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GENERAL APPLIED SCIENCE LABORATORIES INC. — Subsidiary of THE Marquardt CORP

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ENGINEERING ANALYSES AND DESIGN CALCULATIONS OF NASA - LENGLEY RESEARCH CENTER HYDROGEN-AIR-VITIATED HEATER WITH OXYGEN REPLENISHMENT

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SECTION I

INTRODUCTION AND SUMMARY

This report presents the technical basis for the design of the Hydrogen-Air-Vitiated Heater. The heater liner is subjected to a maximum thermal environment at specified conditon z' (Figure I-1), where the combustion gas temperature, pressure and flow rate are 5000 F, 750 psia, and 11.0 lb/sec, respectively, and results in a heat flux of the order of 275 BTU/sec-ft^2. Cooling and stress analyses indicate that water is the logical choice for cooling of the combustor liner. A cylindrical shell of zirconium copper was selected as the combustor liner. This material, a high copper alloy, was chosen primarily because of its high thermal conductivity (k = 200 BTU/hr-ft^2) as well as good yield strength (35,000 psi) in a forged condition. Additionally by using a water cooled liner there is built into the design, potential for future extensions to more severe thermal environments beyond the present specification envelope.

A mixing analysis was undertaken to establish a good combination of combustor length and injector configuration. The analysis, using a conservative analytical approach, indicates a combustor length of the order of 5 ft combined with discrete fuel and oxidizer injection at an approximate 2-1/2 inch radial combustor position, and results in uniform combustion products at the heater exit for all specified envelope conditions.

Equilibrium composition and performance curves were prepared to permit rapid determination of air, oxygen and hydrogen gas flow requirements relative to total gas flow, as well as total flow relative to heater pressure, temperature, and nozzle throat.

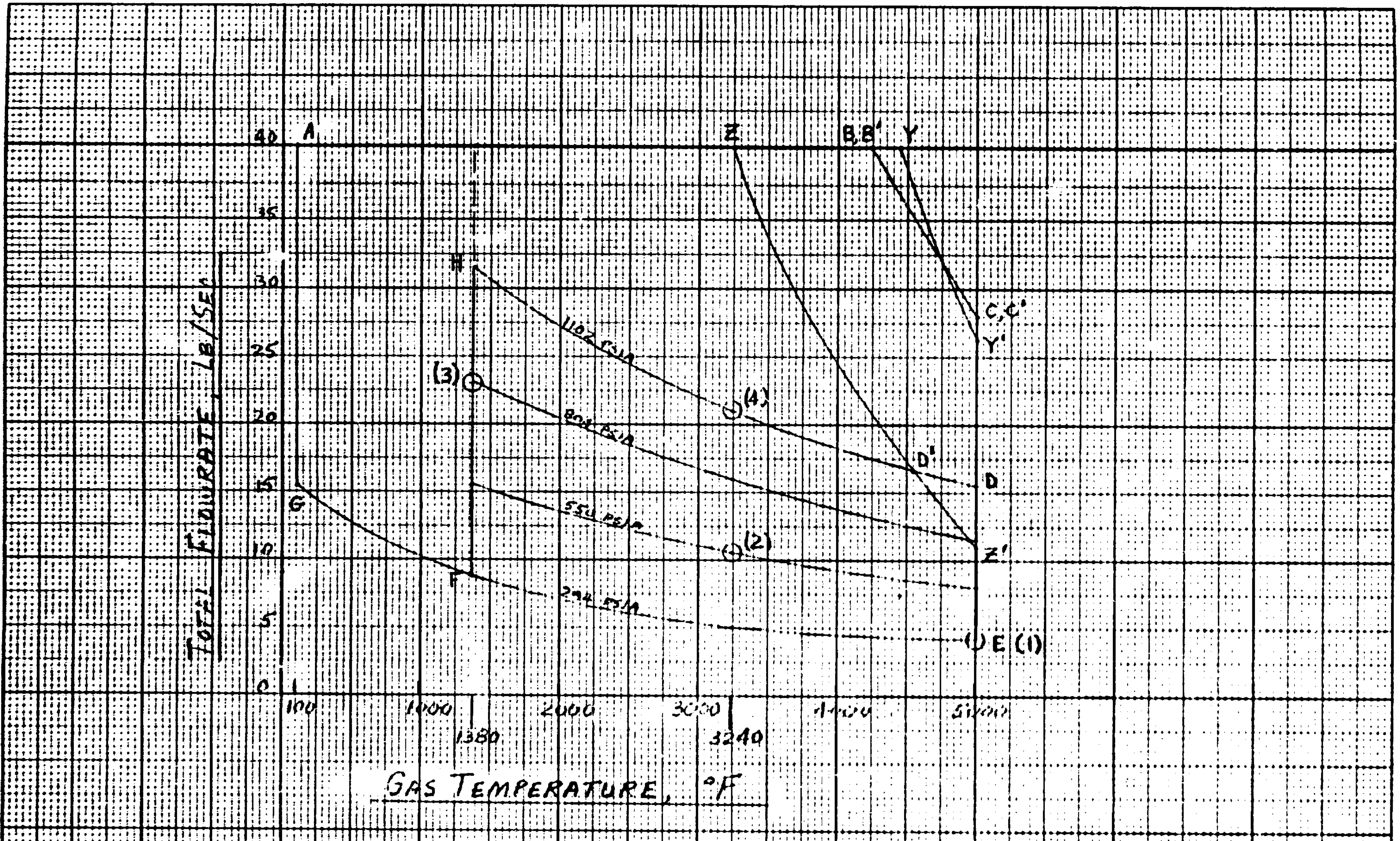

FIG I-1 PERFORMANCE AND DESIGN LIMITS

SECTION II

EQUILIBRIUM COMPOSITION AND PERFORMANCE CURVES

The equilibrium composition of the gaseous system was computed utilizing existing computer programs described in detail in GASL Technical Report No. 676 (Ref. II-1). This report presents the results of a recent analytical investigation of the effects of vitiated air contamination on combustion and hypersonic airbreathing engine ground tests. Shifting equilibrium calculations for hydrogen vitiated facility combustors and nozzles were performed by minimizing the Gibbs Free Energy of the entire system, subject to the constraint of element conservation, and computing the equilibrium composition of the reacting gas. In addition, finite rate chemistry calculations were made employing reaction mechanisms and high speed computational techniques. The results indicate that the assumption of chemical equilibrium is adequate for facility nozzle flow determination over the range of tunnel conditions of interest.

The equations used to compute the combustion gas properties are based on the following assumptions:

1. The process is adiabatic and one-dimensional;
2. The gas is in chemical equilibrium;
3. The nozzle expansion is isentropic.

The initial species concentrations of hydrogen, oxygen, and nitrogen were computed from the following reactions:

$$\frac{Y_1}{2} H_2 + \frac{Y_2}{32} O_2 + \frac{Y_3}{28} N_2 \rightarrow \frac{Y_4}{18} H_2O + 0.232\ O_2 + \frac{Y_5}{28} N_2$$

$$1 \text{ lbm Air} = 0.232 \text{ lbm } O_2 + 0.768 \text{ lbm } N_2$$

where the product species have been evaluated at conditions corresponding to flow in the test section. Thus, there will be 23.2 percent oxygen by weight in the combustion products in the wind tunnel test section.

The flow is assumed to be steady and one-dimensional, and ideal gas is assumed throughout. Under these conditions, the relevant equations are:

Equation of State:

$$\rho = \frac{P\bar{M}}{RT}$$

Momentum Conservation:

$$\rho u \frac{du}{dx} + \frac{dp}{dx} = 0$$

Energy Conservation:

$$\sum_i \left[X_i \frac{dh_i}{dx} + h_i \frac{dX_i}{dx}\right] + u \frac{du}{dx} = 0$$

Continuity:

$$w = \rho u A$$

or

$$\frac{d\rho}{\rho} + \frac{du}{u} + \frac{dA}{A} = 0$$

The results of these calculations are presented in Figures II-1 through II-5 and permit the rapid determination of air, oxygen, and hydrogen propellant requirements relative to total propellant flow, as well as total propellant flow relative to heater pressure, temperature, and nozzle throat area.

Curves of the reactant species concentrations for hydrogen vitiated air are shown in Figure II-1. The reactants were injected into the burner at an initial temperature of 100°F. The results indicate the compositions required to increase the temperature of "clean" air from ambient to stagnation temperatures in excess of 5000^{o}F for stagnation pressures between 300 and 1200 psia. At the lower pressures more energy is absorbed into the increased dissociation, resulting in lower stagnation temperatures. The maximum temperature attainable, with oxygen replenishment, is determined by the case of zero air injection.

The product species resulting from this vitiation, expanded to conditions corresponding to flow in the facility test section where the primary species are frozen in composition, are shown in Figure II-2. These results are for a gas in chemical equilibrium, and are plotted as a function of tunnel stagnation temperature. The molecular weight variation of hydrogen vitiated air is also shown. It is observed that increased vitiation (higher fuel injection) acts to decrease the molecular weight of hydrogen vitiated air due to the increased concentration of low molecular weight hydrogen.

Figure II-3 presents a plot of the vitiated air mass flow parameter ($\dot{m}/P_o A^*$) as a function of tunnel stagnation temperature. This curve permits rapid determination of the total mass flow rate of propellant. Figures II-4 and II-5 present curves of the mass flow parameter specialized for a particular nozzle having a throat area equal to 2.38 in^2 and operating at stagnation pressures of P_o = 1103, 804, 550, 294 psia.

References

II-1 Edelman, R. B. and Spadaccini, L. J., "Analytical Investigation of the Effects of Vitiated Air Contamination on Combustion and Hypersonic Airbreathing Engine Ground Tests," GASL TR-676, August 1968.

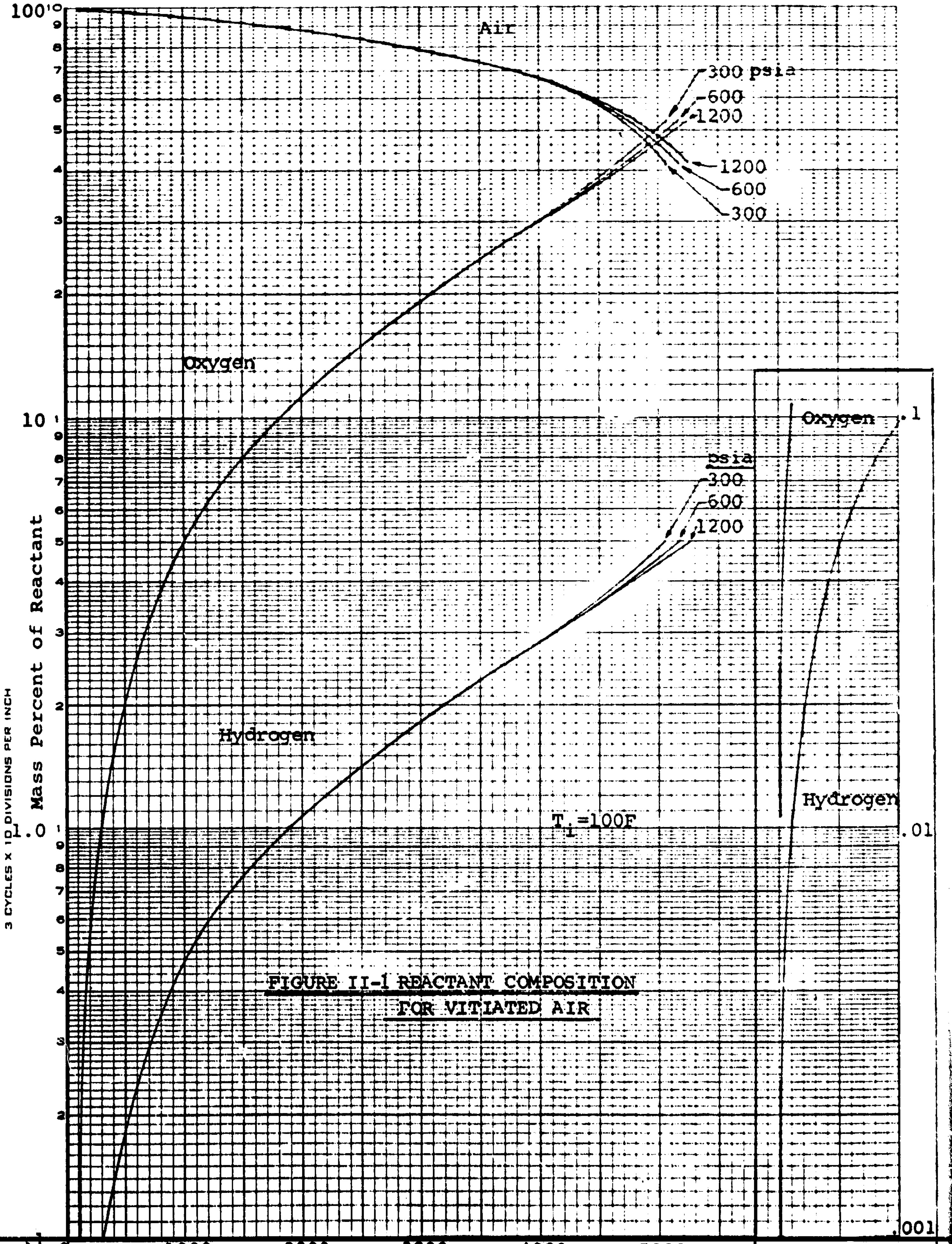

FIGURE II-1 REACTANT COMPOSITION FOR VITIATED AIR

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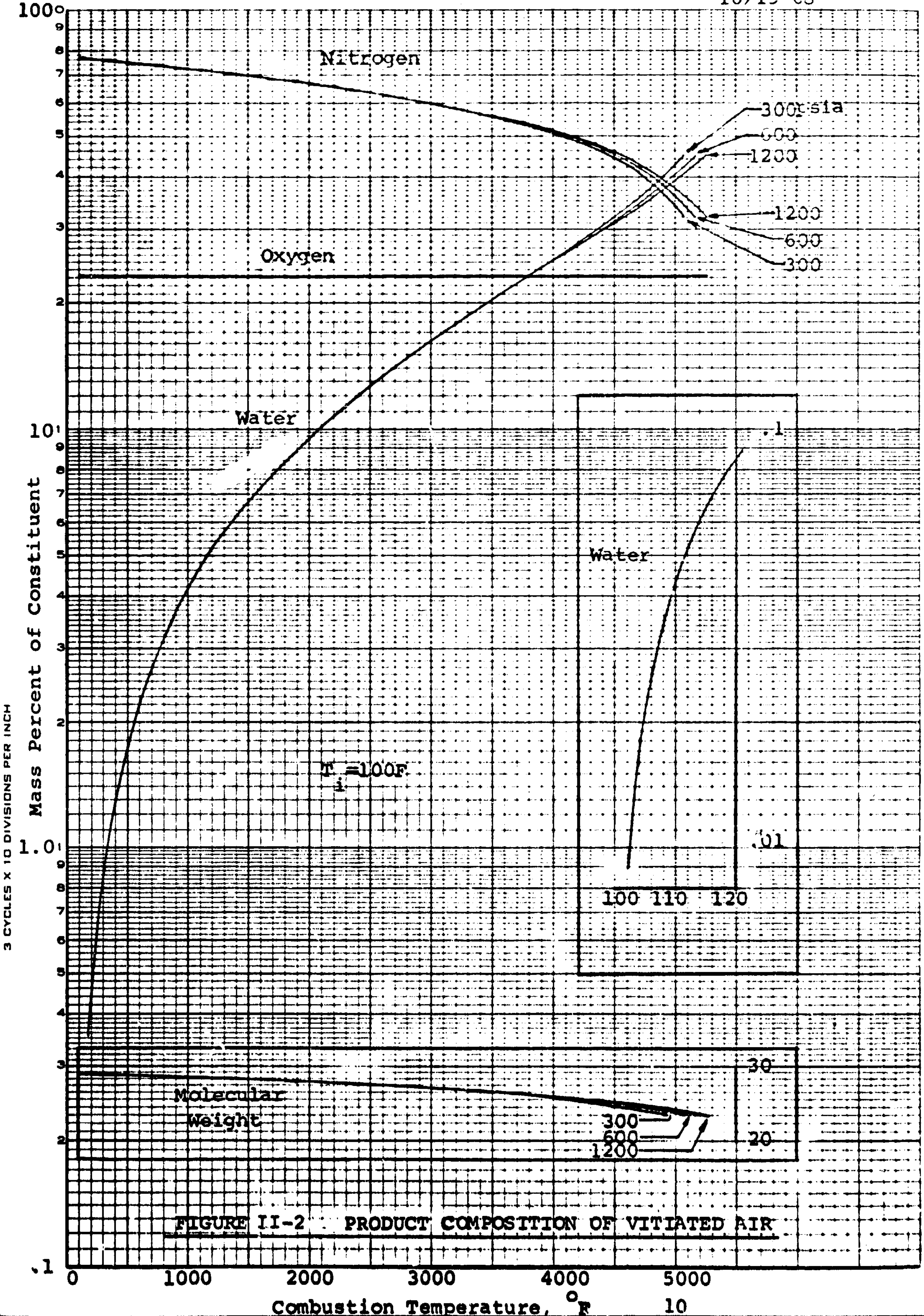


