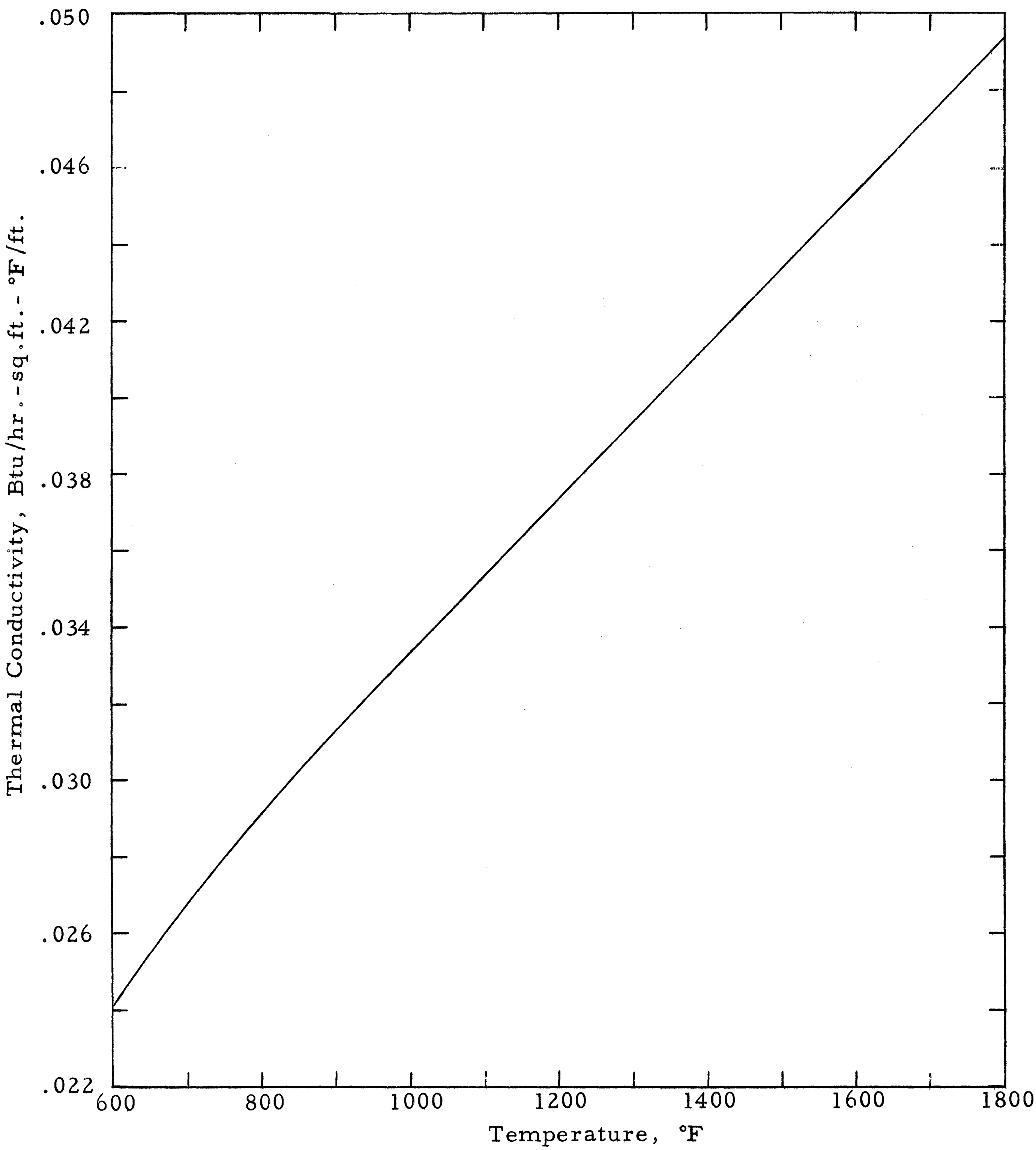

Fig. 13 Thermal Conductivity of Carbon Dioxide

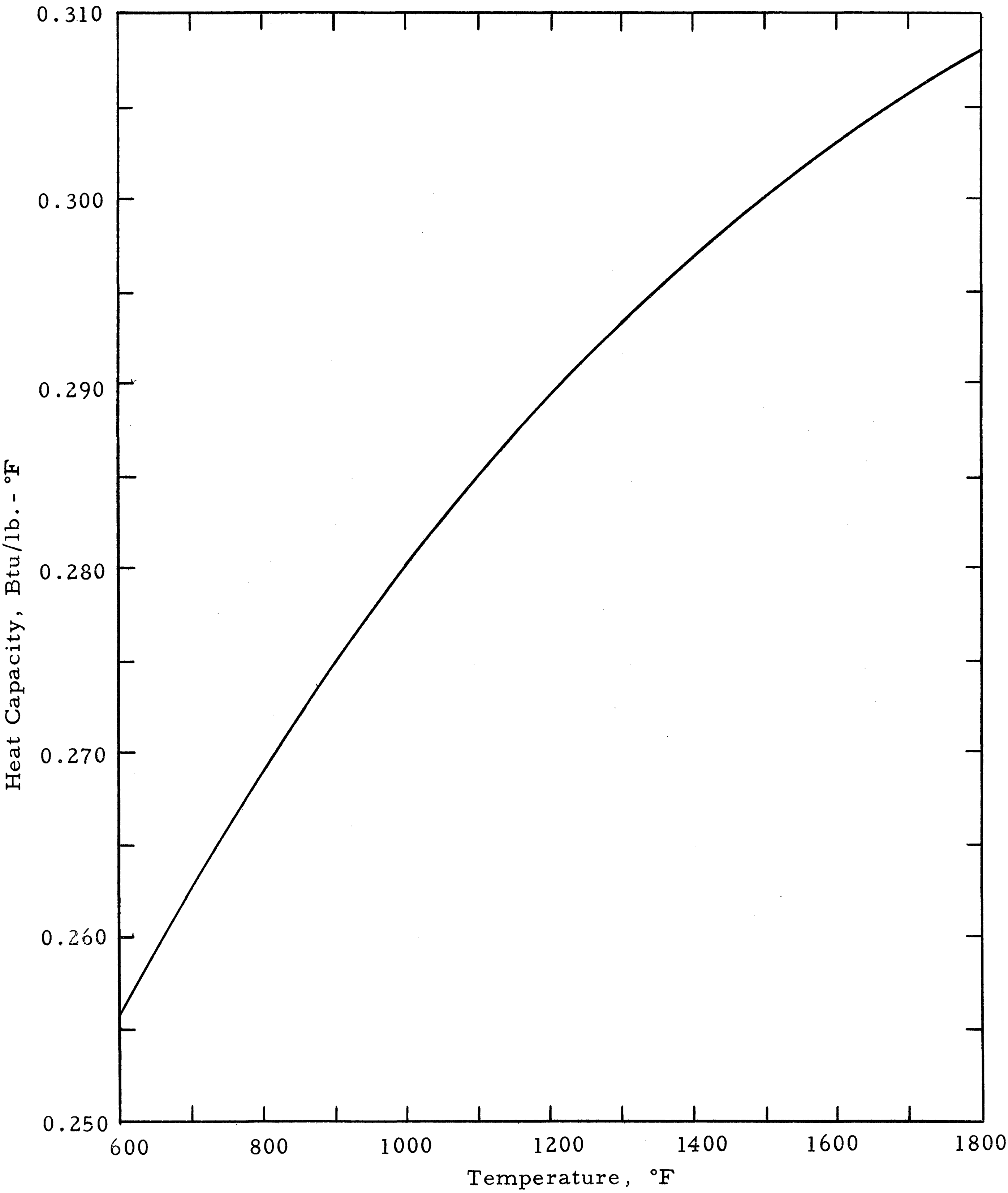