FIGURE II-2 PRODUCT COMPOSITION OF VITIATED AIR

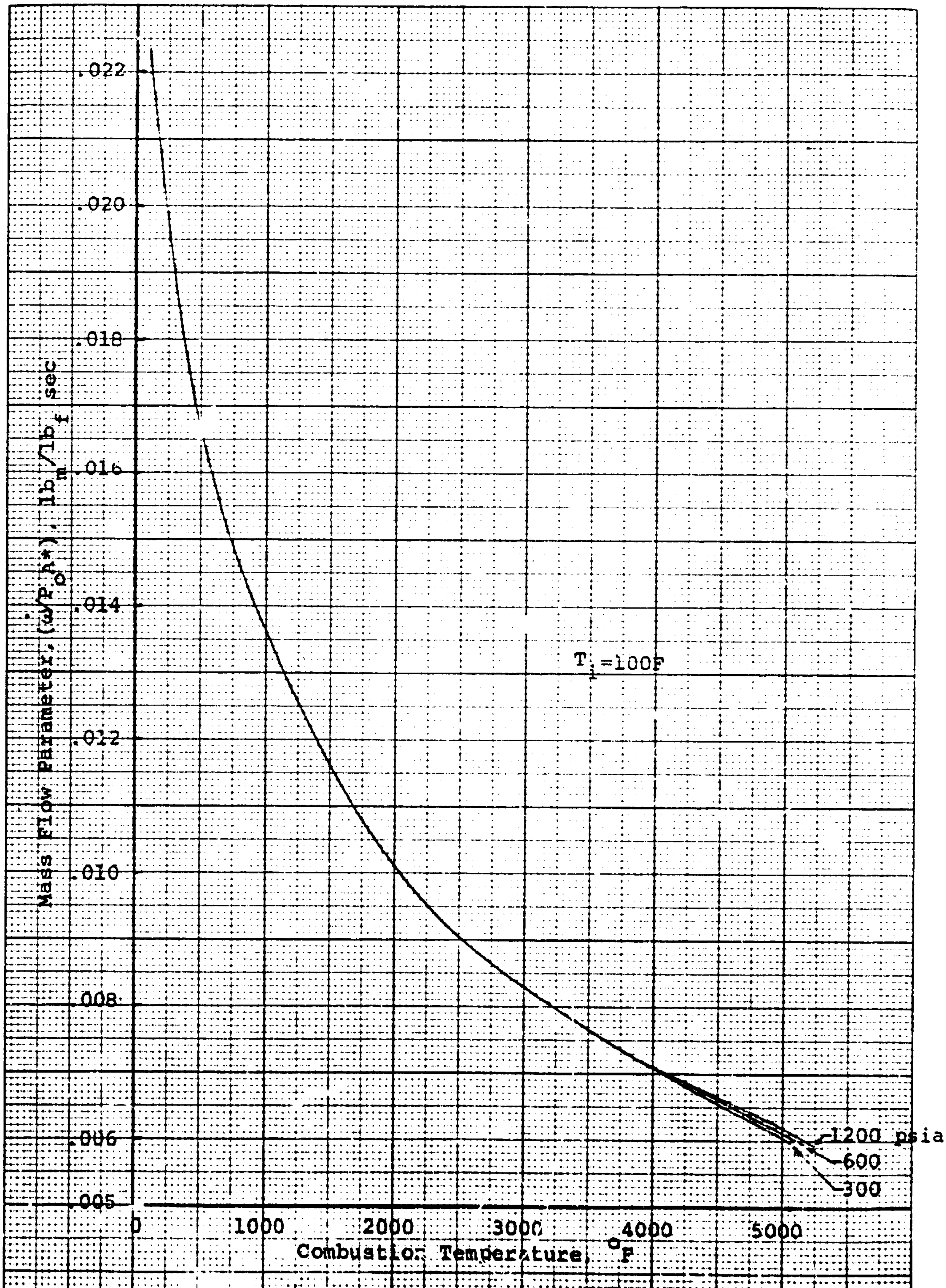


FIGURE II-3 MASS FLOW PARAMETER FOR VITIATED HEATER

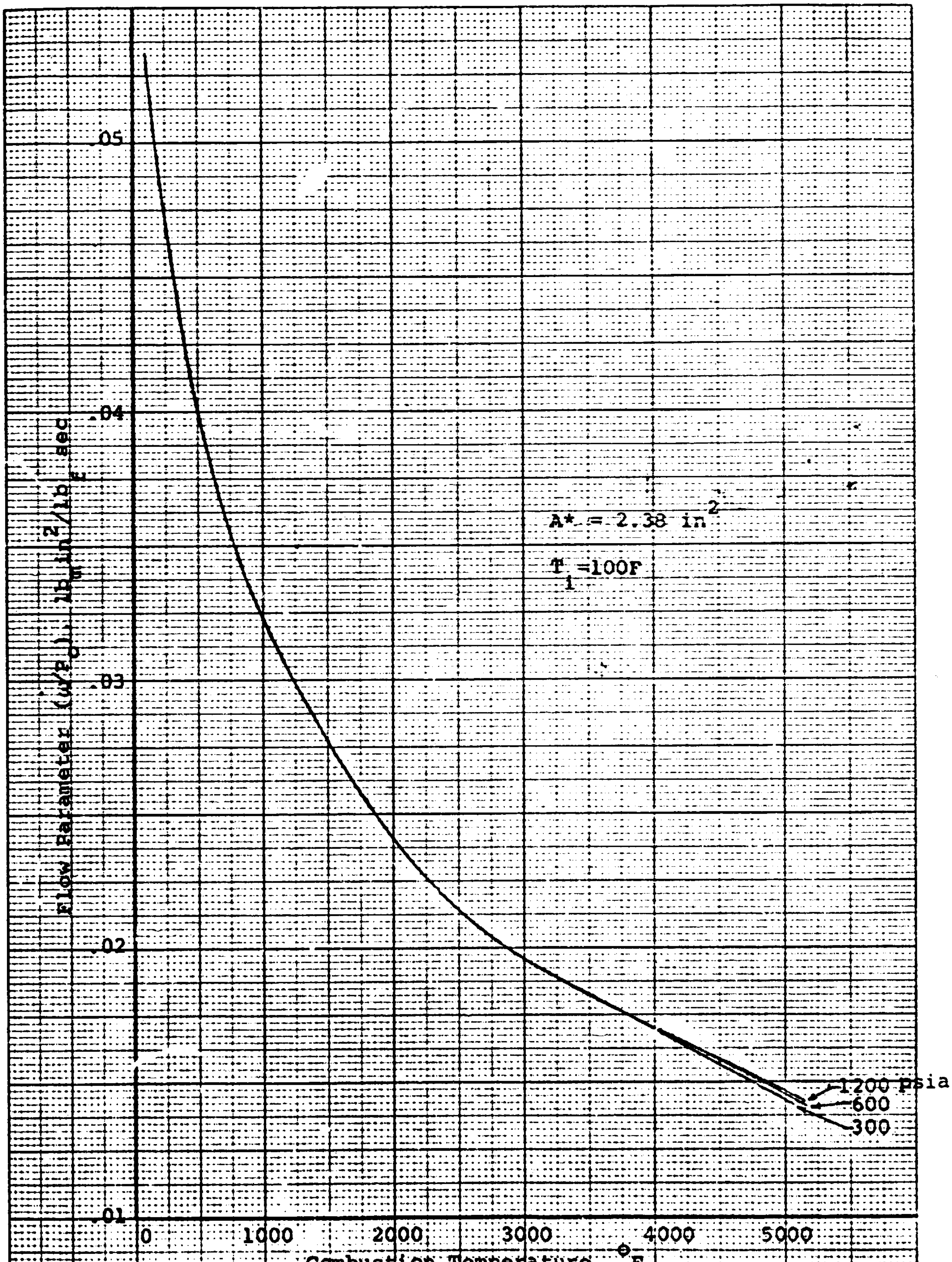


FIGURE II-4 FLOW PARAMETER FOR VITIATED HEATER

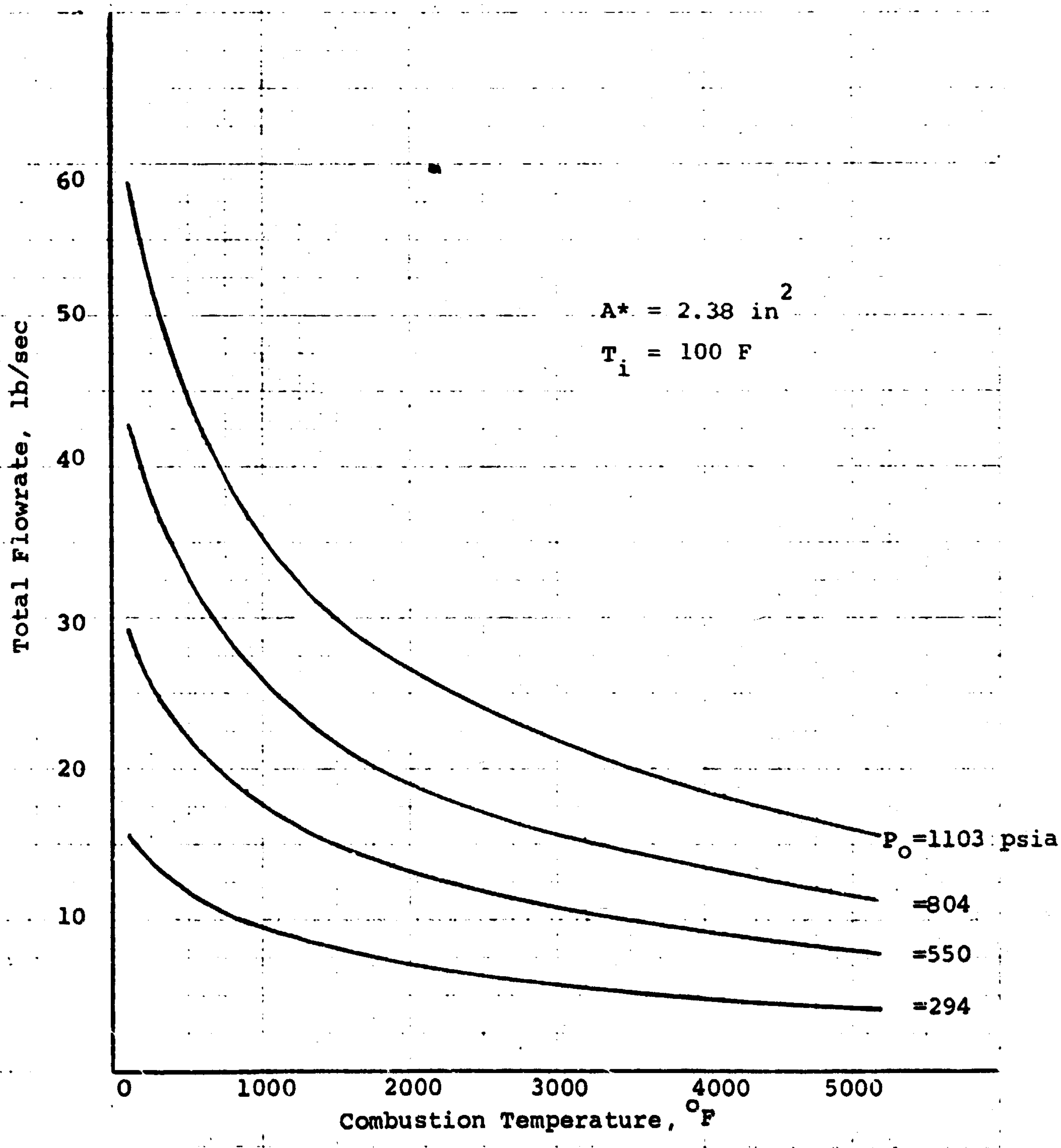


FIGURE II-5 TOTAL FLOWRATE FOR VITIATED HEATER

SECTION III

MIXING ANALYSIS

III. MIXING ANALYSIS

A. CENTRAL AND RADIAL CONFIGURATIONS

In performing a mixing analysis for optimizing the heater design there are initially two chief governing criteria. The first is to ensure that the ignitor system will cause ignition of the hydrogen-oxygen-air flows in a short distance, and the second is to achieve a uniform hot flow in as short a heater length as possible. Since the heater will be started up in a gradual, transient manner there is no problem envolved in igniting all of the fuel. Hence, we are solely concerned with minimizing the length required to obtain a desired degree of uniformity in the flow field.

The basic mixing analysis employing the boundary layer parabolic equations is described in detail in Refs.III-1,2,3, and has been applied to a wide variety of problems at GASL. Briefly, the Conservation equations for a steady-state axisymmetric flow system with equilibrium burning may be written:

Continuity:

$$\frac{\partial(\rho u y)}{\partial x} + \frac{\partial(\rho v y)}{\partial y} = 0$$

Momentum:

$$\rho u \frac{\partial u}{\partial x} + \rho v \frac{\partial u}{\partial y} = -\frac{dp}{dx} + \frac{1}{y}\,\frac{\partial\left(\mu y \frac{\partial u}{\partial y}\right)}{\partial y}$$

Energy:

$$\rho u \frac{\partial H}{\partial x} + \rho v \frac{\partial H}{\partial y} =$$

$$\frac{1}{y}\frac{\partial}{\partial y}\left\{\mu y\left[\frac{1}{Pr}\frac{\partial H}{\partial y} + \left(1-\frac{1}{Pr}\right)\frac{\partial\left(\frac{u^2}{2}\right)}{\partial y} + \sum_{i=1}^{k} h^i \left(\frac{1}{Sc}-\frac{1}{Pr}\right)\frac{\partial \alpha^i}{\partial y}\right]\right\}$$

Diffusion:

(a) elements: $$\rho u \frac{\partial \tilde{\alpha}^j}{\partial x} + \rho v \frac{\partial \tilde{\alpha}^j}{\partial y} = \frac{1}{y}\frac{\partial}{\partial y}\left(\frac{\mu y}{Sc}\frac{\partial \tilde{\alpha}^j}{\partial y}\right)$$

(b) species: $$\rho u \frac{\partial \alpha^i}{\partial x} + \rho v \frac{\partial \alpha^i}{\partial y} = \dot{w}^i + \frac{1}{y}\frac{\partial}{\partial y}\left(\frac{\mu y}{Sc}\frac{\partial \alpha^i}{\partial y} \right)$$

The von Mises stream function transformations

$$\psi \psi_y = \rho u y \qquad \psi \psi_x = \rho v y$$

are used to transform the governing equations to a streamline coordinate system where the continuity equation is now explicit in the definition of the stream function:

Momentum:

$$\frac{\partial u}{\partial x} = -\frac{1}{\rho u}\frac{dp}{dx} + \frac{1}{\psi}\frac{\partial}{\partial \psi}\left(a \frac{\partial u}{\partial \psi} \right)$$

Energy:

$$\frac{\partial H}{\partial x} = \frac{1}{\psi}\frac{\partial}{\partial \psi}\left\{ \frac{a}{Pr}\left[\frac{\partial H}{\partial \psi} + (Pr - 1)\frac{\partial \left(\frac{u^2}{2} \right)}{\partial \psi} + \sum_{i=1}^{k} h^i (Le - 1)\frac{\partial \alpha^i}{\partial \psi} \right] \right\}$$

Diffusion:

$$\frac{\partial \tilde{\alpha}^j}{\partial x} = \frac{1}{\psi}\frac{\partial}{\partial \psi}\left(\frac{a}{Sc}\frac{\partial \tilde{\alpha}^j}{\partial \psi} \right)$$

where

$$a = \frac{\mu \rho u y^2}{\psi}$$

$$H = \frac{u^2}{2} + h = \frac{u^2}{2} + \sum_{i=1}^{k} h^i \alpha^i$$

and $\alpha^i = f(\tilde{\alpha}^j$ and the chemical system employed)

Using a finite difference technique, as described in Ref.III-1, it is possible to obtain a numerical solution of the above set of equations providing that adequate models for the turbulent eddy viscosity, μ and the chemical burning process are provided.

Semi-empirical turbulent mixing models have been developed for ducted flows as described in Refs. III-3,4. Since we are concerned with the mixing problem, and the residence time in the low speed heater of a typical mass element is far in excess of the time required for hydrogen-air mixtures to burn completely and reach chemical equilibrium, a simple "complete combustion" chemistry model was employed. The difference between the complete combustion chemistry model and equilibrium chemistry is that the possible dissociation of H_2, O_2 and H_2O into H, O, and OH at high temperatures is ignored. Thus, the final calculated temperature level of the flow is somewhat higher than it is in physical reality. However, this does not significantly effect the mixing process as will be shown by a comparison of an equilibrium chemistry calculation with a complete combustion calculation.

The calculations were performed at the extremes of the desired performance conditions of the heater. The initial conditions of each calculation are presented in Table III-1.

The philosophy adopted in performing the analytical calculations was to try to make each step as conservative as possible. Thus the first heater fuel injection configuration to be considered was to have concentric rings of hydrogen, oxygen, and air about the axis. From the viewpoint of mixing, this is the worse possible configuration since the fuel has a minimal surface area in contact with the oxidizer.

Constant area calculations were begun for the above configuration and in all four cases local flow reversal (velocities in the upstream direction) were encountered. This phenomena is partly due to the very low initial velocities in the air stream. One-dimensional calculations showed that there was no possibility of thermal choking occurring in the heater. Hence, the initial flow reversal was a local eddy phenomena.

A boundary layer type analysis cannot be used to analyze a flow field where local flow reversal occurs. However, by specifying that the flow be considered constant pressure instead of constant area, this difficulty was bypassed. Since the flow in the heater is quite subsonic (bulk Mach numbers varied from .01 to.1) the variation in static pressure between the constant area calculations and constant pressure calculations was less than 1% in three of the cases, and a maximum of 5% in case E .

It should also be noted that having local flow reversal enhances the mixing process, and hence makes the calculations even more conservative. As is shown in Figs.III-1 through 10, satisfactory burning and mixing is achieved in a five foot length for high temperature cases D and E. However for the low final temperature cases F and H, as much as twelve feet is needed to get a reasonable degree of uniformity.

Upon obtaining this result, a more realistic configuration was considered. Here, the ring of hydrogen injectors was located at a radial distance of three inches from the axis, with the ring of oxygen injectors immediately above it. Complete combustion calculations were performed for cases D and F, since they represented the best and worse mixing performance, respectively, for the central configuration.

As is shown in Fig.III-11 through 15, satisfactory uniformity is obtained for case F using the radial configuration, and the mixing performance of high temperature case D is not adversely effected by the change from the central to the radial configuration.

Thus, the above calculations indicate that it is possible, using the radial configuration of hydrogen and oxygen tubes, to design the heater to provide a satisfactory degree of mixing within a five foot length, over the entire range of desired operating conditions.

The final calculation to be performed involved repeating the radial configuration calculation for case F, using equilibrium chemistry instead of the complete combustion chemistry model. When the results shown in Figs.III-16 and 17 are compared with Figures III-12 & 14 it is clear that the use of the complete combustion chemistry model has been justified, since the more physically realistic equilibrium chemistry model indicates slightly better mixing. Thus, the complete combustion model, as desired, is conservative, and may be safely used for the design of the heater.

One final point to be considered is what the effect would be if it were necessary to increase the velocities (for purposes of improving fuel injection system control and stability) at which the hydrogen and oxygen flows enter the heater.

Zakkay, in Ref. III -5 found experimentally that the potential core length of a jet (defined as the length required for a one percent change in chemical composition on the centerline of the jet) was determined by

$$x_o = k_1 r_{jet} \sqrt{\frac{(\rho u)_{jet}}{(\rho u)_{\substack{outer \\ flow}}}} = k_1 \sqrt{\frac{\pi r_j^2 (\rho u)_j}{\pi (\rho u)_{o.f}}}$$

$$x_o = k_1 \sqrt{\frac{\dot{m}_j}{\pi(\rho u)_{o.f}}} \quad ; \text{ where } k_1 \text{ is constant.}$$

Furthermore, it was also found that the subsequent mixing was a unique function of (x/x_o). Now even though the H_2 and O_2 injection velocities and densities may be modified, neither their mass flow rates nor the density and velocity of the air will be affected. Hence, the potential core length is unaffected, and the mixing process in the duct will not be significantly changed by varying the injection velocities.

B. DISCRETE TUBE INJECTION

Practical engineering design requires that a discrete tube injection pattern be utilized, rather than the radial (annular) configuration studied in the preceding analysis. Calculations have been performed which indicate that injection from discrete tubes, after a short mixing distance, is equivalent to annular injection. These results show that sets of 10 oxidizer and fuel injectors located as close as possible to a radial position of 2.25 inches (from the center) gives the best (10 percent variation about mean temp.) profile.

The analysis consists of scaling a single fuel (H_2) and oxidizer (O_2) central injector where the propellant flow area is enclosed by a radius $r_{(H_2O_2)} = .056$ ft. Case F shows that the edge of the mixing region at $L = .8$ ft is $y_{air} = .425$ ft.

If now we assume 10 "tubes" having the equivalent fuel and oxidizer areas, we get:

$$A_{equiv} = \frac{A_{H_2} + A_{O_2}}{10} = \frac{(6.52 \times 10^{-4}) + (37.5 \times 10^{-4})}{10}$$

$$= 4.4 \times 10^{-4} \text{ ft.}$$

The equivalent radius is:

$$r_{equiv} = \sqrt{\frac{A_{equiv}}{\pi}} = \sqrt{\frac{4.4\times10^{-4}}{\pi}} = \sqrt{1.4\times10^{-2}} = 1.2\times10^{-2}$$

$$= .012 \text{ ft.}$$

By scaling, we have:

$$\frac{y_{air}}{r_{(H_2,O_2)}} = \frac{y_{mix}}{r_{equiv}}$$

where the edge of the mixing region is at

$$y_{mix} = .425 \frac{(.012)}{(.056)} = .091 \text{ ft.}$$

$$= 1.1 \text{ in.}$$

and

$$\frac{L}{r_{(H_2,O_2)}} = \frac{\ell_{mix}}{r_{equiv}}$$

where the length of the mixing region is,

$$\ell_{mix} = .8 \frac{(.012)}{(.056} = .171 \text{ ft.}$$

$$= 2.06 \text{ in.}$$

As a typical configuration, let's assume the discrete injector tubes are located at a radial position, r_{tubes} = 2.5 inches from the center. Then the distance between tubes is approximately:

$$C = \frac{2\pi(r_{tubes})}{10} = \frac{2(3.14)(2.5)}{10}$$

$$= 1.57 \text{ in.}$$

The interaction radius, obviously is at

$$r_{interaction} = \frac{C}{2} = \frac{1.57}{2}$$

$$= .78$$

In this interaction radius, the interactionlength is

$$\ell_{interaction} = \ell_{mix} \frac{(r_{interaction})}{(\gamma_{mix})}$$

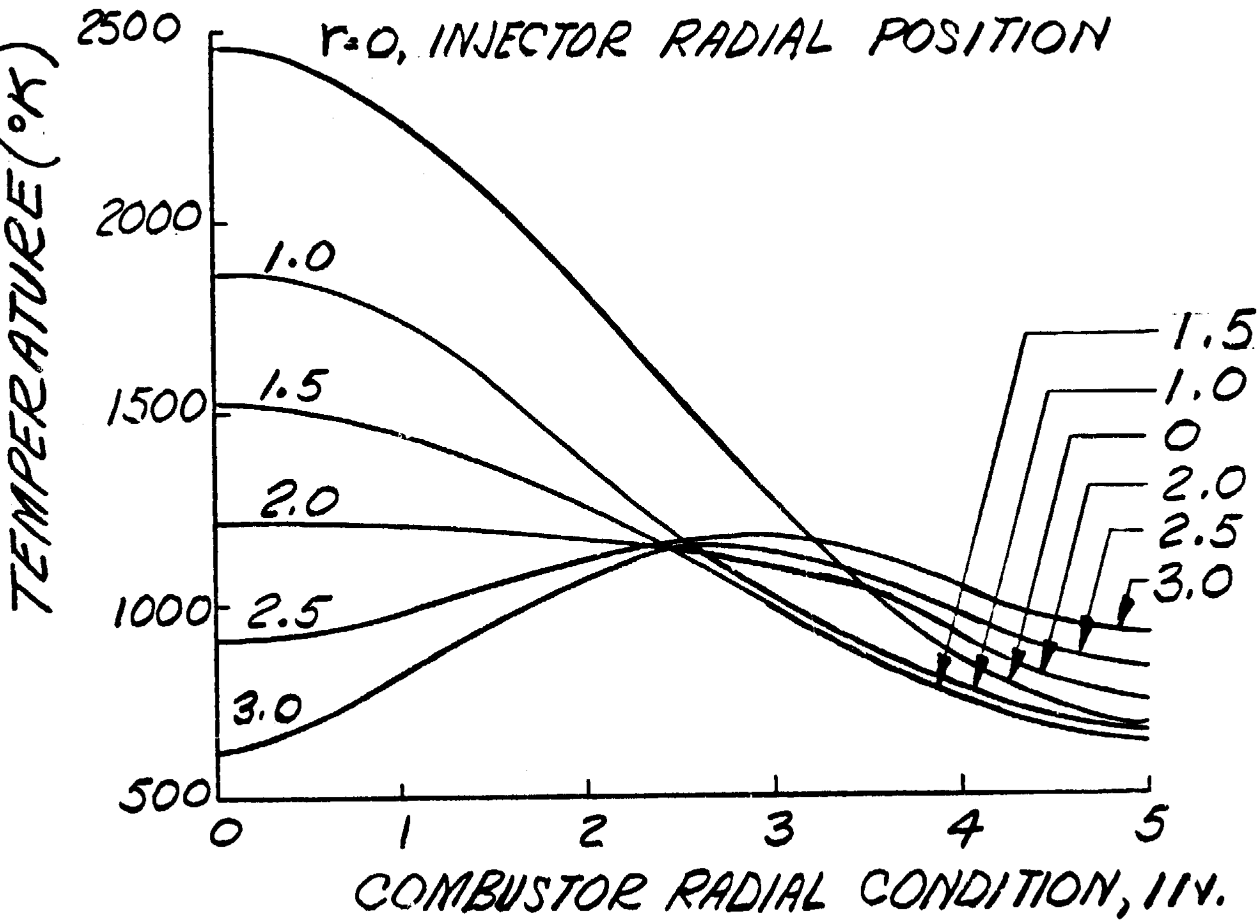
$$= 2.06 \frac{(.78)}{(1.1)}$$

$$= 1.46 \text{ in.}$$

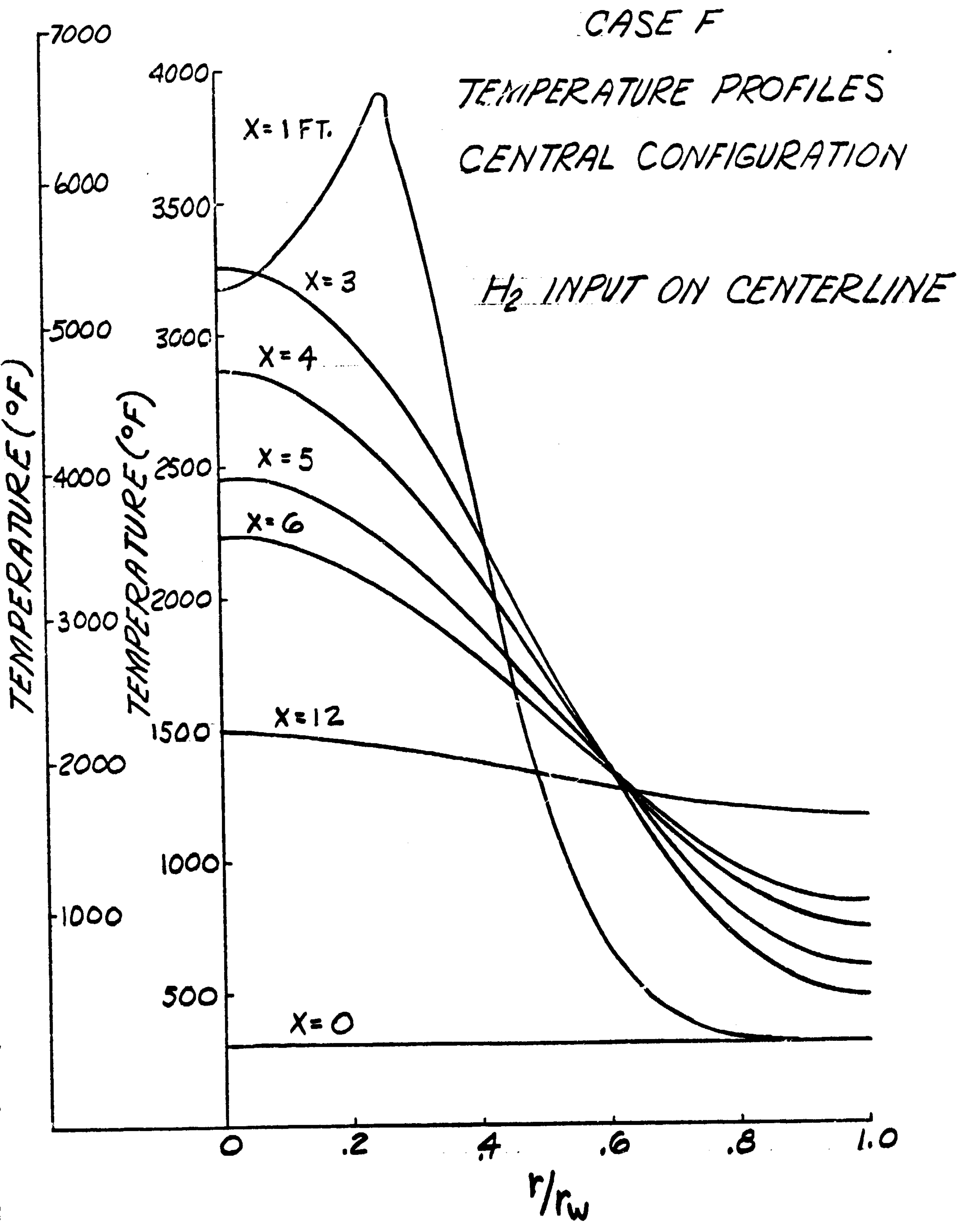
Because interaction takes place in such a short length, it is concluded that a practical representation (i.e. one that can be treated analytically) is the annulus model. Further since condition F is the worst mixing case, a design utilizing 10 sets of injectors is a conservative one, since better mixing will be achieved for all other conditions.

Similar results may be obtained by gross scaling. For a single central fuel, oxidizer injector, we conclude that a length of 15 ft is required (for condition F) to achieve uniform mixing.

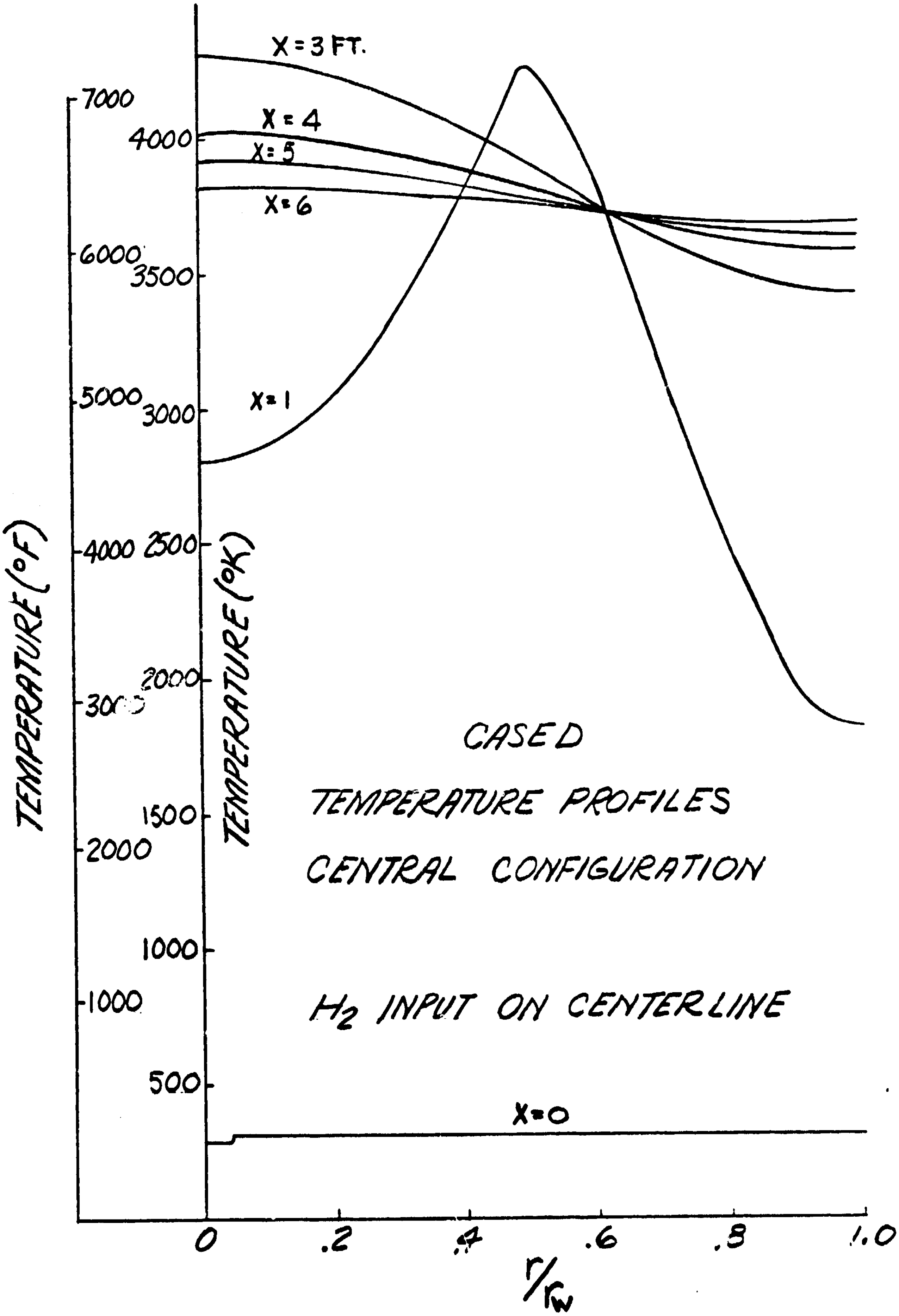
CONDITION F
L = 5 FT.



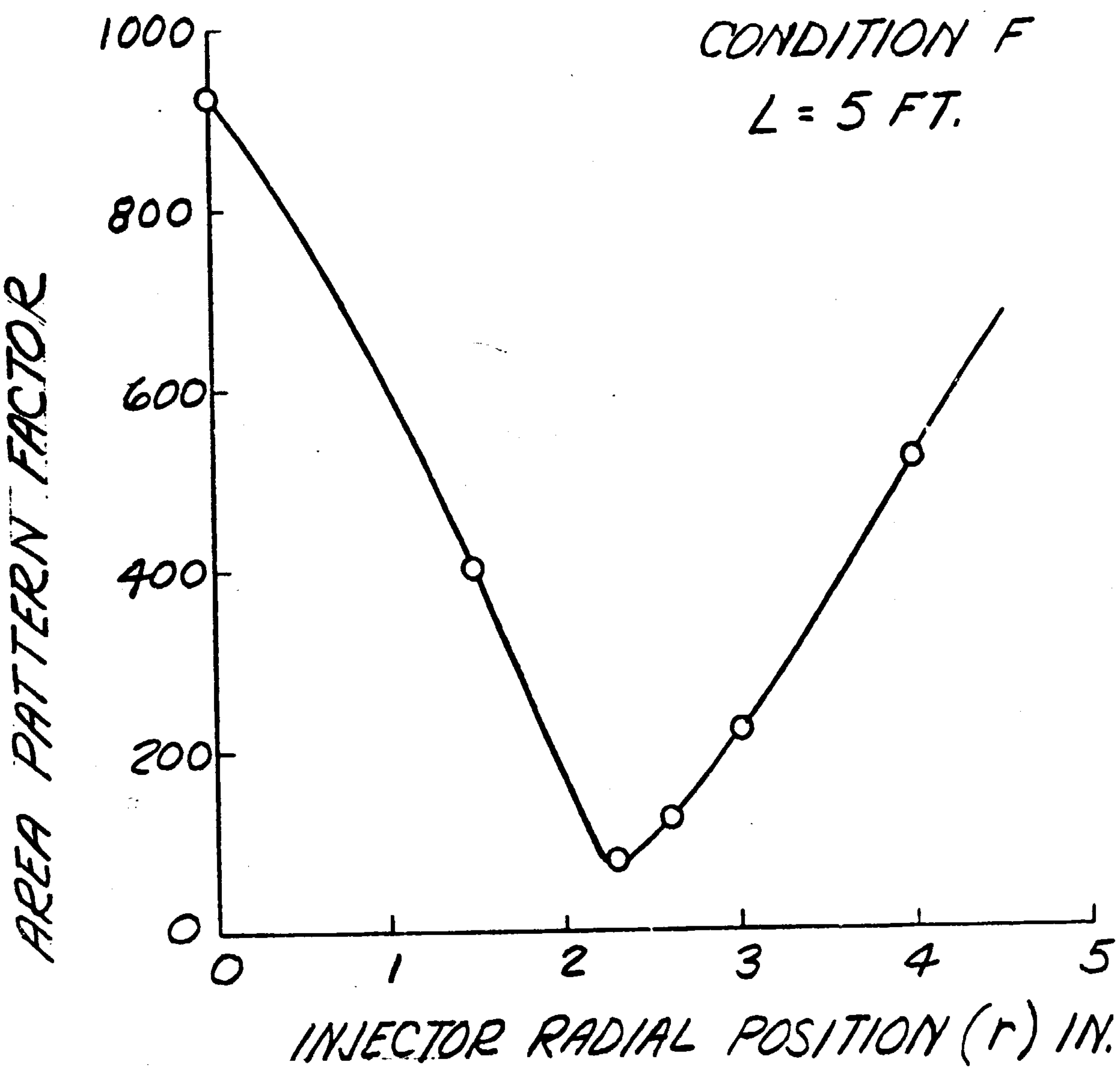
TEMPERATURE PROFILE FOR
ANNULAR INJECTION



CASE F
TEMPERATURE PROFILES
CENTRAL CONFIGURATION
H2 INPUT ON CENTERLINE
X=1 FT.
X=3
X=4
X=5
X=6
X=12
X=0
TEMPERATURE (°F)
7000
6000
5000
4000
3000
2000
1000
TEMPERATURE (°F)
4000
3500
3000
2500
2000
1500
1000
500
0
.2
.4
.6
.8
1.0
r/rw



TEMPERATURE (°F)
TEMPERATURE (°K)
7000
6000
5000
4000
3000
2000
1000
4000
3500
3000
2500
2000
1500
1000
500
X = 3 FT.
X = 4
X = 5
X = 6
X = 1
X = 0
0
.2
.4
.6
.8
1.0
r/rw
CASED
TEMPERATURE PROFILES
CENTRAL CONFIGURATION
H2 INPUT ON CENTERLINE



AREA PATTERN FACTOR FOR
ANNULAR INJECTION

Since the practical length of the combustor is 5 ft, we require a scale reduction of 15/5 = 3. This implies that the central injector area (mass flows) is reduced by $(3)^2=9$. Therefore, by gross scaling to achieve the total mass flow for the complete system, we require 9 sets of injector tubes.

The analysis for the annular configuration is used to find the optimum radial location. Temperature profiles for case F at L=5 ft are first plotted for various radial positions of the fuel-oxidizer annuli, as illustrated in Figure III-18. Next, an area pattern factor, $A_{P.F.}$, defined as the area for each curve in Figure III-18, that is measured below its maximum horizontal tangent point. The area pattern factor for each injector radial position is plotted in Figure III-19. It may be observed that the minimum area pattern factor occurs at an injector radial position of about r = 2.25 inches. This radial location results in an approximate temperature profile of 10 percent variation about mean temperature. It is concluded that fuel and oxidizer injection locations should be as close as practical to this radial dimension.

C. VITIATED HEATER OPERATION AT GASL

A compilation has been made of vitiated heater operating data as obtained in a GASL combustor which comes closest to the LRC design.

Figure III-20 shows the general arrangement of GASL Combustor No. 2. The H_2-O_2 injection configuration and the ignition sources are quite similar to the one designed for the LRC heater. The relative locations of the injectors and the ignitor as installed in the heater are shown in Figure III-20. Oxygen is introduced from an annulus surrounding the igniter; hydrogen injection is from twelve tubes located outside the oxygen annulus. The hydrogen injection direction from the twelve tubes is approximately 45° to the combustor horizontal centerline.

Operating data for Combustor No. 2 have been obtained at five conditions. These data points, which correspond to various test programs undertaken with this facility, are indicated in Figure III-21. The relationship of these operating conditions to the operating envelope and the NASA design points is also provided by Figure III-21. Table III-2 gives the pertinent operating data corresponding to the five operating conditions.

Temperature profile data, taken at the test section, with cooling air injection at the throat, are presented in Figures III-22 and III-23. The profile data correspond to operating conditions 2 and 3, respectively, in Table III-2. These profiles were adequate for our purposes since the test article dimensions are obviously always smaller than the test section size, thus permitting locating the model hardware in the core region of the flow field where the temperature is essentially uniform. Profile data for the other operating conditions are not available.

In presenting this additional data, our purpose is to show that a burner design, operating at conditions within the LRC design envelope and similar in design concept to the LRC heater, has been successfully operated. It should be emphasized that point by point comparisons are not possible because of differences in design details and in operating conditions.