Fig. 14 Heat Capacity of Carbon Dioxide

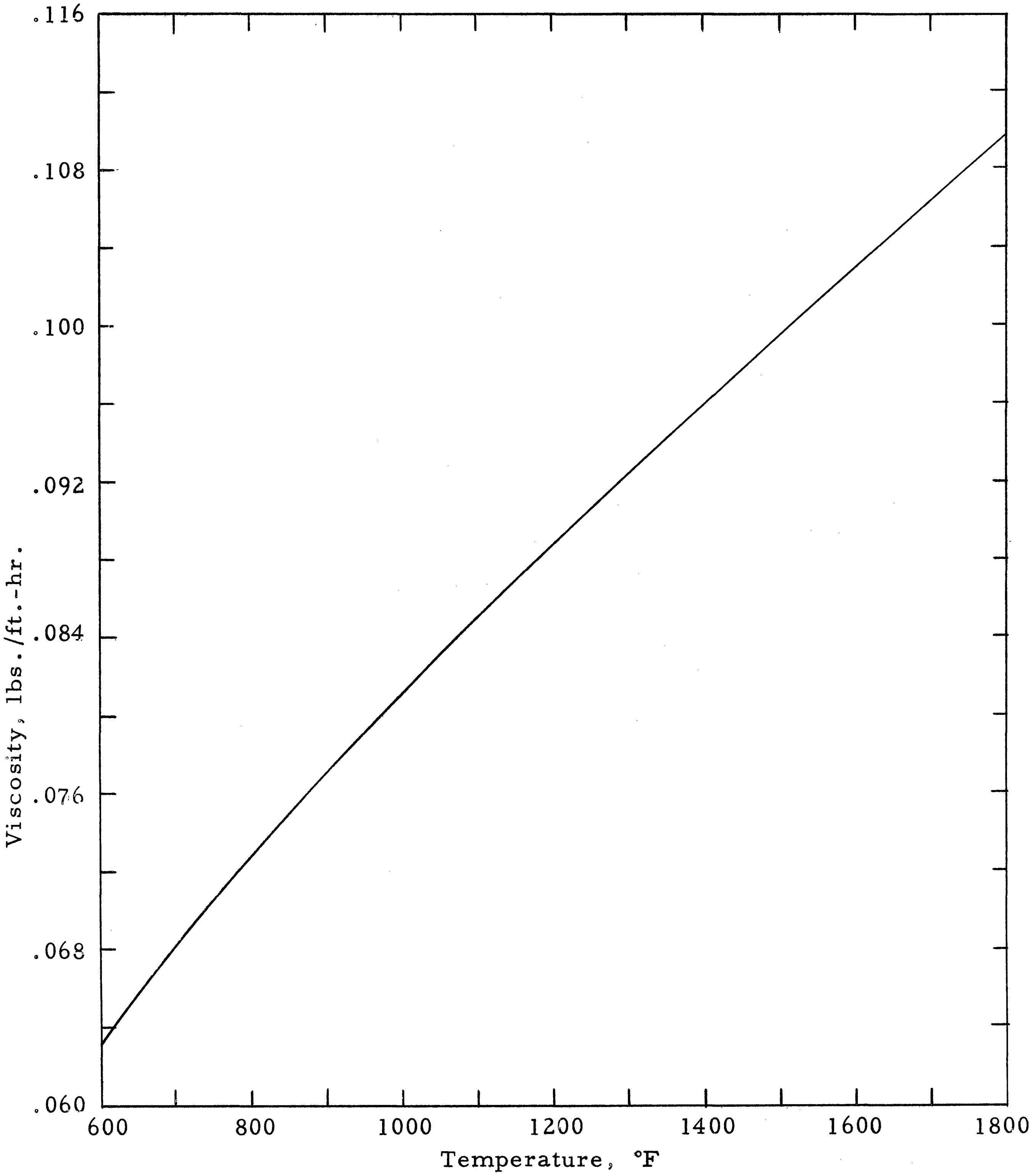

Fig. 15 Viscosity of Carbon Dioxide