References

III.1 Zeiberg, S. L. and Bleich, G. D., "Finite-Difference Calculation of Hypersonic Wakes," AIAA J. 2, No. 9, pp. 1396-1402.

III-2 Edelman, R., "Diffusion Controlled Combustion for Scramjet Applications," GASL TR-569, Dec. 1965.

III-3 Edelman, R. and Fortune, O., "An Analysis of Mixing and Combustion in Ducted Flow," AIAA Paper 68-114, Jan. 1968.

III-4 Siegelman, D. and Fortune, O., "Computer Programs for the Mixing and Combustion of Hydrogen in Air Streams," GASL TR-618, July 1966.

III-5 Zakkay, V. and Krause, E., "Turbulent Transport Properties for Axisymmetric Heterogeneous Mixing," AIAA Preprint 64-99, Jan. 1964.

TABLE III-1 - HEATER OPERATING CONDITIONS

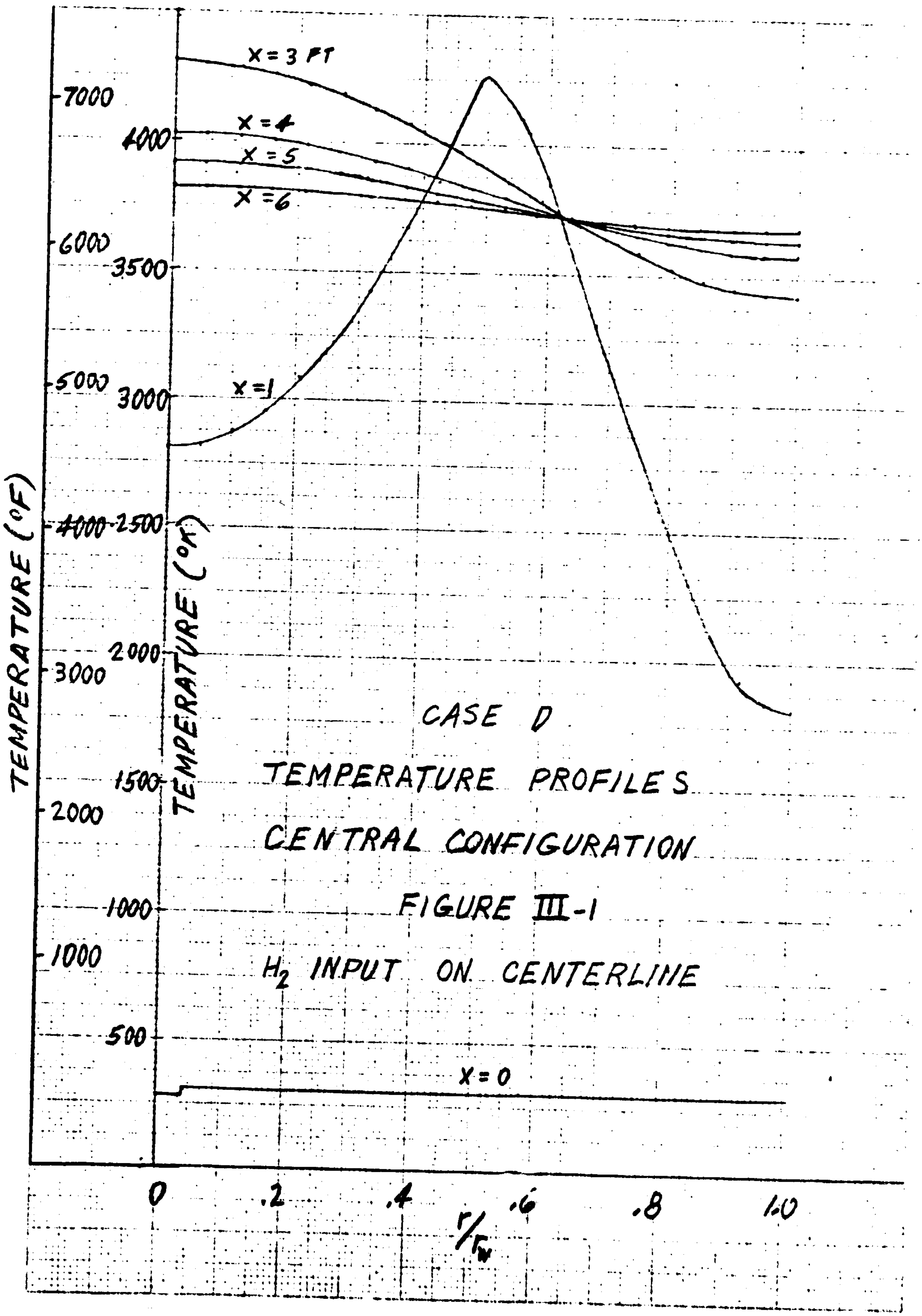
	CASE			
	D	E	F	H
Desired Final Temperature (°F)	5000	5000	1380	1380
Heater Pressure (psia)	1102	294	294	1102
H_2 Flow rate (lbm/sec)	.713	.234	.063	.220
O_2 Flow rate (lbm/sec)	7.96	2.56	.66	2.30
Air Flow rate (lbm/sec)	6.81	1.71	8.28	29.0
H_2 Velocity (ft/sec)	2720	3330	978	909
O_2 Velocity (ft/sec)	354	415	113	108
Air Velocity (ft/sec)	2.37	2.22	10.76	10.07

TABLE III-2

PREVIOUS OPERATING CONDITIONS FOR GASL HIGH TEMPERATURE BURNER

(COMBUSTOR II)

No.	P_O psia	T_O ^oR	w_{Air} lb/sec	$\dot{w}_{O_2}$ lb/sec	$\dot{w}_{H_2}$ lb/sec	V_{Air} (Low Temp. Flow) ft/sec	$\dot{m}$ Throat Cooling lbs/sec
1	400	2200	14	1.4	.13	27.50	None
2	685	2790	20.1	3.200	.306	29.0	2.25
3	1065	2985	4.08	.74	.069	4.8	1.9
4	1000	3700	3.70	1.06	.098	4.7	2.0
5	790	2900	5.00	.88	.081	5.4	2.20



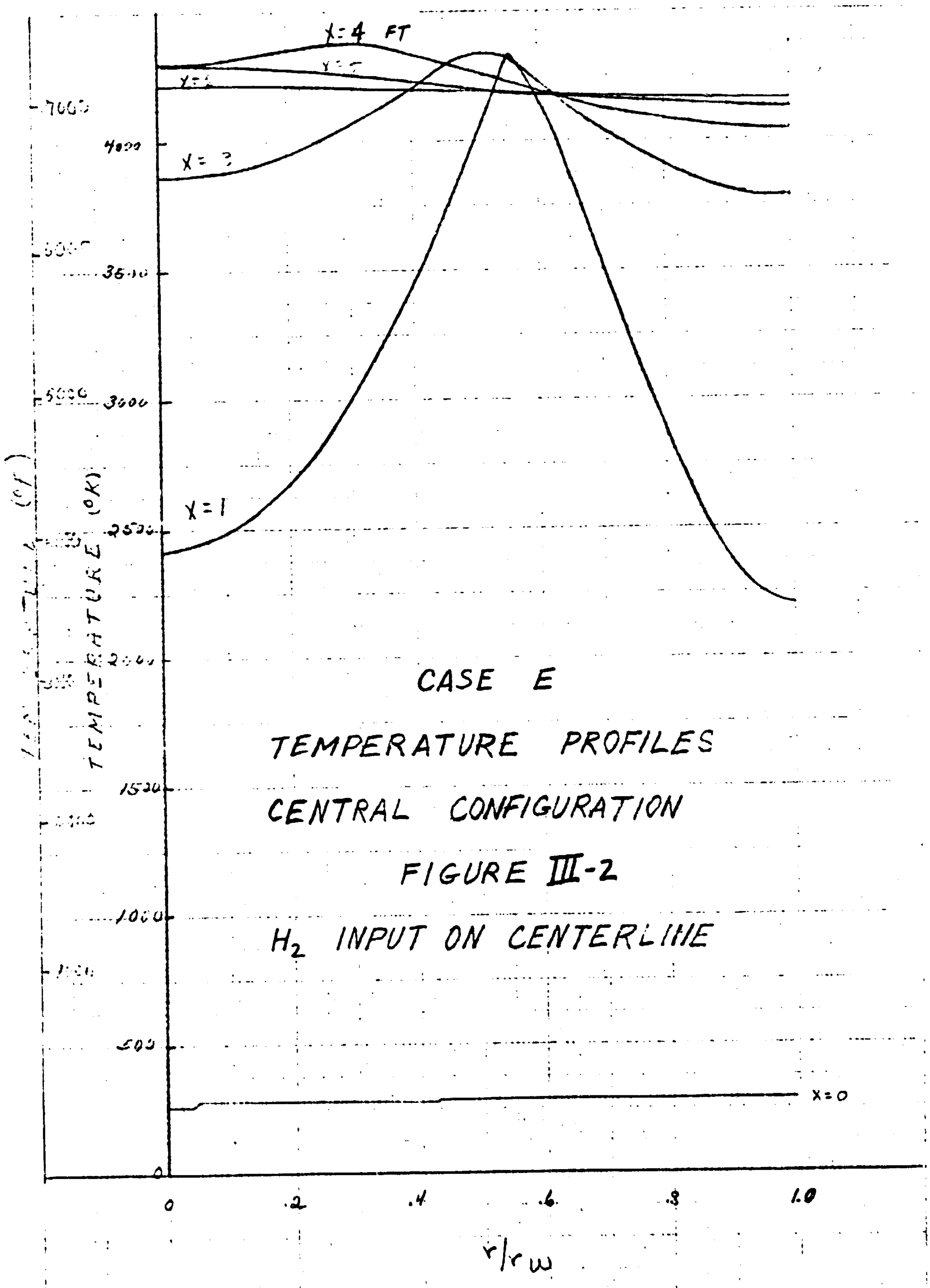
CASE D

TEMPERATURE PROFILES

CENTRAL CONFIGURATION

FIGURE III-1

H_2 INPUT ON CENTERLINE



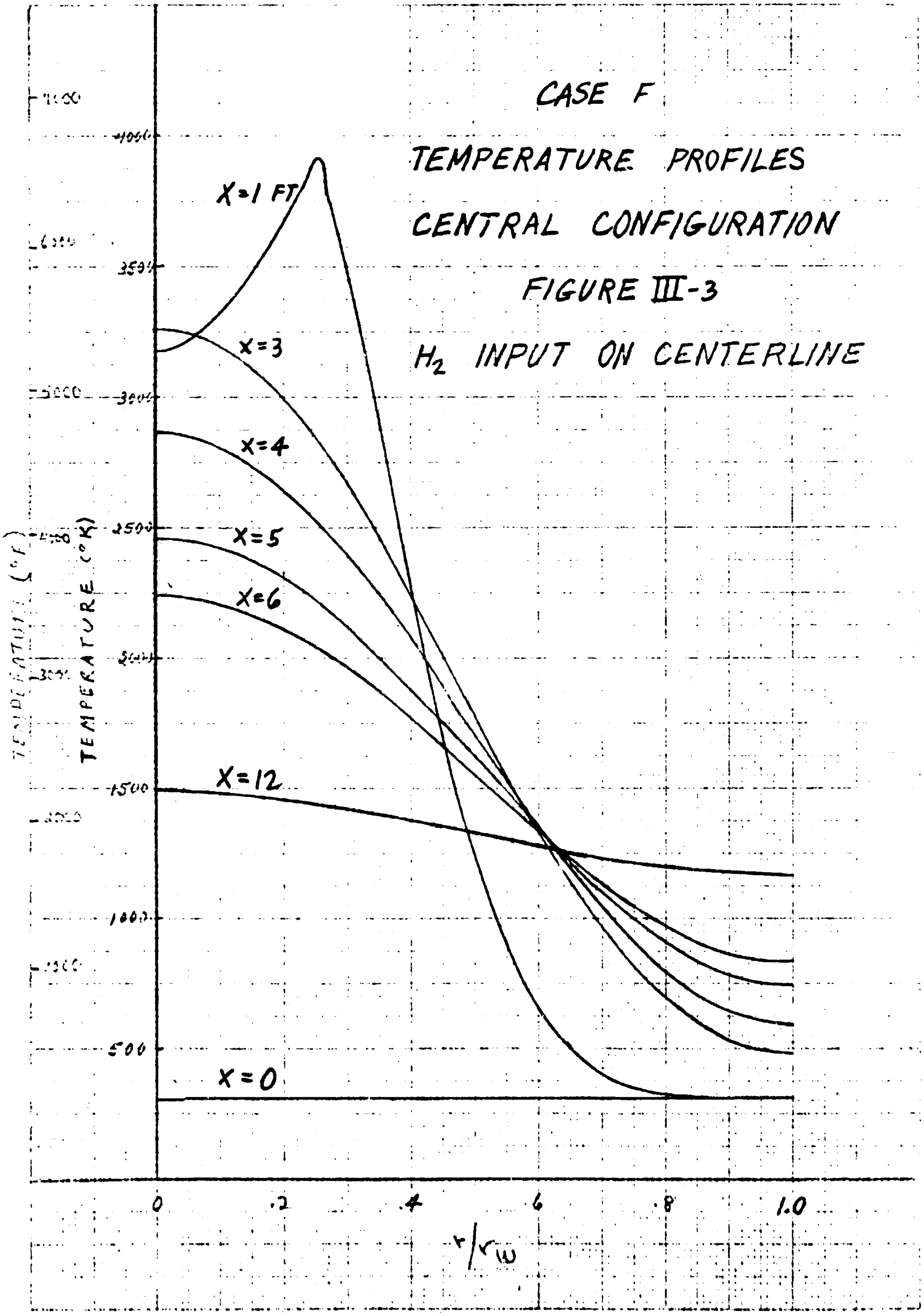
CASE E

TEMPERATURE PROFILES

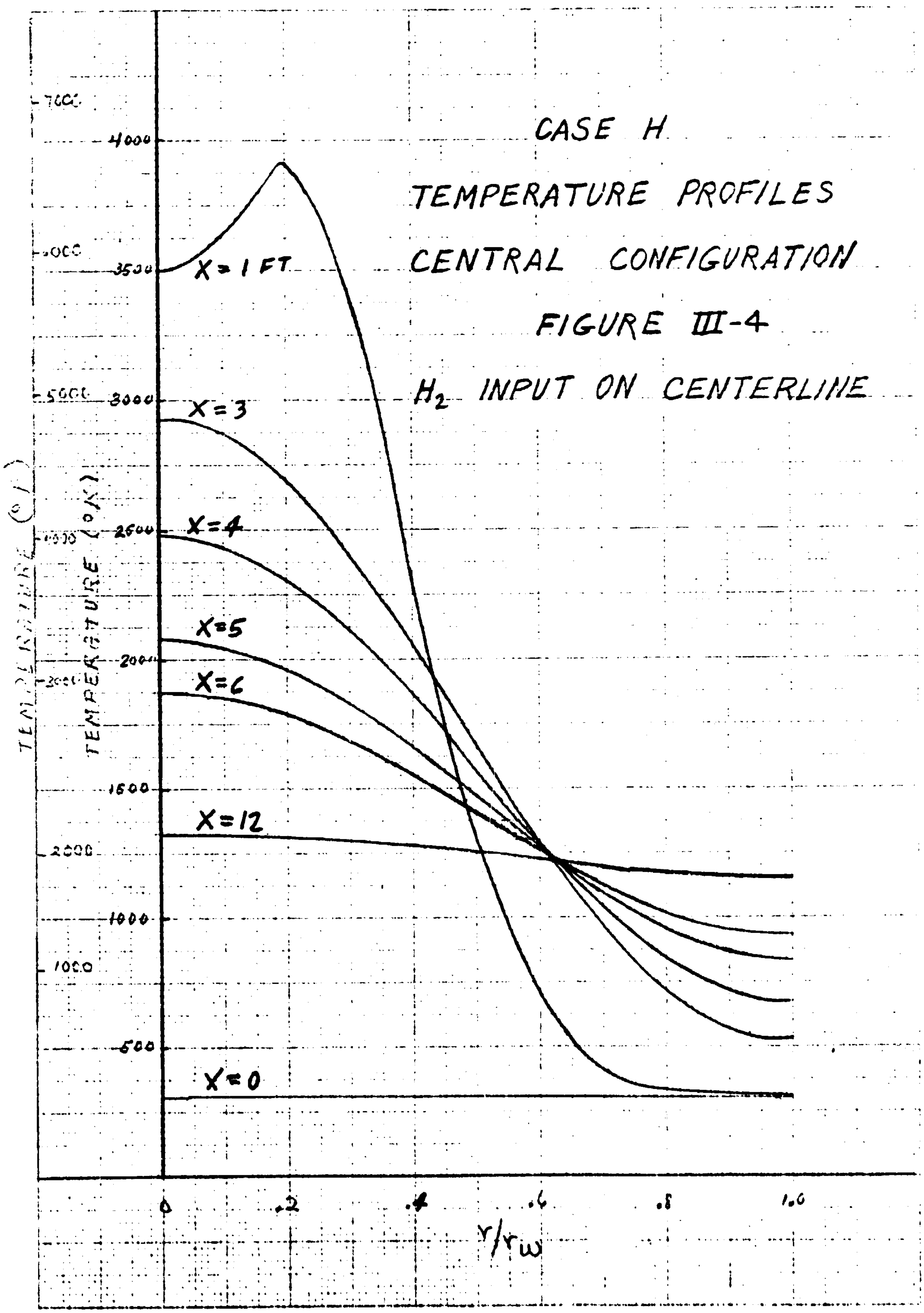
CENTRAL CONFIGURATION

FIGURE III-2

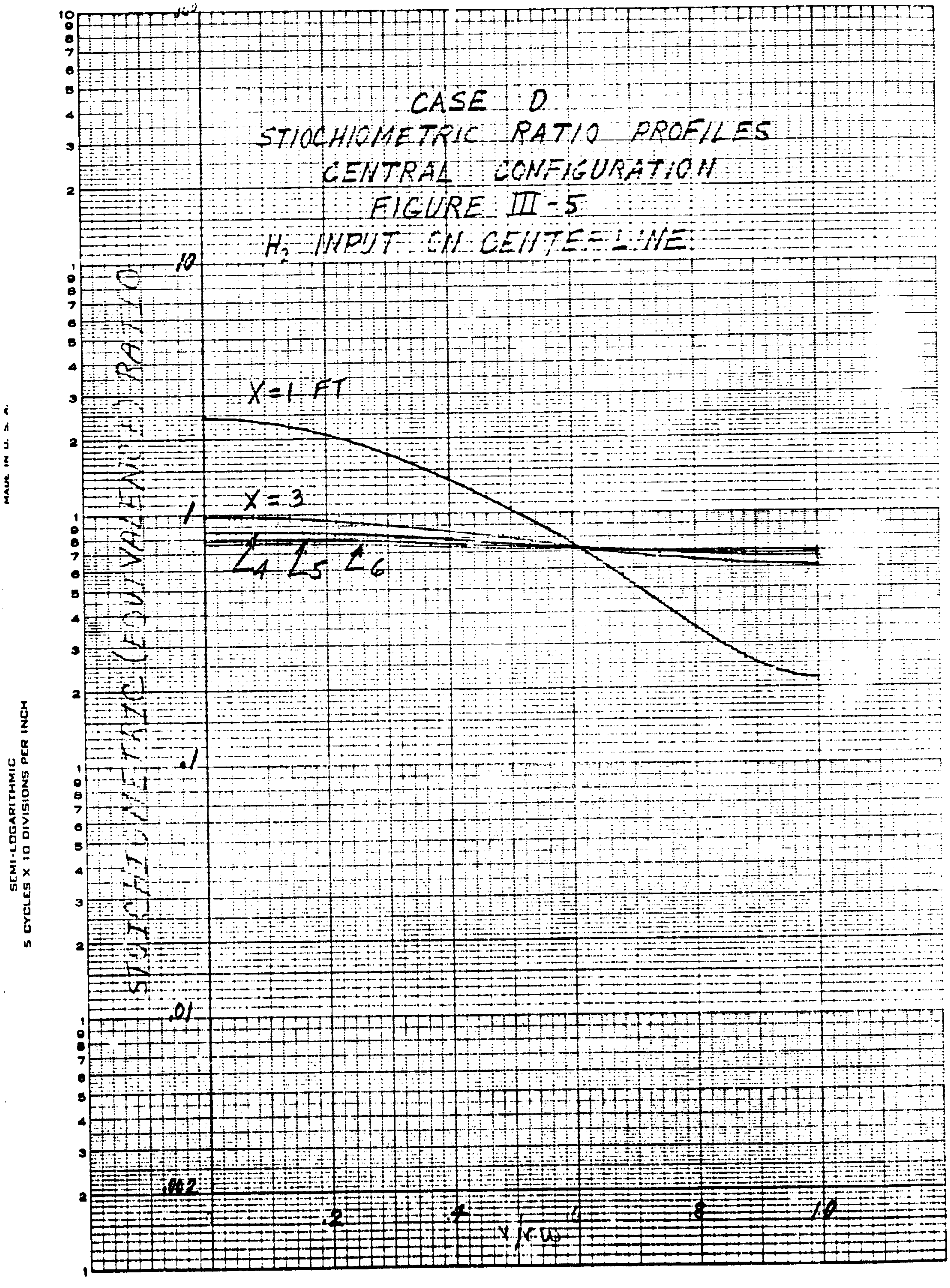
H_2 INPUT ON CENTERLINE

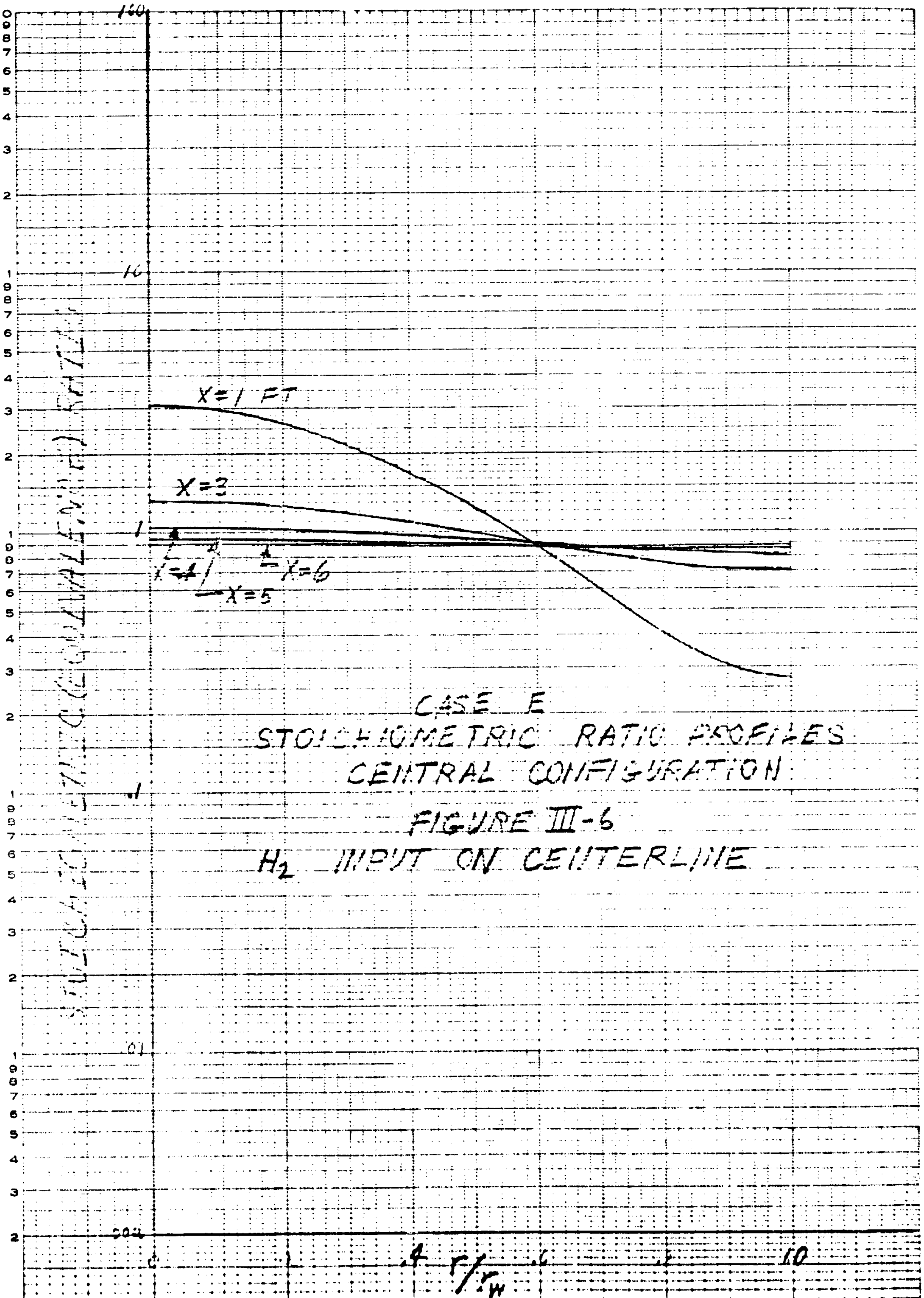


CASE F
TEMPERATURE PROFILES
CENTRAL CONFIGURATION
FIGURE III-3
H2 INPUT ON CENTERLINE
X=1 FT
X=3
X=4
X=5
X=6
X=12
X=0
TEMPERATURE (°K)
4000
3500
3000
2500
2000
1500
1000
500
0
.2
.4
.6
.8
1.0
r/rw

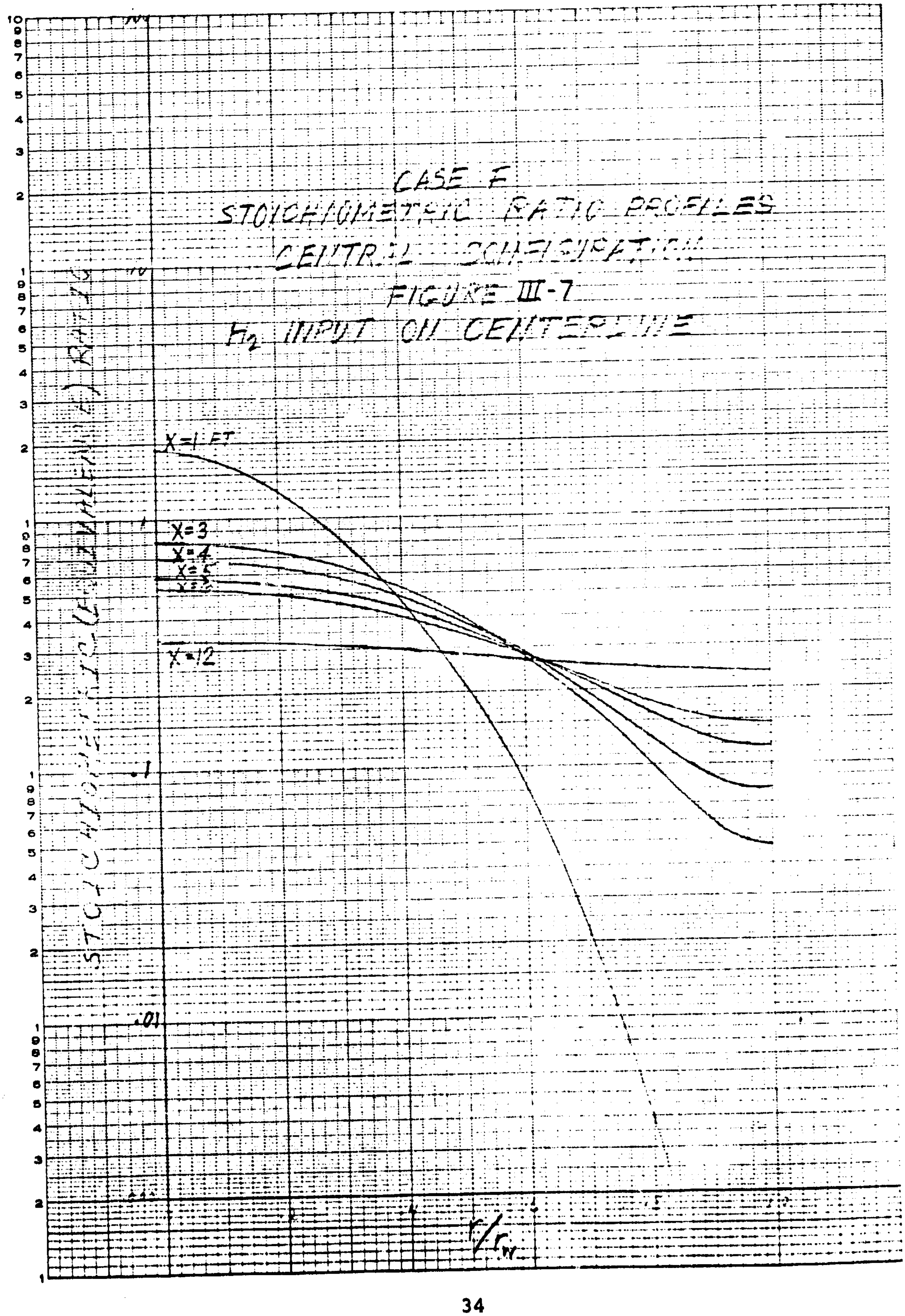


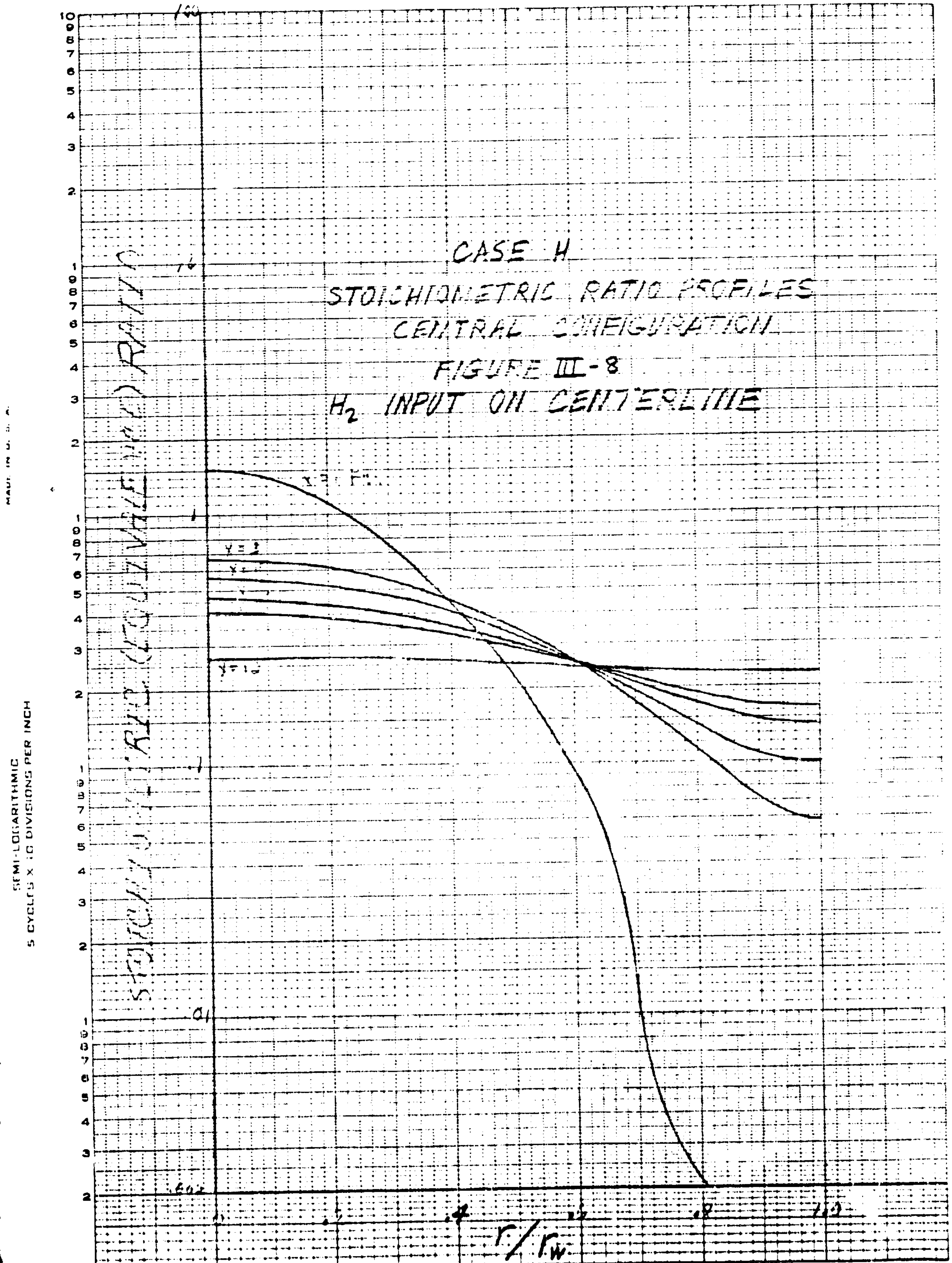
CASE H
TEMPERATURE PROFILES
CENTRAL CONFIGURATION
FIGURE III-4
H_2 INPUT ON CENTERLINE
X = 1 FT
X = 3
X = 4
X = 5
X = 6
X = 12
X = 0
TEMPERATURE (°K)
TEMPERATURE (°F)
4000
3500
3000
2500
2000
1500
1000
500
7000
6000
5000
4000
3000
2000
1000
0
.2
.4
.6
.8
1.0
r/rw

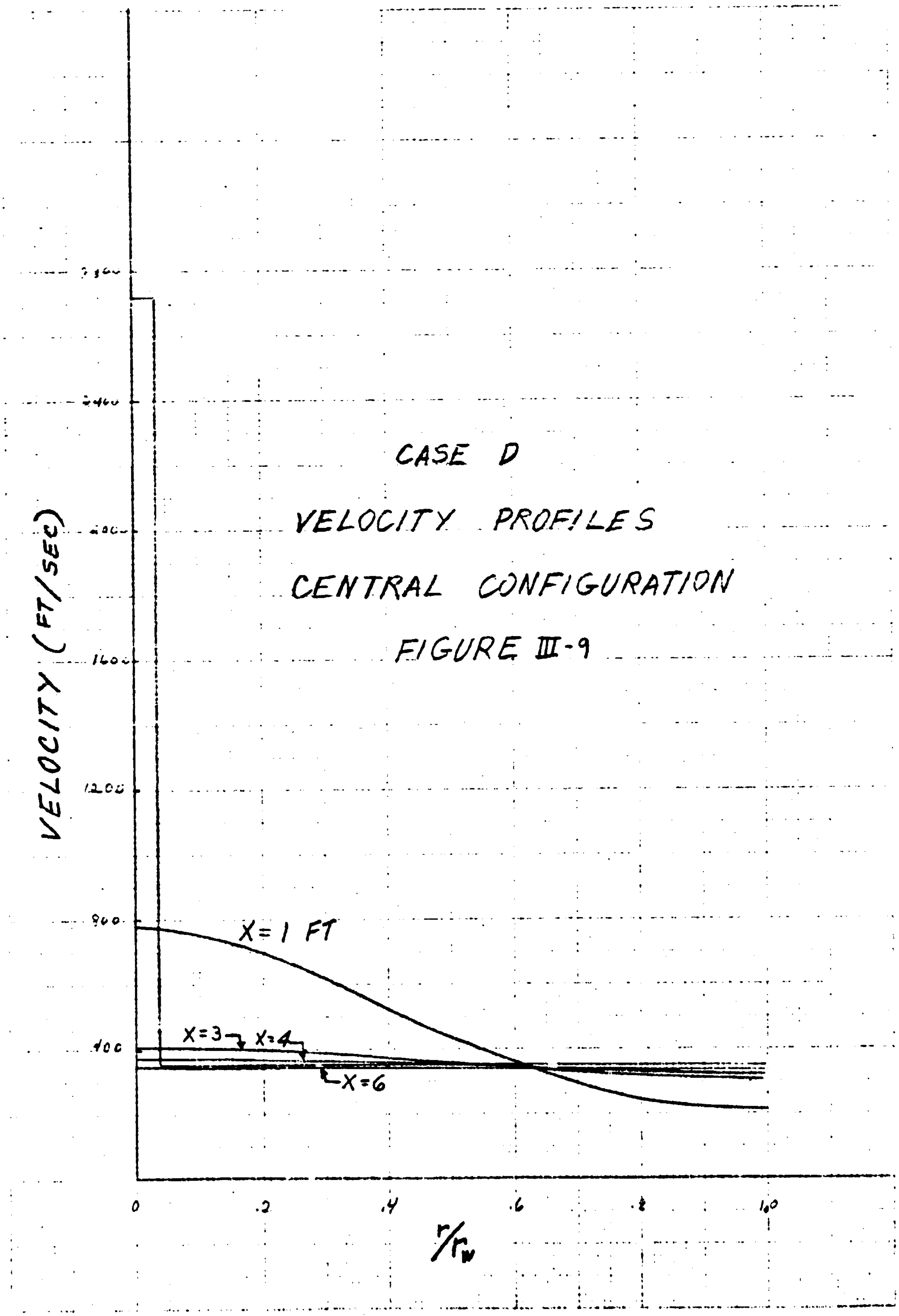

CASE D
STIOCHIOMETRIC RATIO PROFILES
CENTRAL CONFIGURATION
FIGURE III-5
H2 INPUT ON CENTERLINE
X=1 FT
X=3
4 5 6
10
1
.1
.01
.2
.4
.8
1.0