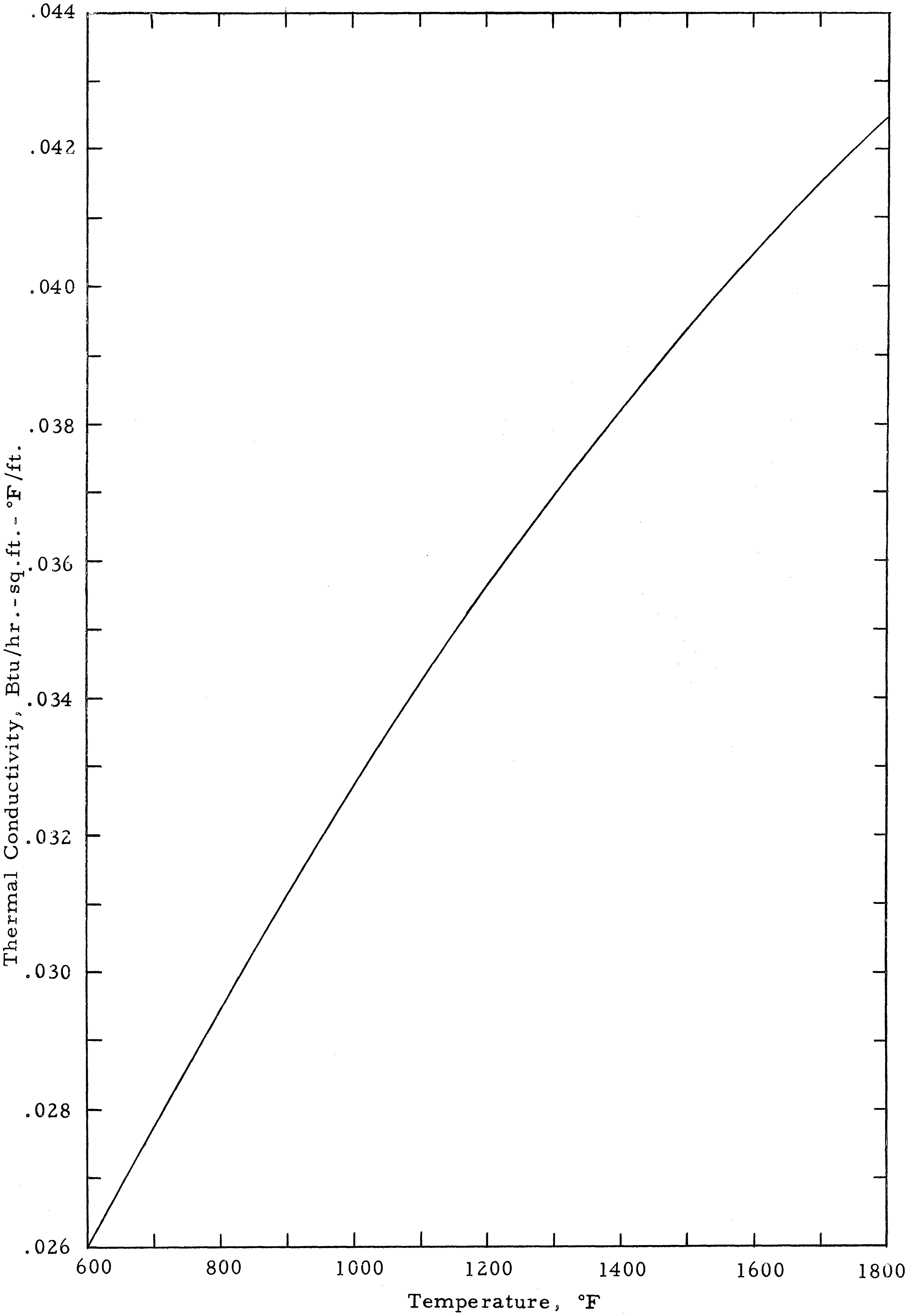

Fig. 16 Thermal Conductivity of Nitrogen

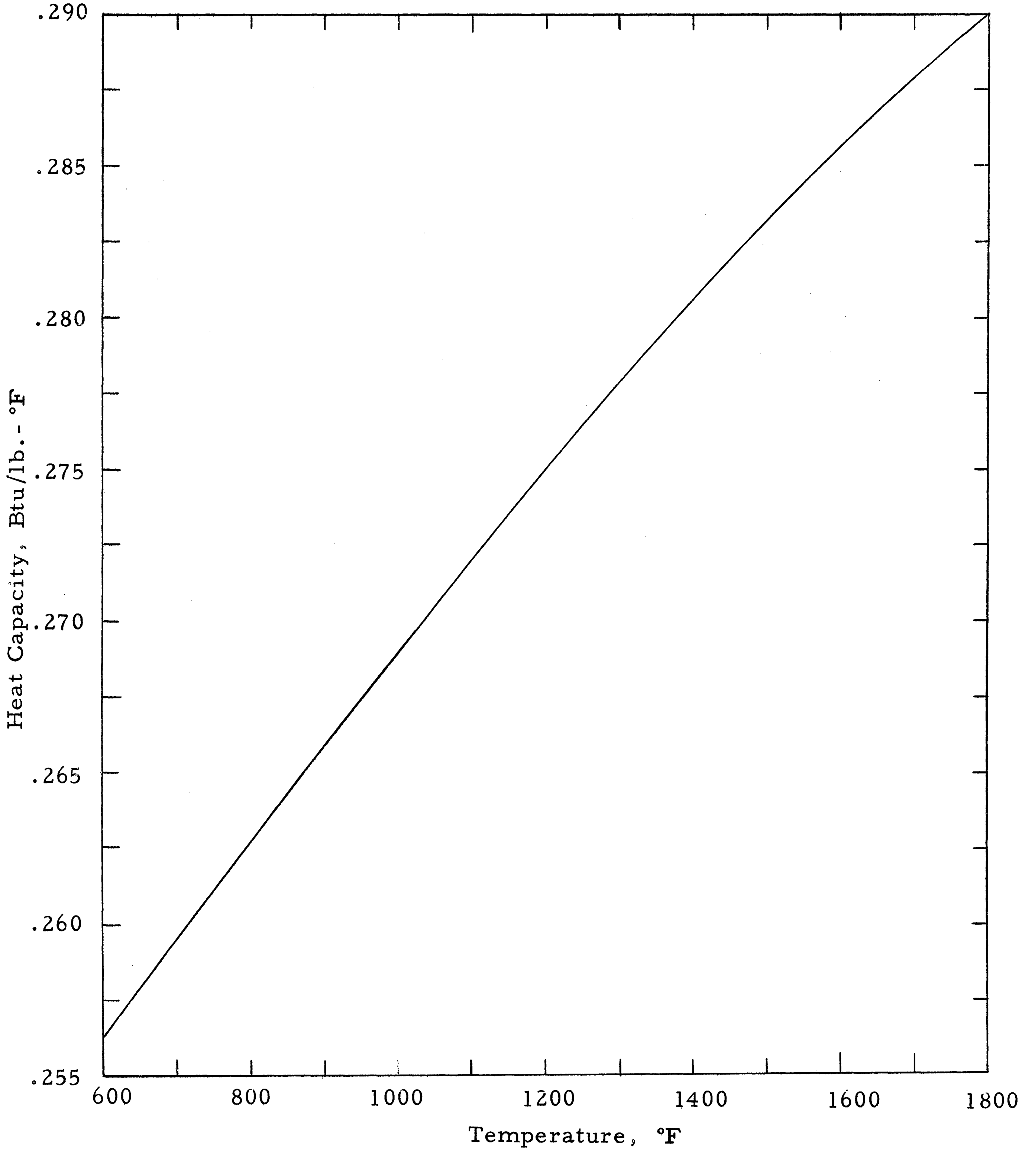

Fig. 17 Heat Capacity of Nitrogen

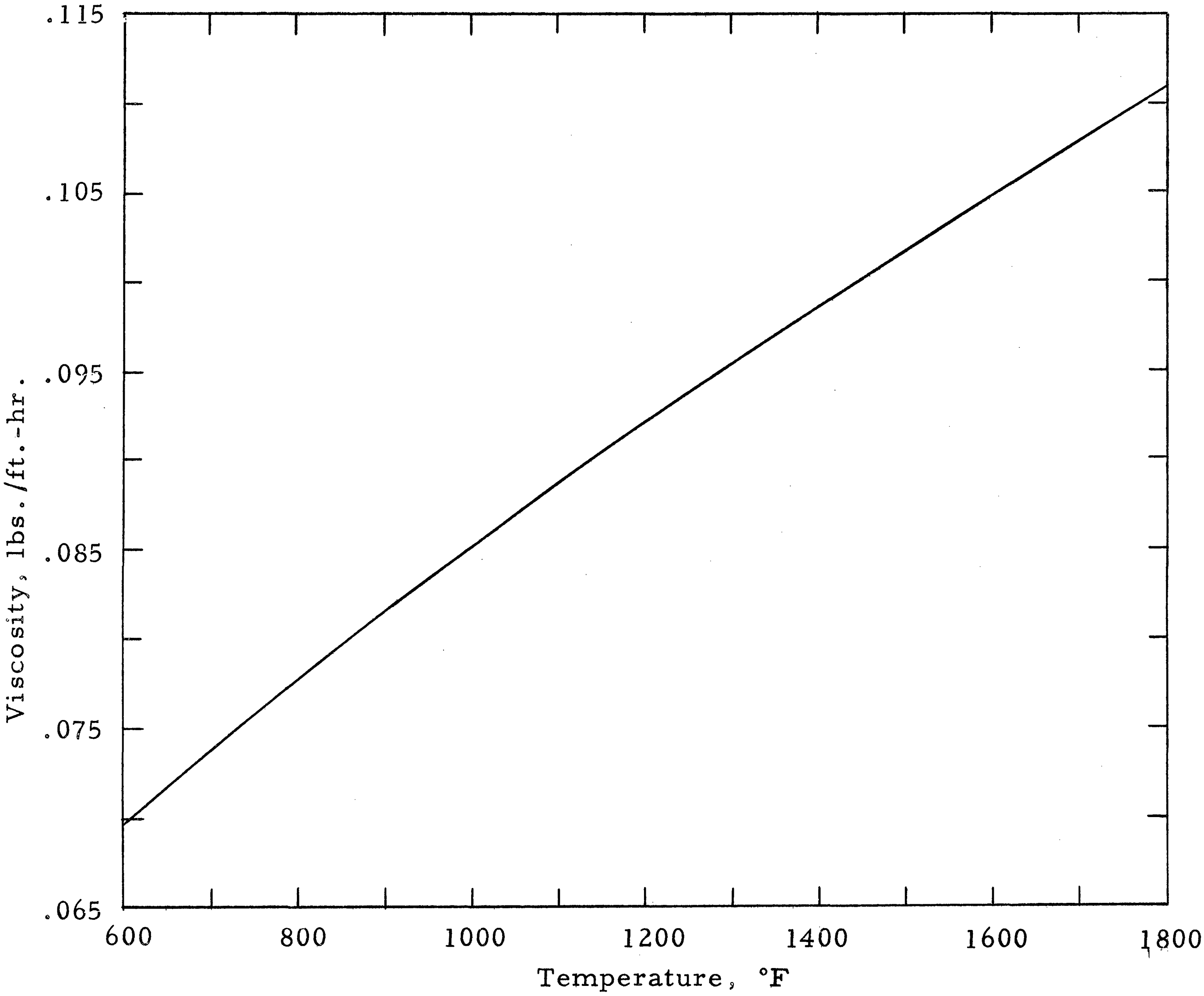

Fig. 18 Viscosity of Nitrogen