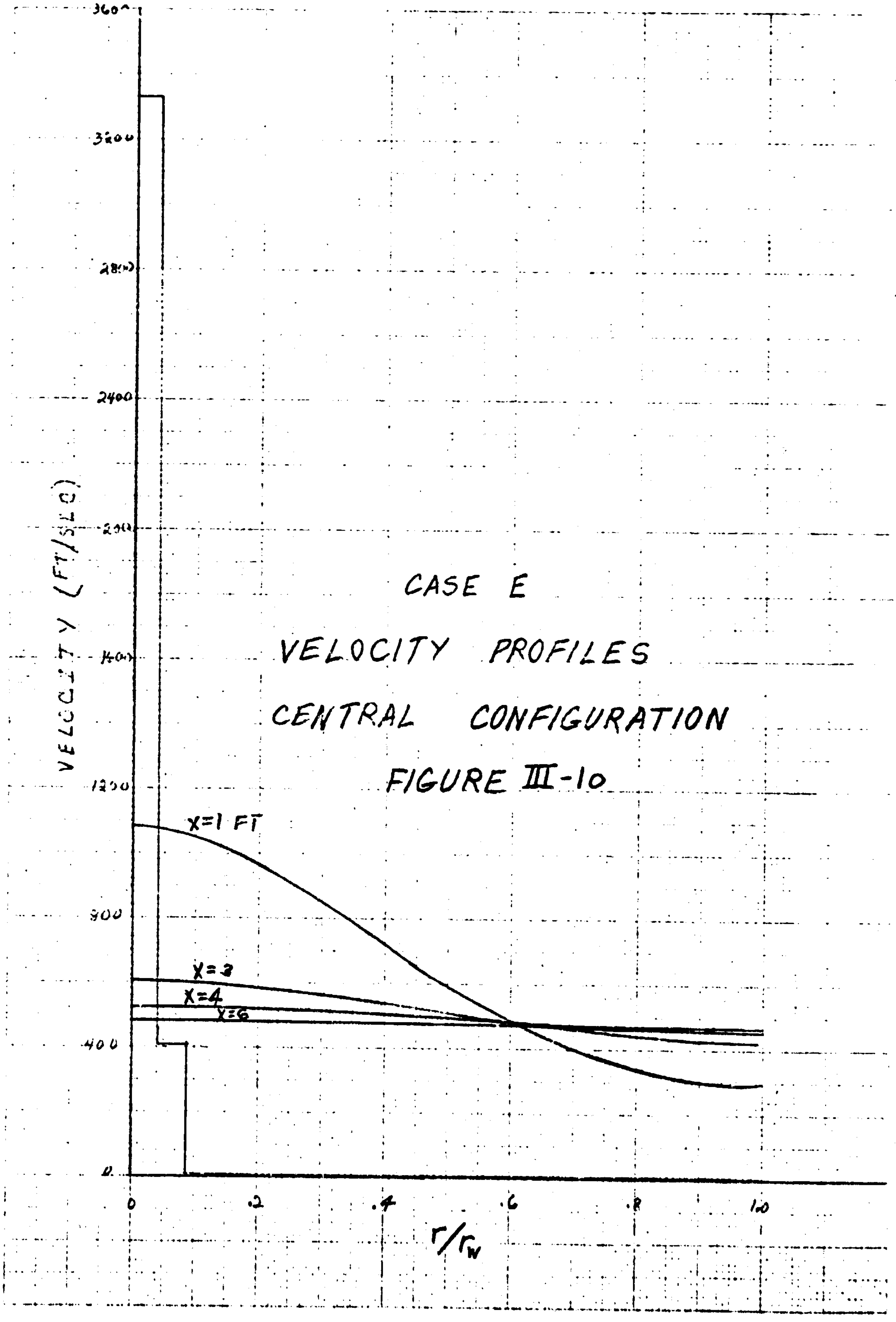


CASE E
STOICHIOMETRIC RATIO PROFILES
CENTRAL CONFIGURATION
FIGURE III-6
H_2 INPUT ON CENTERLINE


CASE F
STOICHIOMETRIC RATIO PROFILES
CENTRAL ...
FIGURE III-7
H_2 INPUT ON CENTERLINE
X=1 FT
X=3
X=4
X=5
X=12
r/r_w


CASE H
STOICHIOMETRIC RATIO PROFILES
CENTRAL CONFIGURATION
FIGURE III-8
H_2 INPUT ON CENTERLINE
r/r_w
SEMI-LOGARITHMIC
5 CYCLES X 10 DIVISIONS PER INCH


CASE D
VELOCITY PROFILES
CENTRAL CONFIGURATION
FIGURE III-9
VELOCITY (FT/SEC)
X = 1 FT
X=3
X=4
X=6
400
800
1200
1600
2400
0
.2
.4
.6
1.0
r/rw

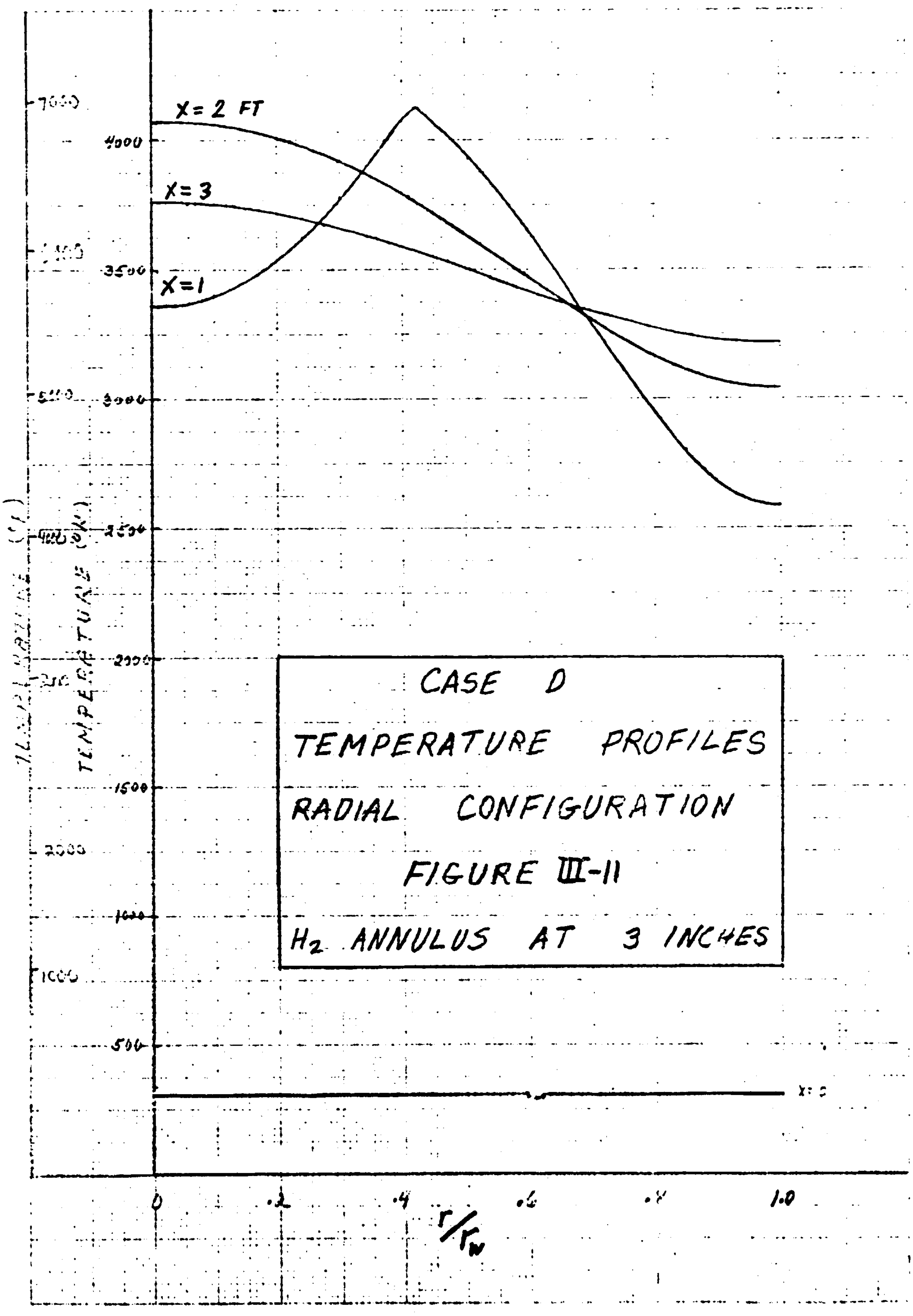


CASE E

VELOCITY PROFILES

CENTRAL CONFIGURATION

FIGURE III-10



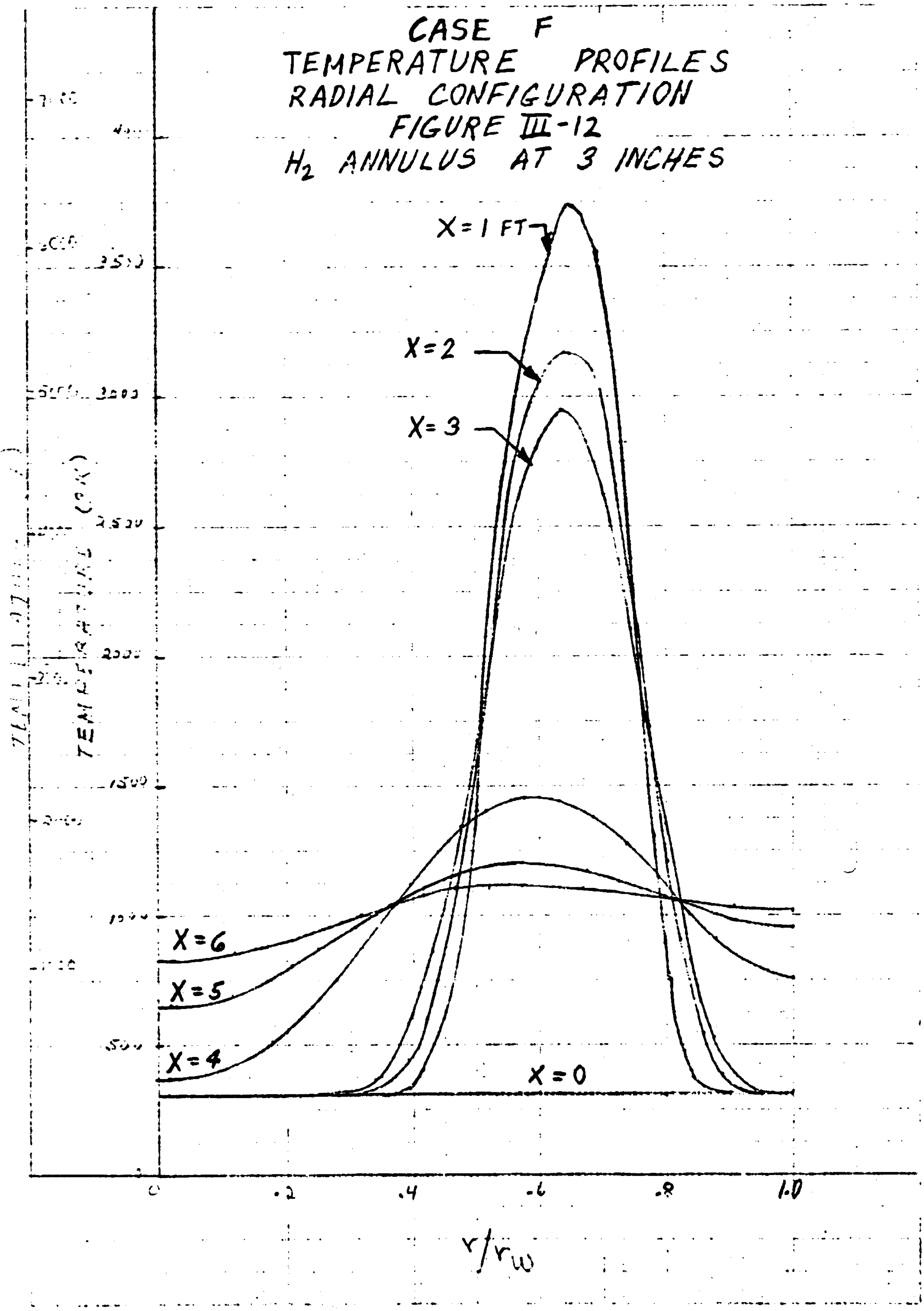
CASE D

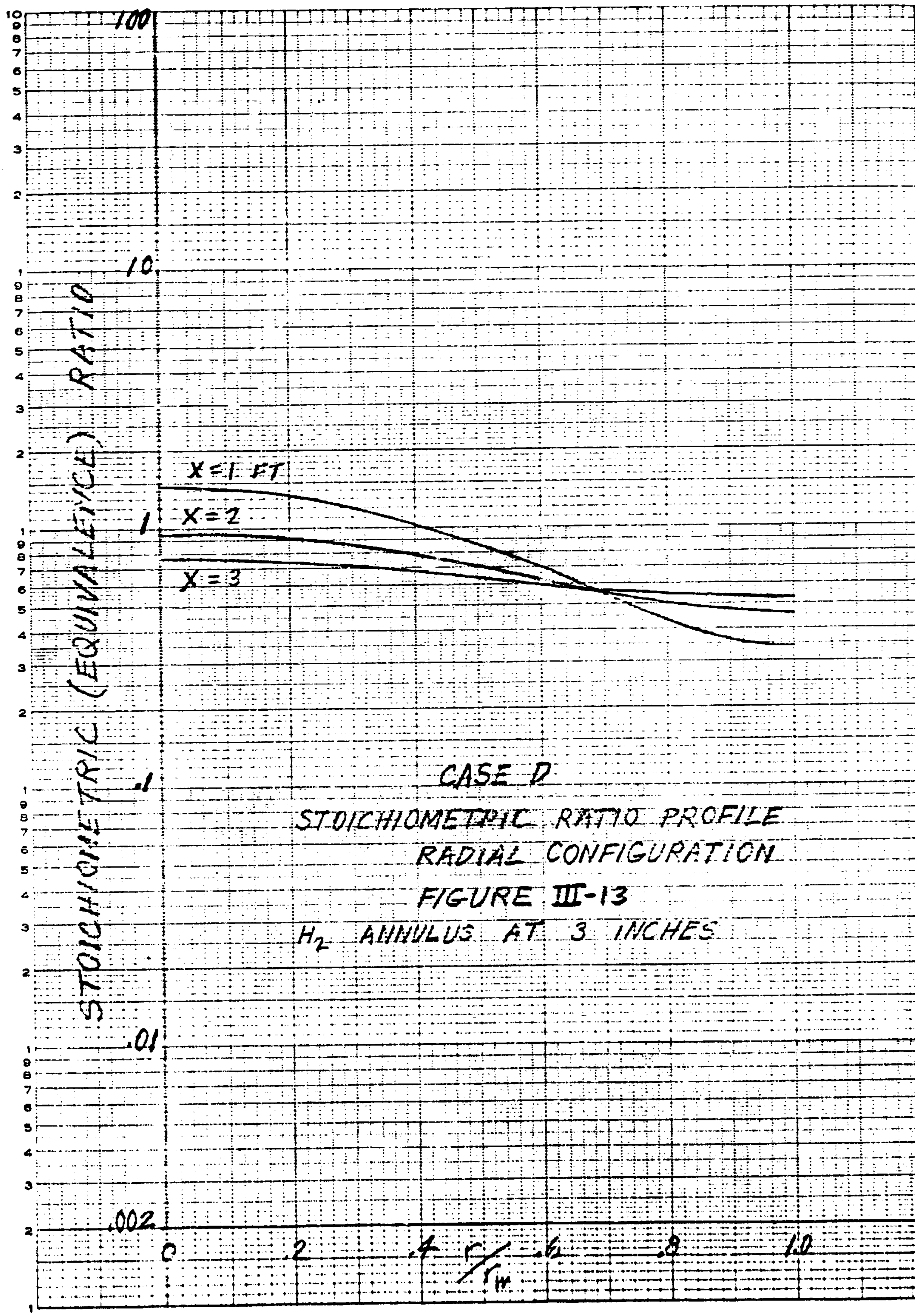
TEMPERATURE PROFILES

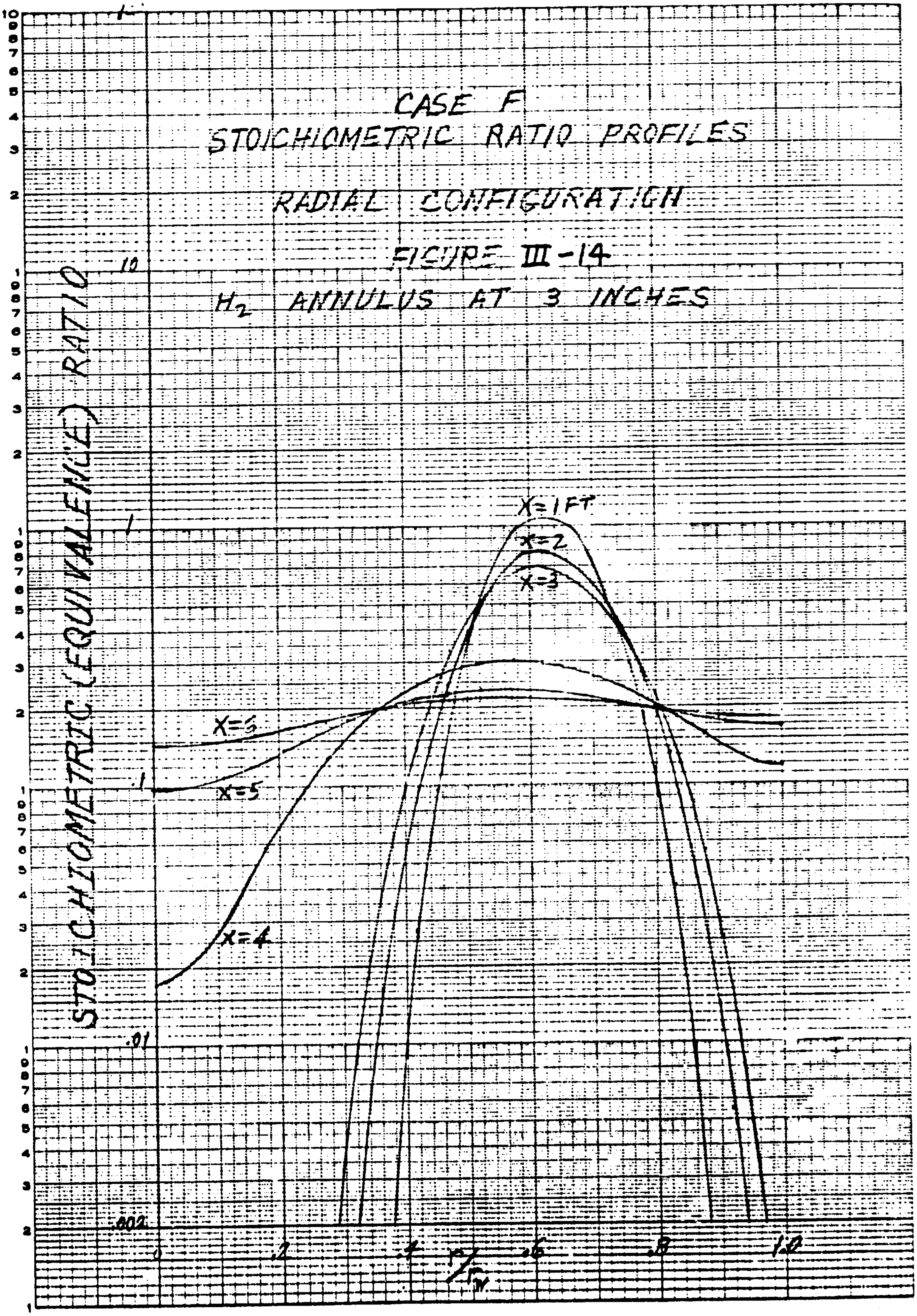
RADIAL CONFIGURATION

FIGURE III-11

H_2 ANNULUS AT 3 INCHES


CASE F
TEMPERATURE PROFILES
RADIAL CONFIGURATION
FIGURE III-12
H_2 ANNULUS AT 3 INCHES
X = 1 FT
X = 2
X = 3
X = 4
X = 5
X = 6
X = 0
TEMPERATURE (°K)
2500
2000
1500
1000
500
0
.2
.4
.6
.8
1.0
r/r_w


100
10
1
.1
.01
.002
STOICHIOMETRIC (EQUIVALENCE) RATIO
X = 1 FT
X = 2
X = 3
CASE D
STOICHIOMETRIC RATIO PROFILE
RADIAL CONFIGURATION
FIGURE III-13
H2 ANNULUS AT 3 INCHES
0
.2
.4
r/rw
.6
.8
1.0