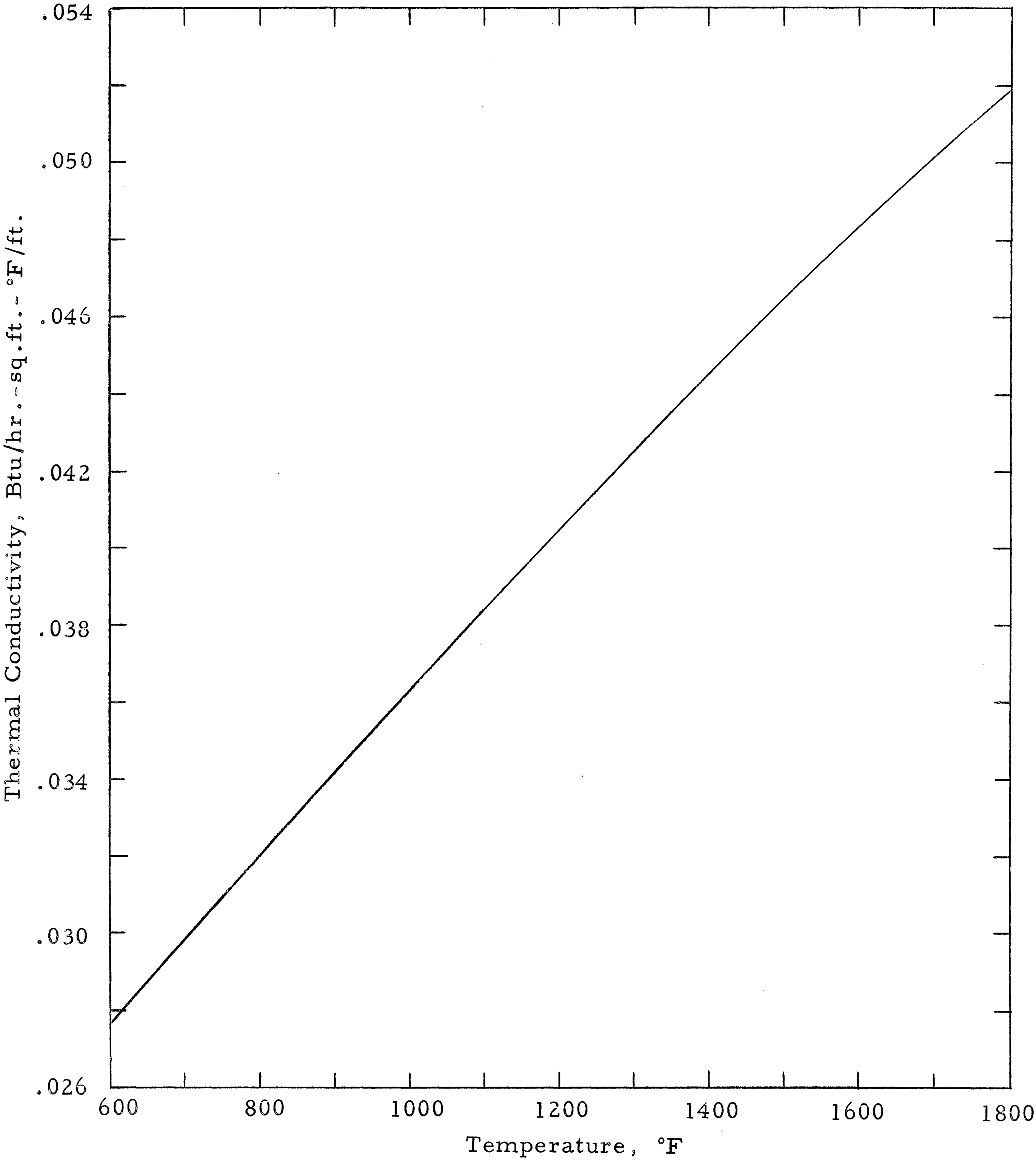

Fig. 19 Thermal Conductivity of Oxygen

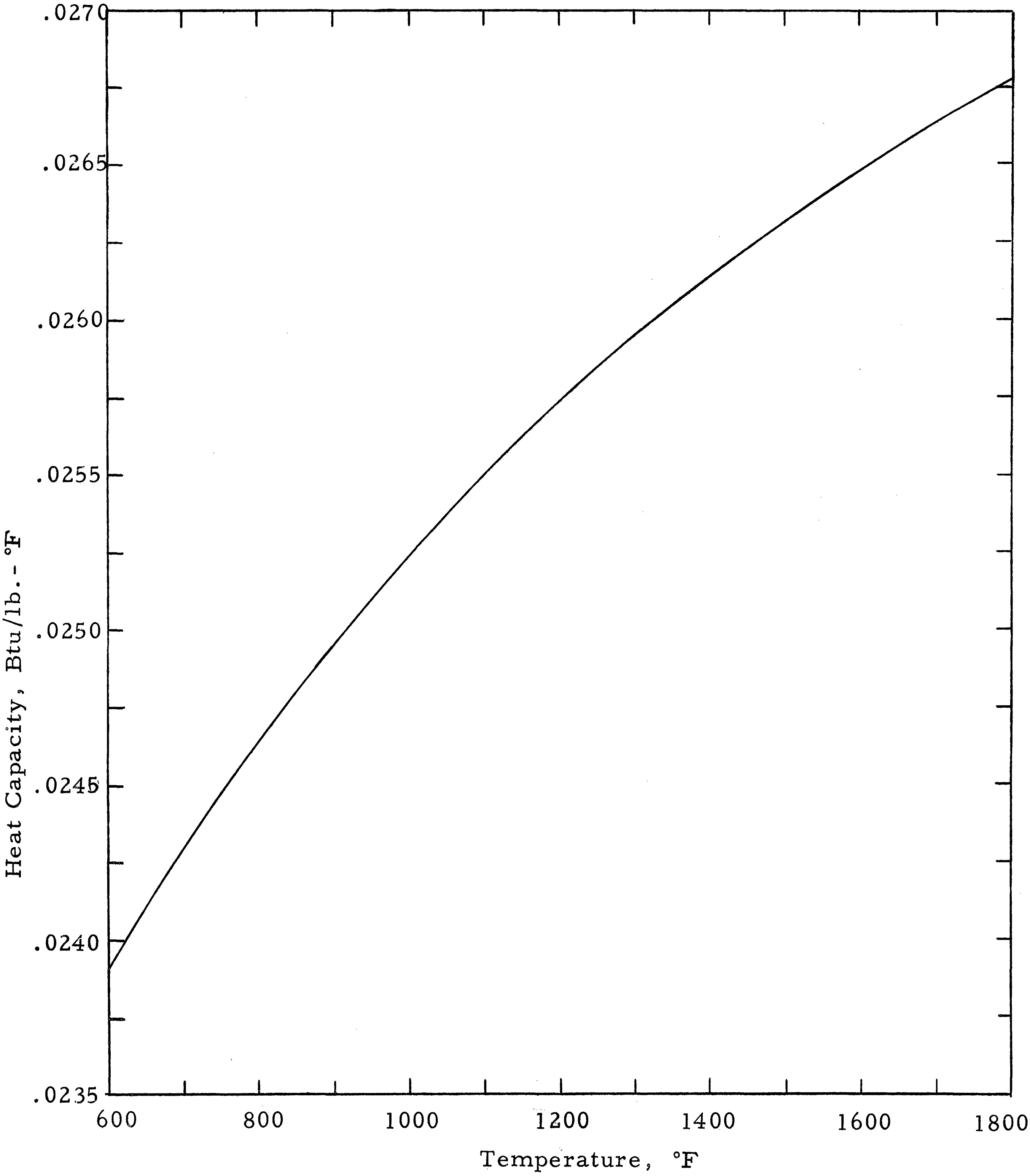