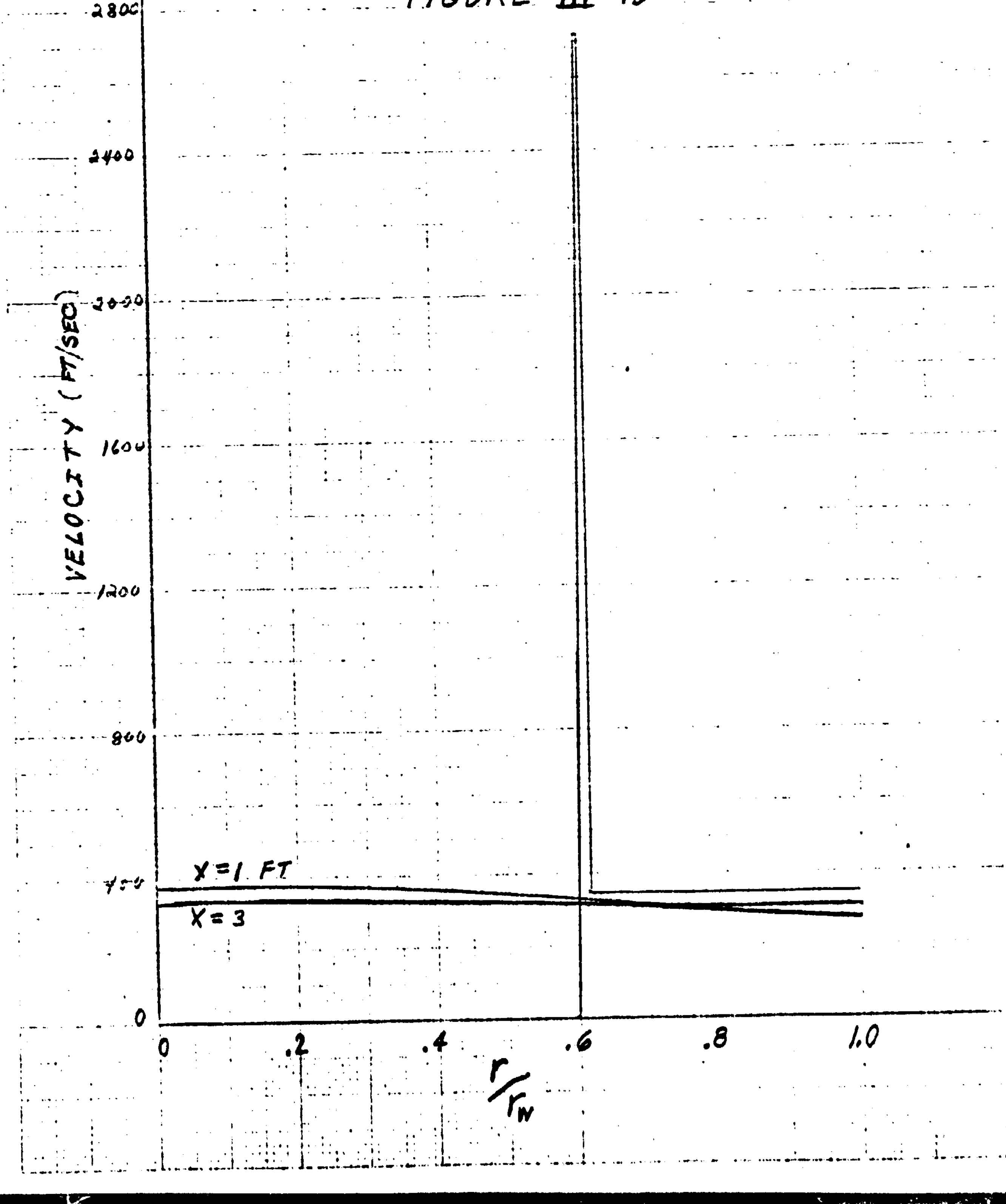

CASE F
STOICHIOMETRIC RATIO PROFILES
RADIAL CONFIGURATION
FIGURE III-14
H_2 ANNULUS AT 3 INCHES
STOICHIOMETRIC (EQUIVALENCE) RATIO
X=1FT
X=2
X=3
X=4
X=5
X=6
10
1
.1
.01
.002
0
.2
.4
.6
.8
1.0
r/rw

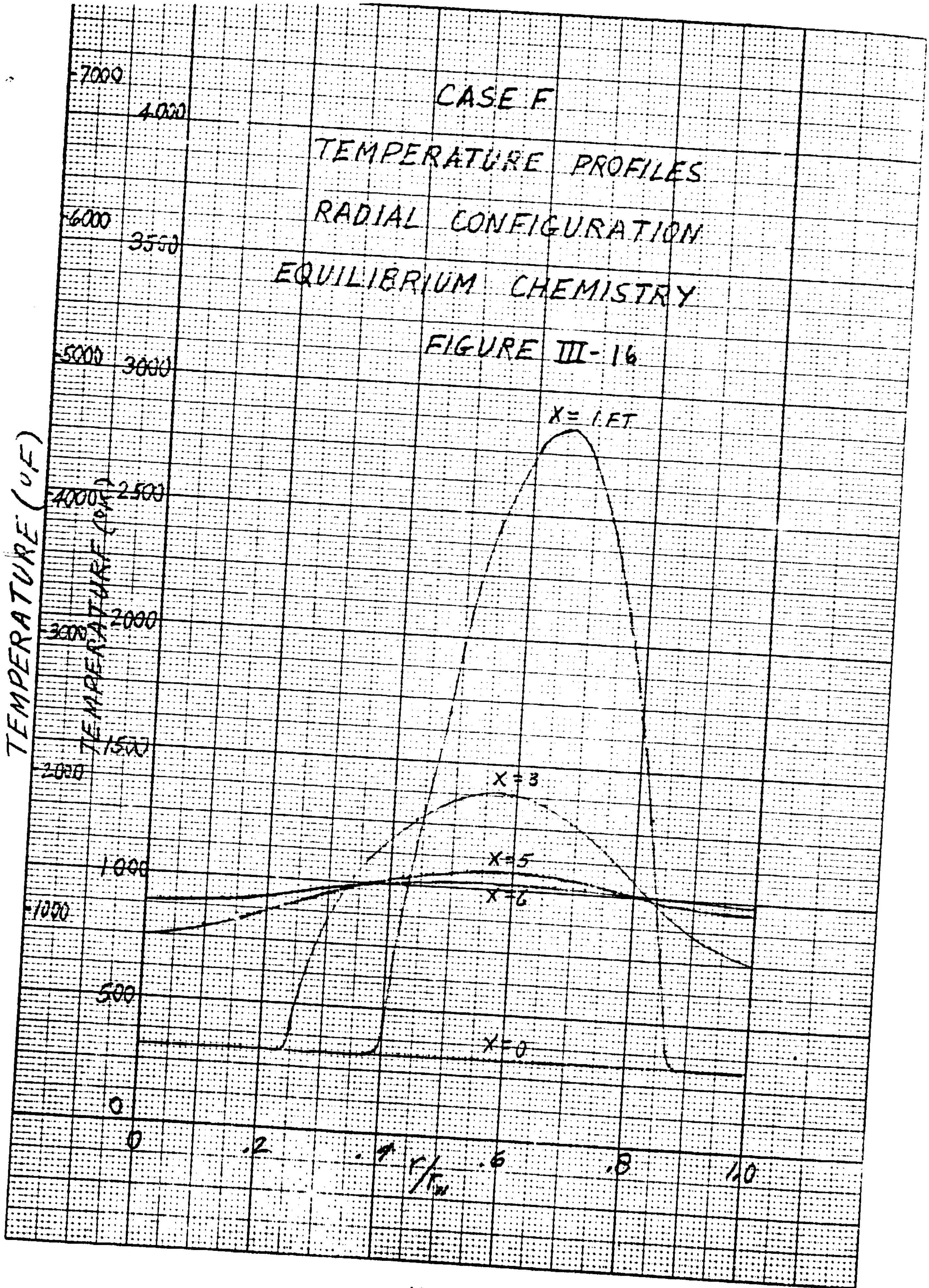
CASE D

VELOCITY PROFILES

RADIAL CONFIGURATION

FIGURE III-15




CASE F
TEMPERATURE PROFILES
RADIAL CONFIGURATION
EQUILIBRIUM CHEMISTRY
FIGURE III-16
X = 1 FT
X = 3
X = 5
X = 6
X = 0
TEMPERATURE (°F)
7000
6000
5000
4000
3000
2000
1000
4000
3500
3000
2500
2000
1500
1000
500
0
0
.2
.4
.6
.8
1.0
r/rw

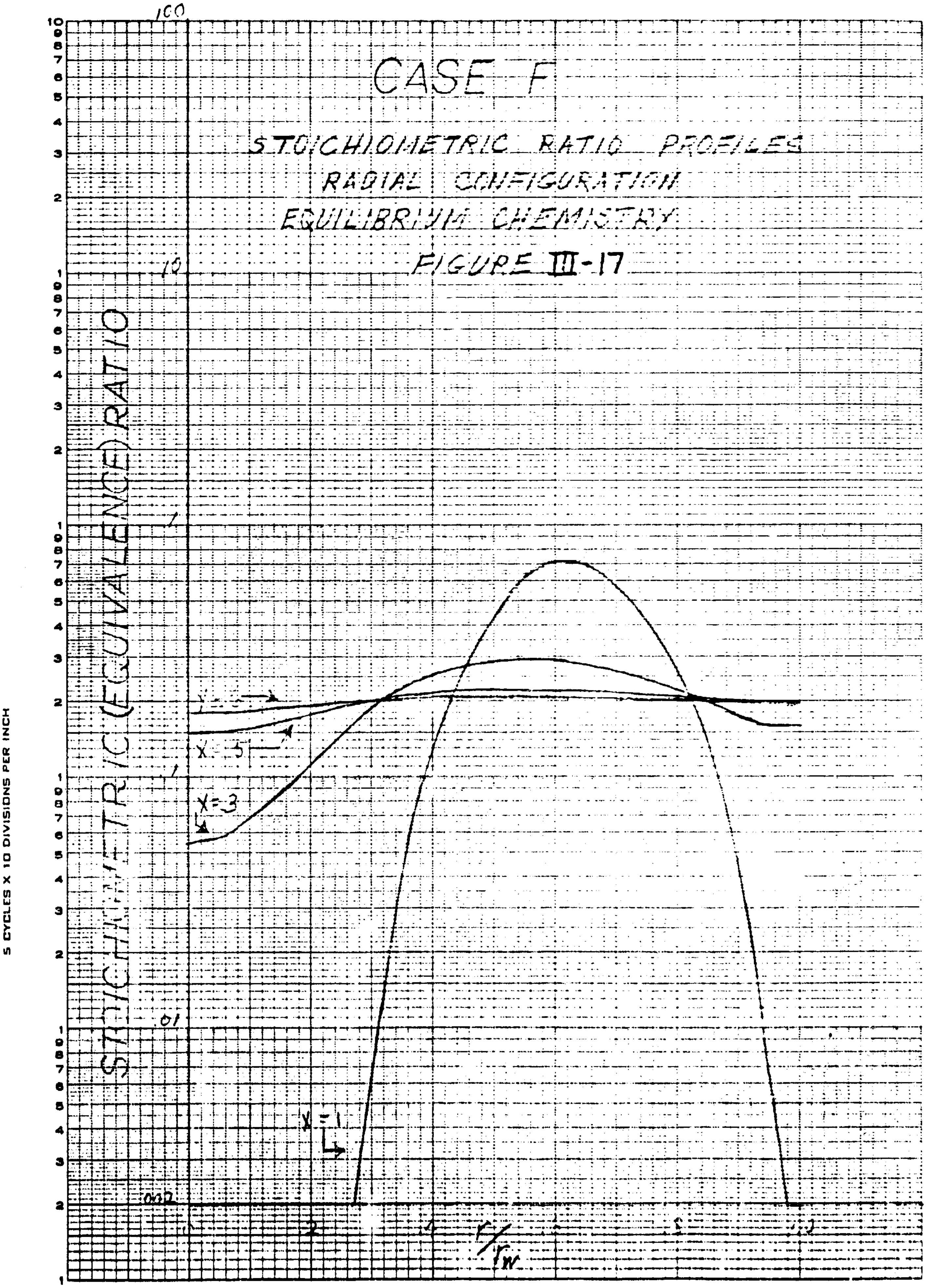

CASE F
STOICHIOMETRIC RATIO PROFILES
RADIAL CONFIGURATION
EQUILIBRIUM CHEMISTRY
FIGURE III-17
STOICHIOMETRIC (EQUIVALENCE) RATIO
X=5
X=3
X=1
r/rw

FIGURE III-18

TEMPERATURE PROFILE FOR ANNULAR INJECTION

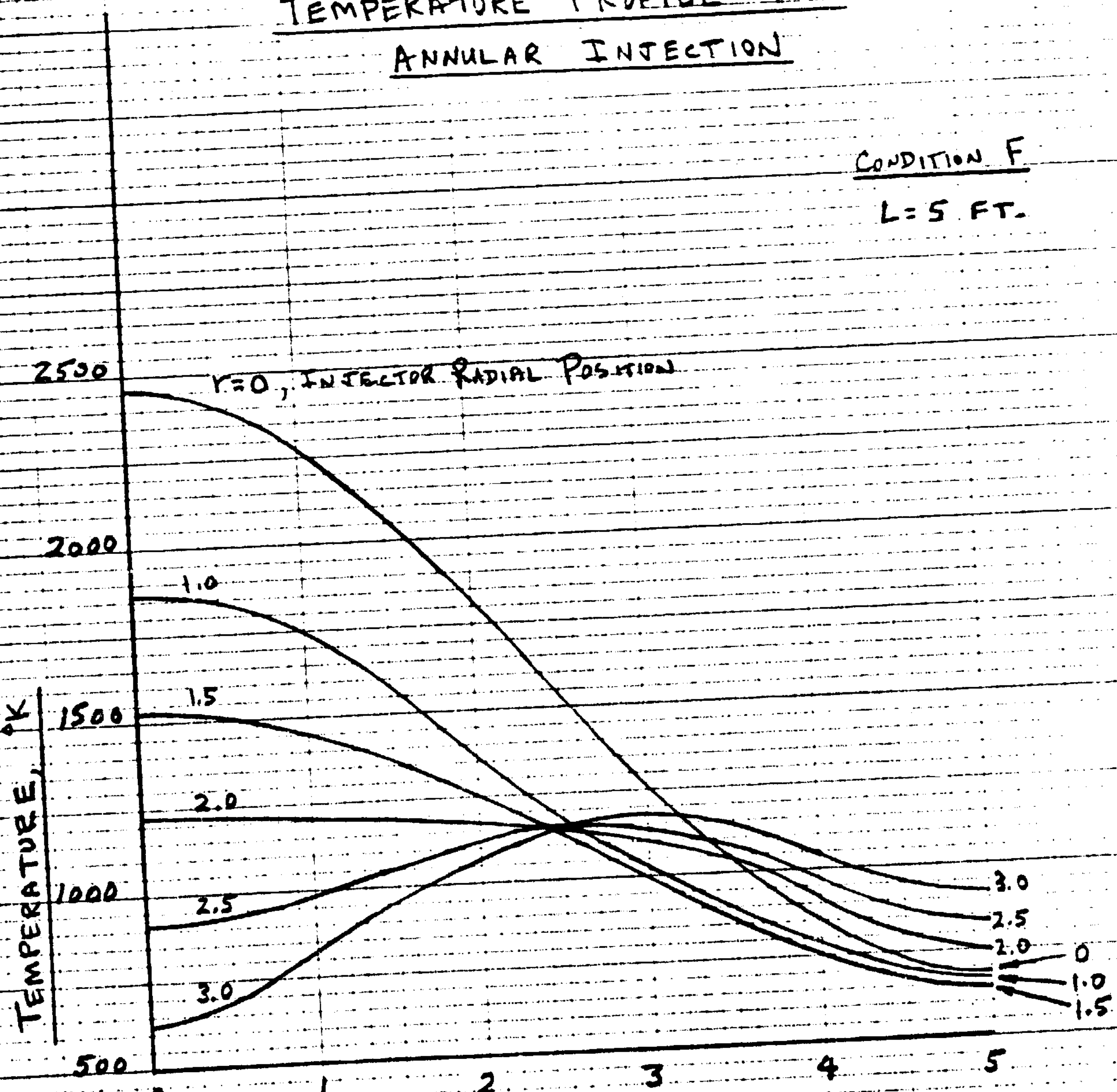


FIG. III-18

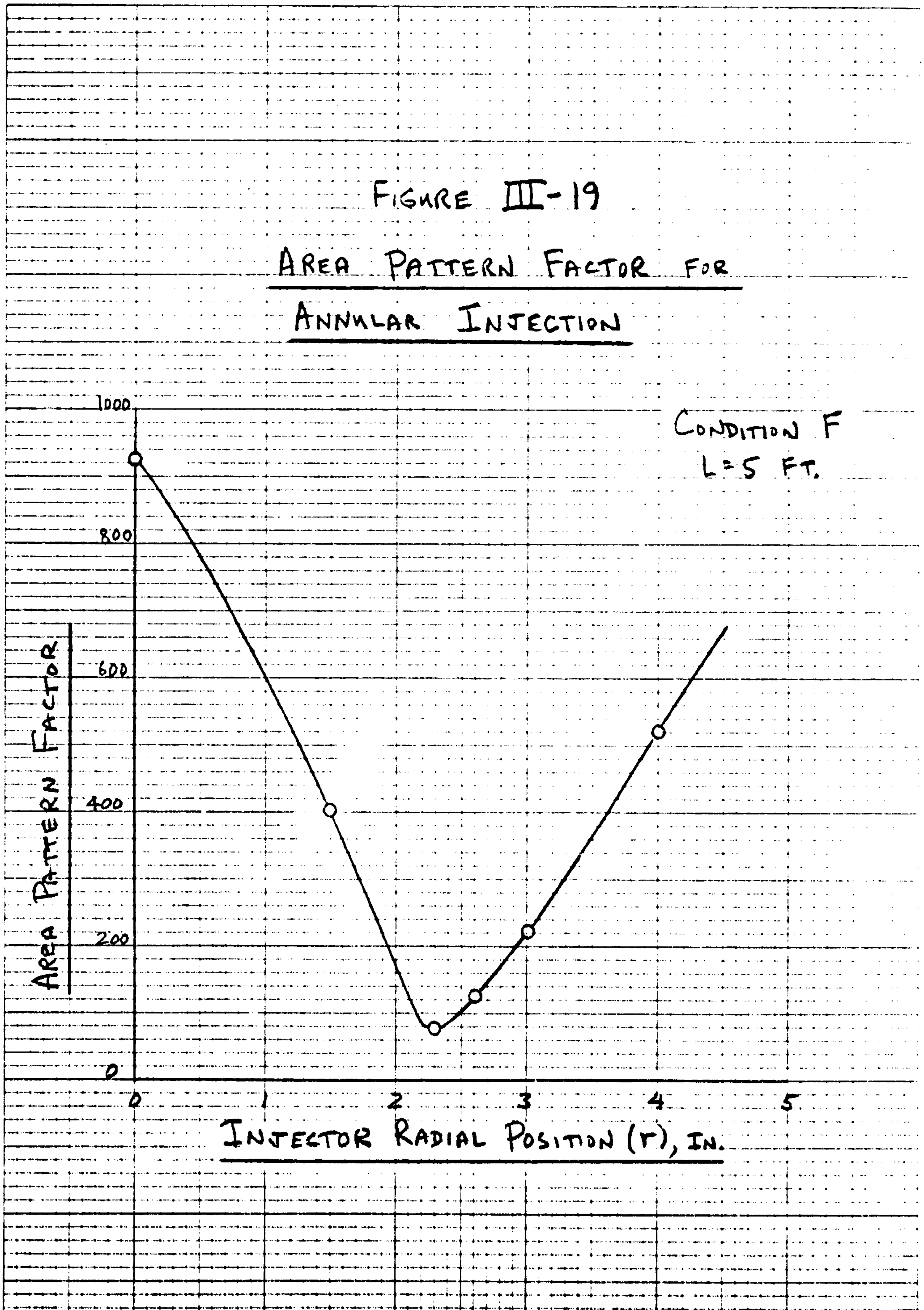


FIGURE III-19
AREA PATTERN FACTOR FOR
ANNULAR INJECTION
CONDITION F
L=5 FT.
AREA PATTERN FACTOR
1000
800
600
400
200
0
0
1
2
3
4
5
INJECTOR RADIAL POSITION (r), IN.