Fig. 20 Heat Capacity of Oxygen

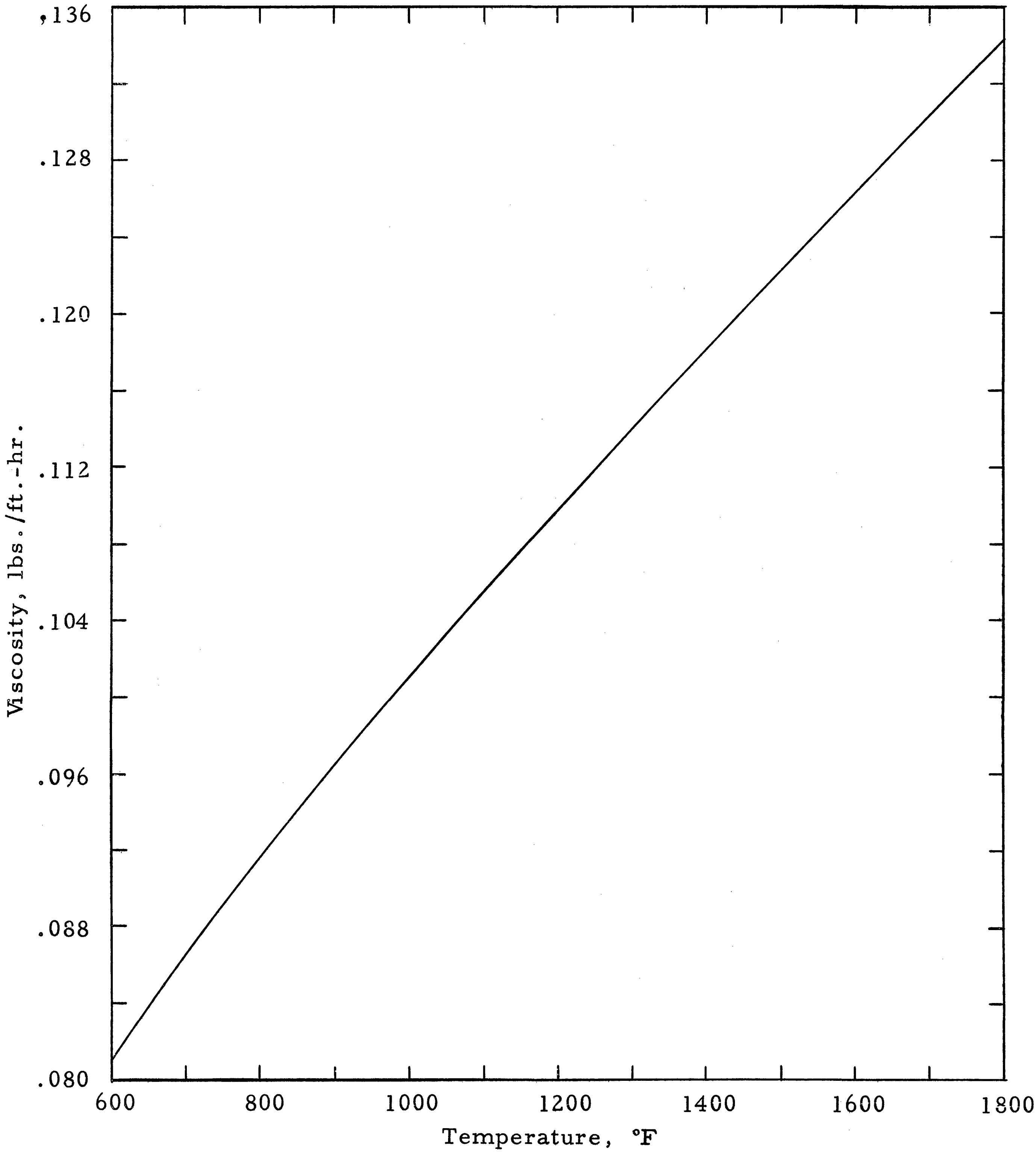

Fig. 21 Viscosity of Oxygen

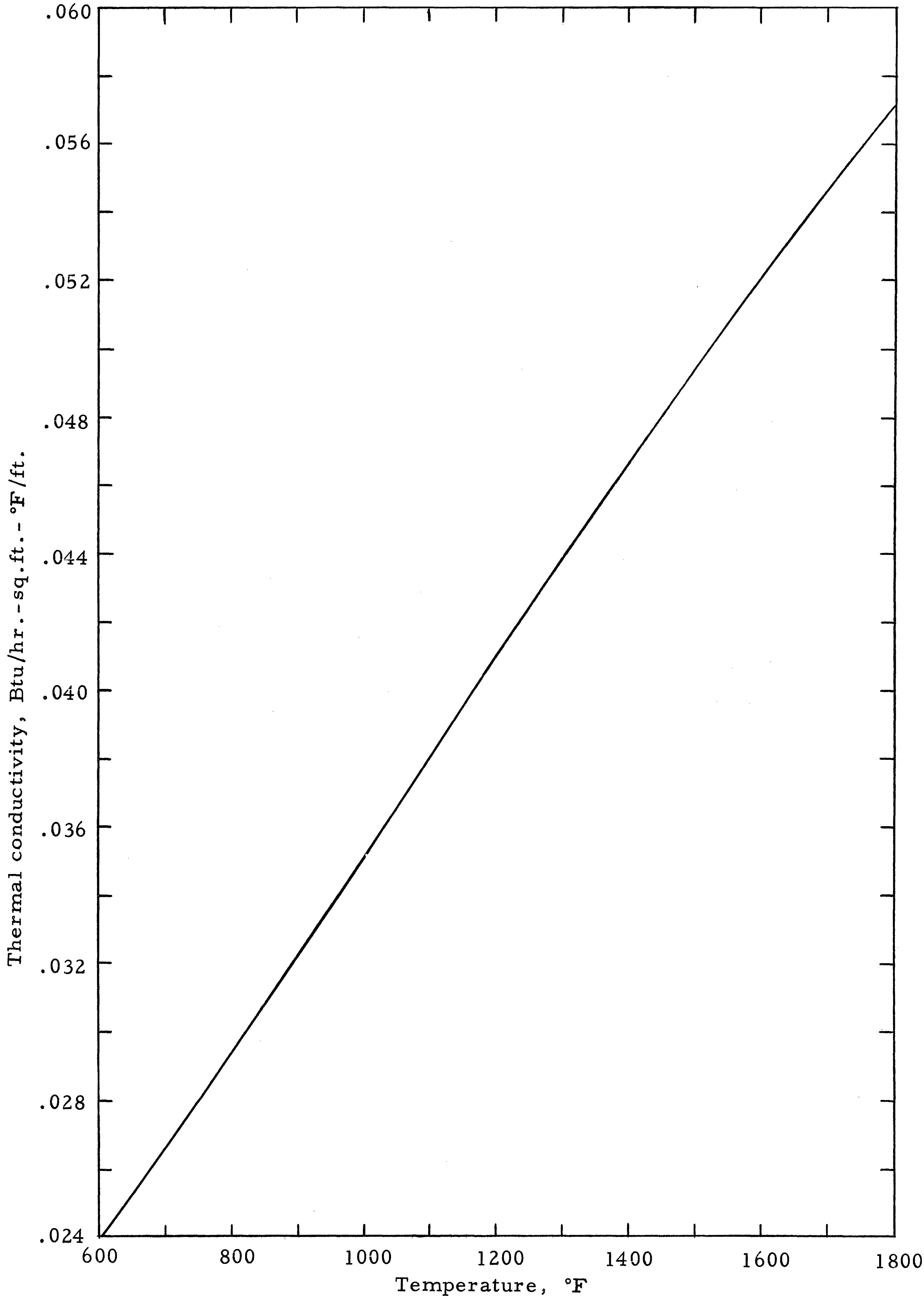

Fig. 22 Thermal Conductivity of Water Vapor

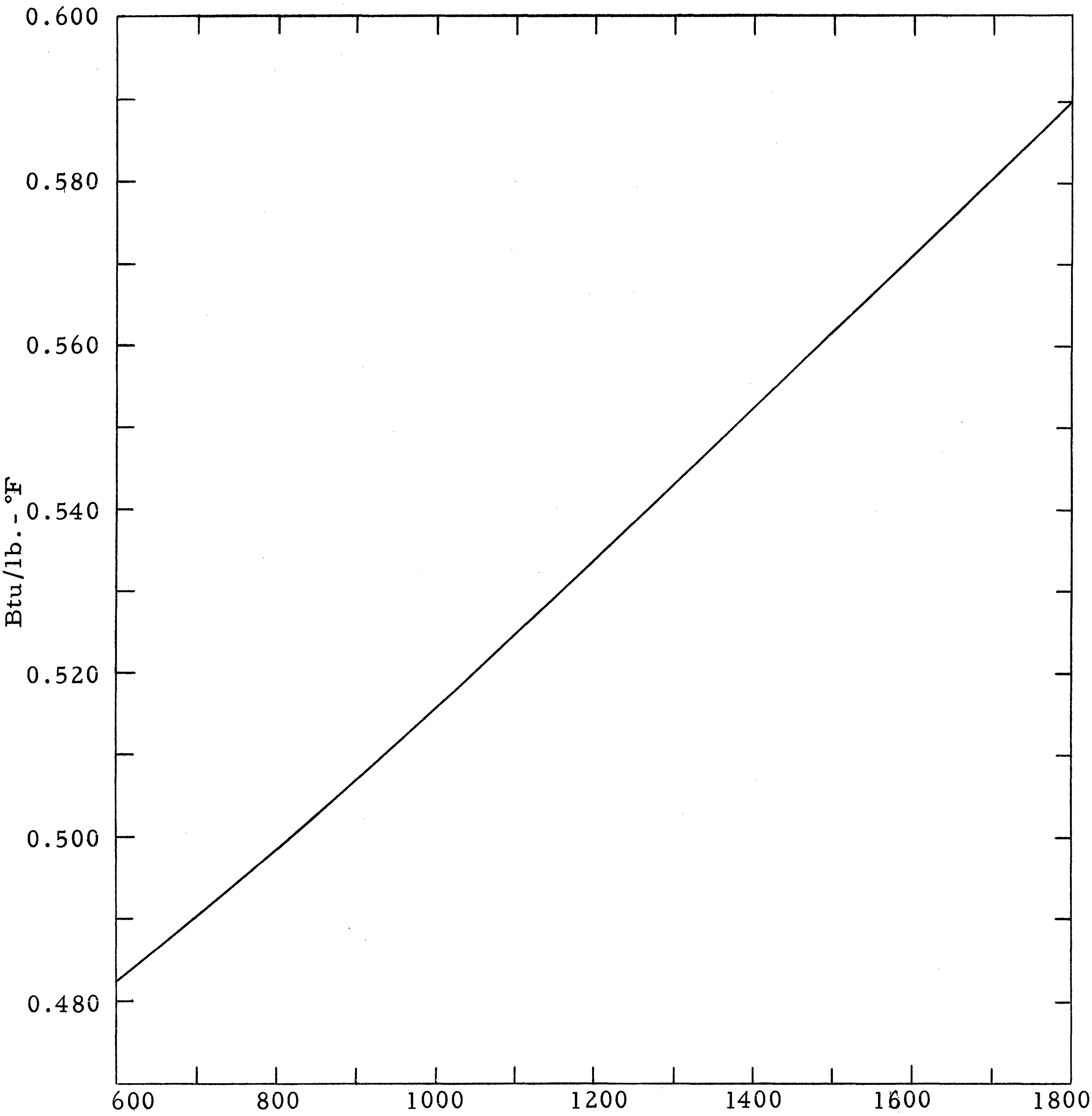

Fig. 23 Heat Capacity of Water Vapor

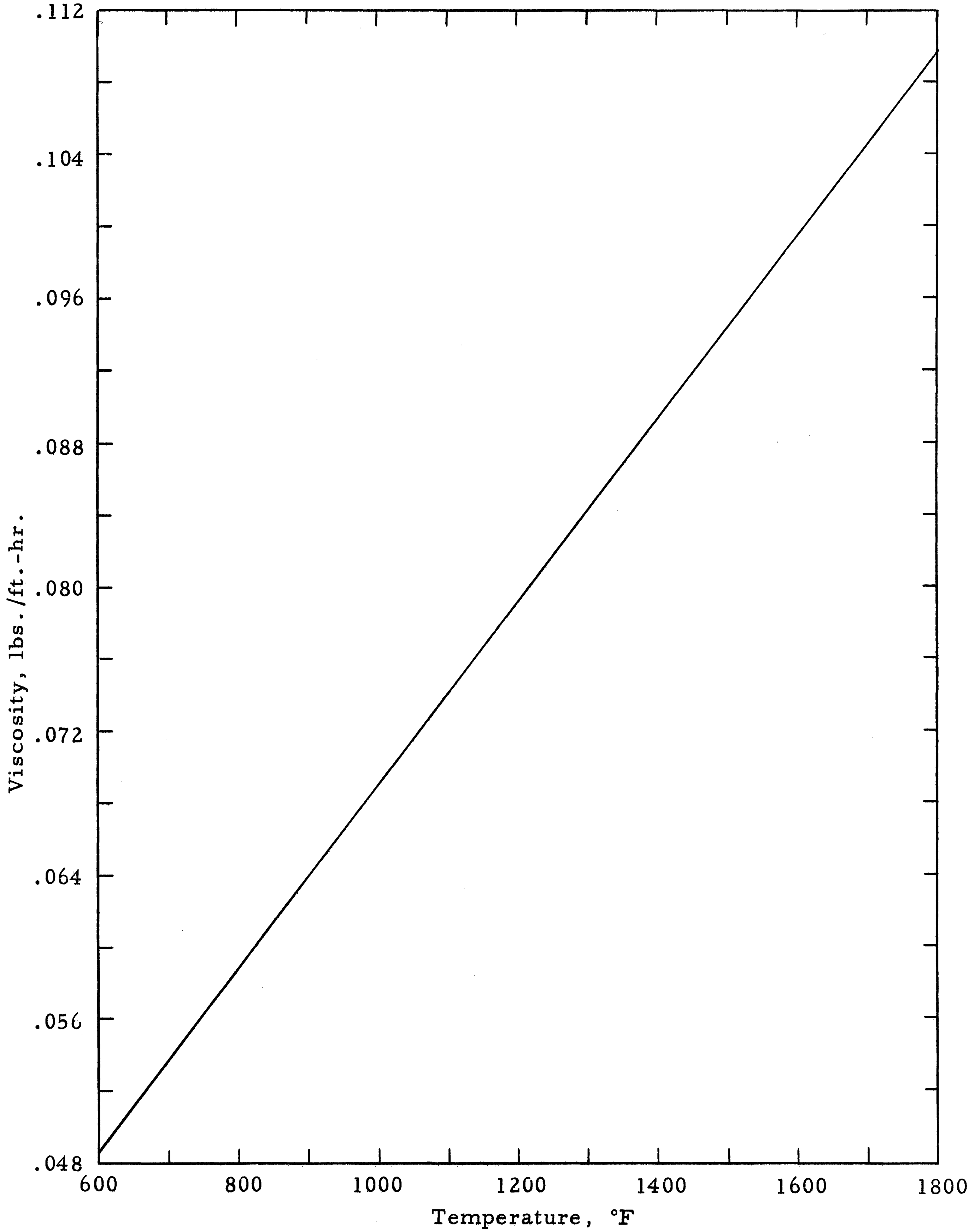

Fig. 24 Viscosity of Water Vapor