FIG. III-19

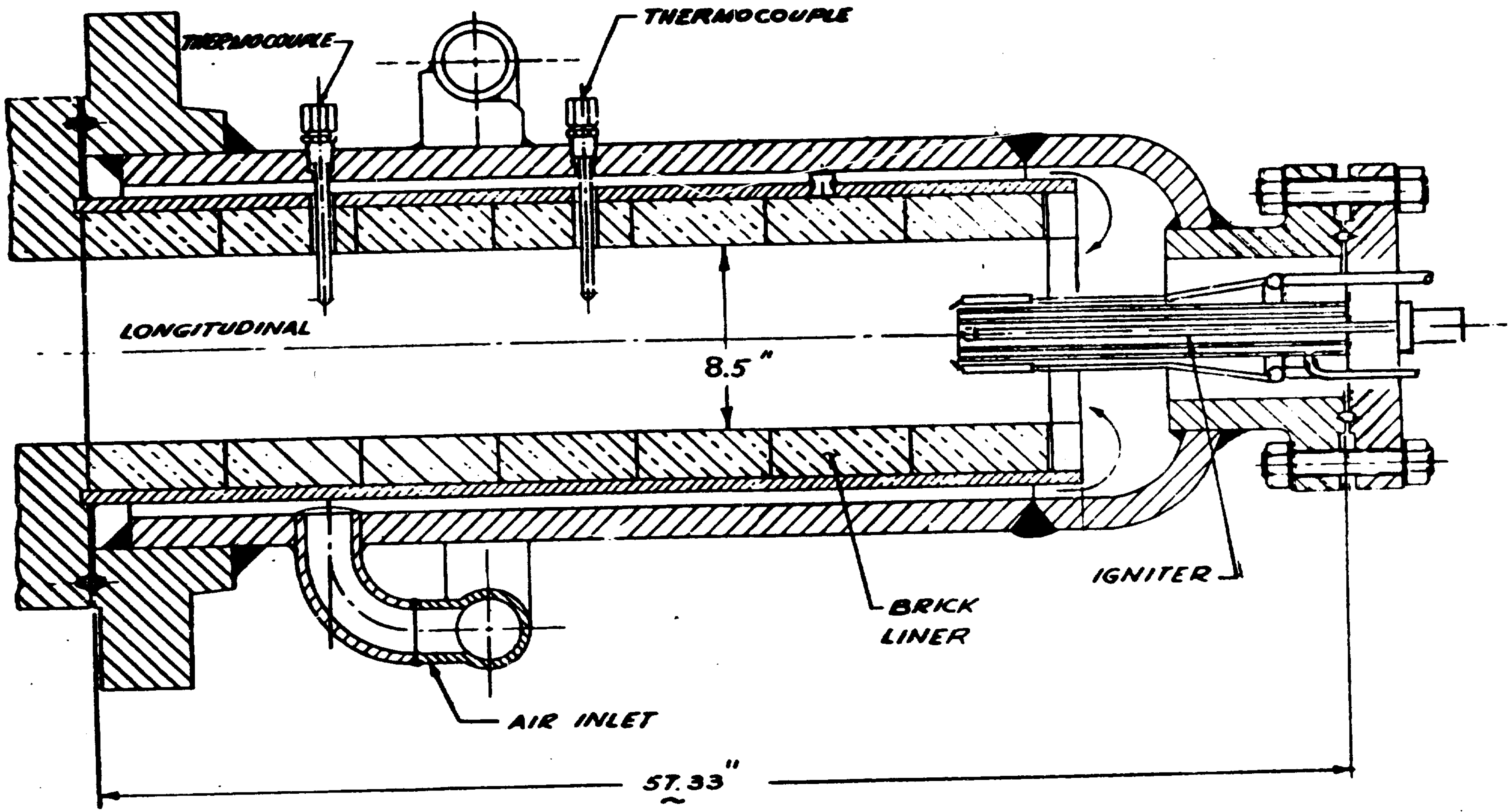


FIGURE III-20 HIGH TEMPERATURE COMBUSTOR NO. 2

PERFORMANCE & DESIGN LIMITS

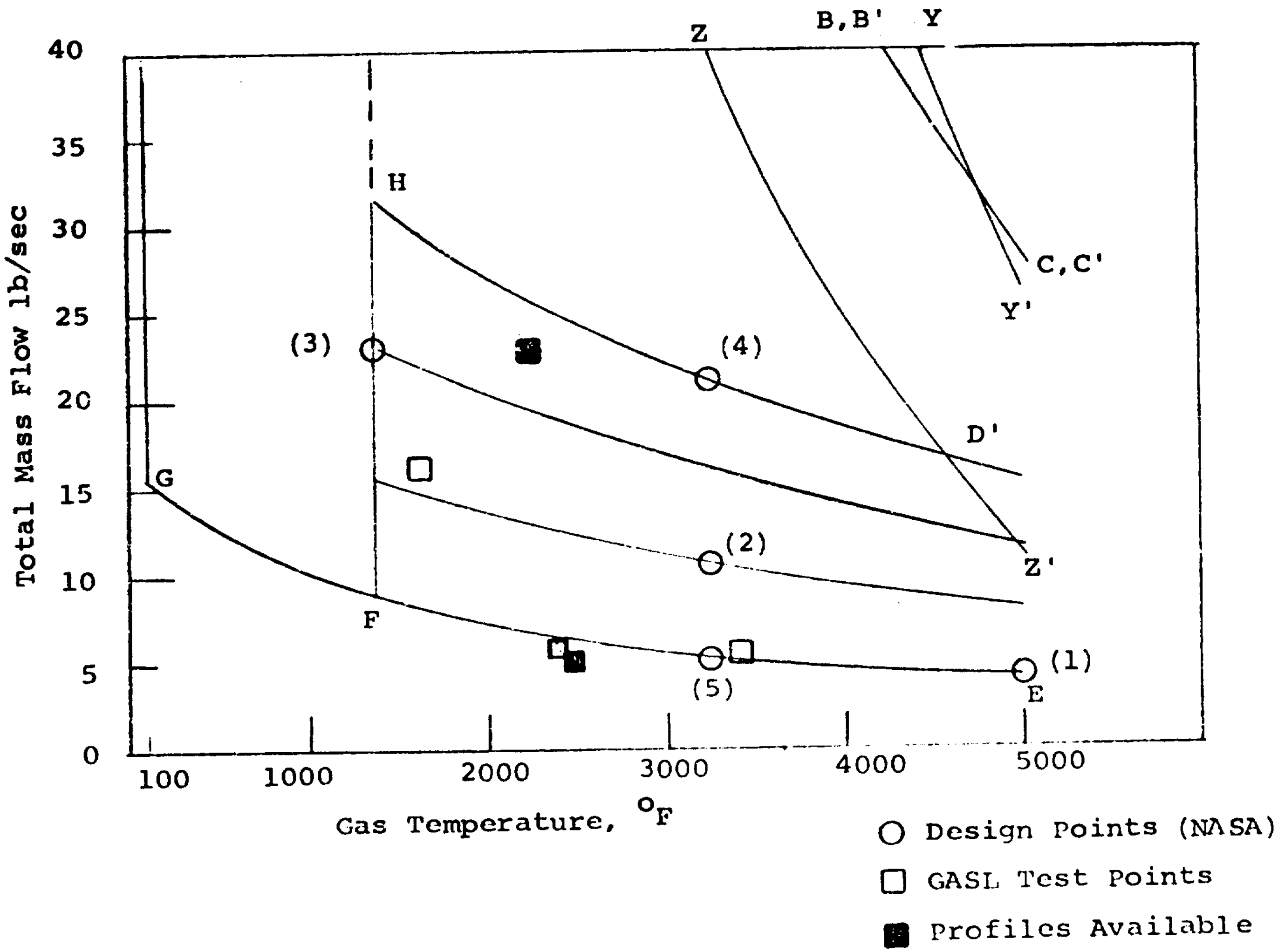


FIGURE III-21- GASL HIGH TEMPERATURE BURNER TEST CONDITIONS

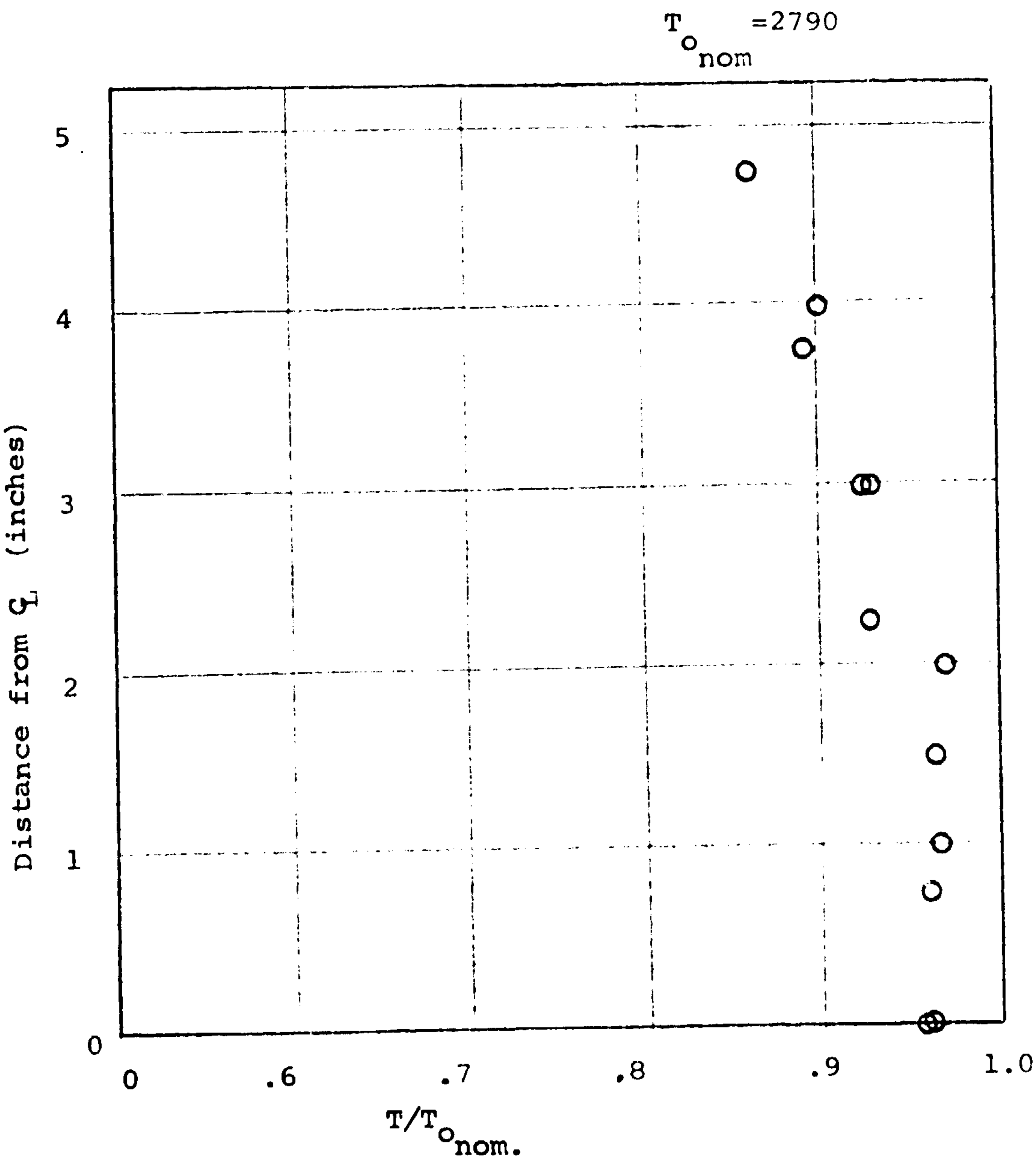
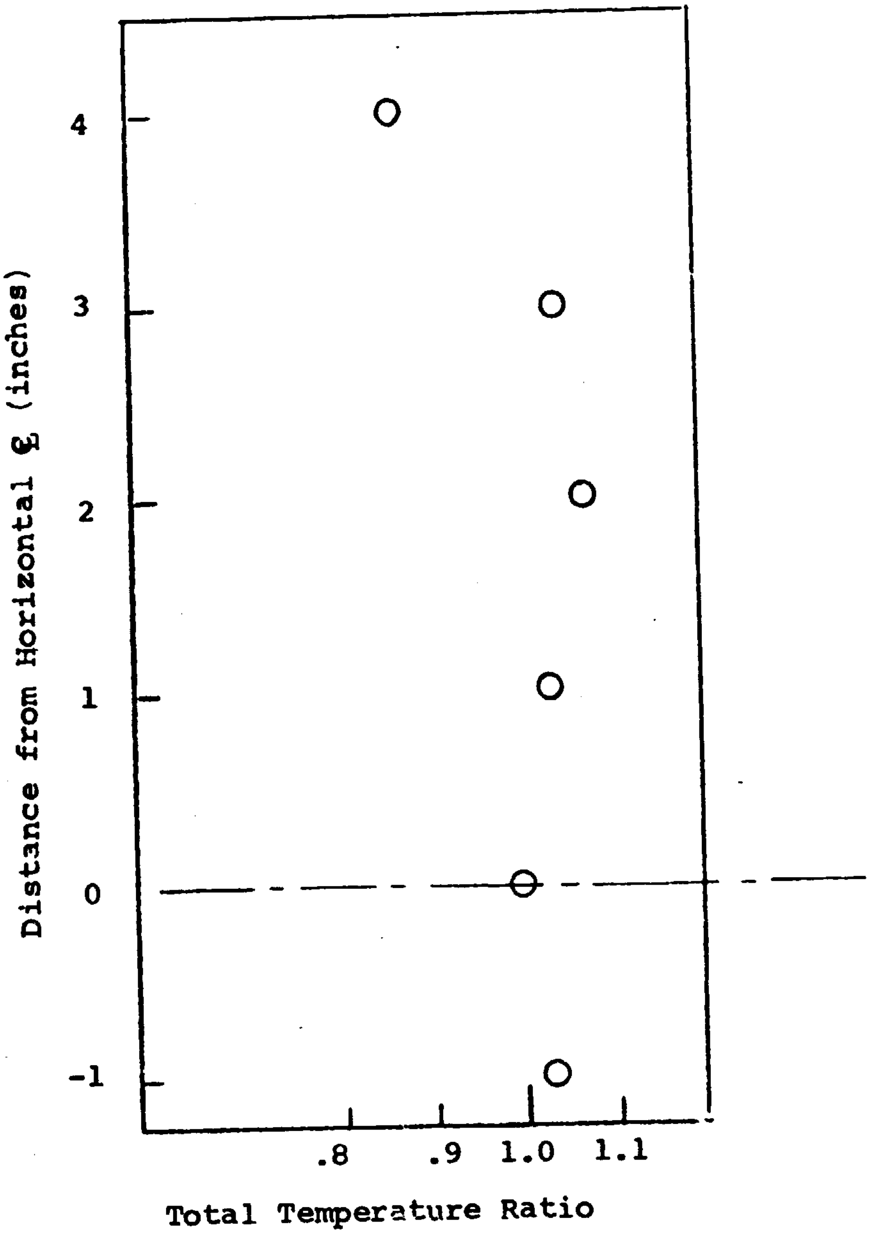


FIGURE III-22- TEMPERATURE MEASUREMENTS IN VERTICAL AXIS OF EXIT PLANE OF M = 5.7 WITH ~12% THROAT COOLING.



$T_{c_{nom}} = 2985^{\circ}R$

FIGURE III-23- TEMPERATURE CALIBRATION IN VERTICAL PLANE OF M = 7.4 CONICAL NOZZLE WITH ~40% THROAT COOLING

SECTION IV

ENGINEERING & DESIGN CALCULATIONS

A. HEAT TRANSFER

1. Heat Transfer to Combustor Wall

The combustor walls are subjected to flow conditions illustrated in the performance envelope of Figure IV-1, defined by points A Z D′ Z′ E F G.

Since efficient mixing and combustion is based on a parallel flow configuration, the determination of peak steady state convective heat flux can be closely approximated by assuming fully developed turbulent flow in a tube. Considering the effect of partial dissociation of the combustion mixture in the higher temperature range (above 4500°R), a heat flux correlation expressed in terms of enthalpy difference, derived from Reference IV-1, determines the convective heat transfer to the combustor wall.

The convective heat flux equation used is derived as follows:

For large temperature differences between gas and wall, the convective heat flux correlation of Reference IV-2 is:

$$\frac{h\,D_e}{k_f} = .023\left(\frac{\rho_f V_s D_e}{\mu_f}\right)^{.8} (Pr)^{1/3}$$

where

h = film coefficient, BTU/sec-ft^2-°R

D_e = hydraulic diameter, ft.

k = thermal conductivity BTU/sec-ft-°R

ρ = density, lb/ft^3

V = velocity, ft/sec

μ = viscosity, lb/ft-sec

Pr = Prandtl No.

Subscripts

f = evaluation at film temperature (average between stream and wall)

s = evaluation at stream temperature

Since $k_f = \frac{C_p \mu_f}{Pr}$, we can substitute for k_f to obtain

$$h = \frac{.023 \; (\rho_f V_s)^{.8} (\mu_f)^{.2} (C_p}{(D_e)^{.2} (Pr)^{2/3}}$$

where

C_p = specific heat, BTU/lb-^oR.

Since $\dot{q}_c = h(T_s - T_w)$

where

$\dot{q}_c$ = convective heat flux, BTU/sec-ft^2

T = temperature, ^oR

Subscript

w = evaluation at wall

and $\rho_f V_s = \rho_s V_s \left(\frac{T_s}{T_f}\right) = \left(\frac{\dot{\omega}}{A}\right) \left(\frac{T_s}{T_f}\right)$

we may again substitute to obtain

$$\dot{q}_c = \frac{.023 \left(\frac{\dot{\omega}}{A}\right)^{.8} \left(\frac{T_s}{T_f}\right)^{.8} (\mu_f)^{.2} (C_p) \; (T_s - T_w)}{(D_e)^{.2} \; (Pr)^{2/3}}$$

where

$\dot{\omega}$ = total gas flow rate, lb/sec

A = flow area, ft^2

Since

ΔH = enthalpy difference between hot gas and cold wall, BTU/lb

H_g = enthalpy of gas at stream temperature conditions, BTU/lb

H_w = enthalpy of gas at wall temperature conditions, BTU/lb

Thus:

$$q_c = \frac{.023 \left(\frac{\dot{\omega}}{A}\right)^{.8} \left(\frac{T_s}{T_f}\right)^{.8} (\mu_f)^{.2} (H_g - H_w)}{(D_e)^{.2} (Pr)^{.67}}$$

In Reference IV-3 and 4, it is recommended that properties be evaluated at a reference enthalpy when there is a wide variation of gas properties with temperature. This is especially true at specification condition Z′ where, at 5460 R, the gas has a very high specific heat and some dissociation.

In References IV-3 and 4 the reference enthalpy is evaluated as

$$H_* = H_{st} + .5\ (H_w - H_{st}) + .22\ (H_r - H_{st})$$

where subscripts

$*$ = evaluation of reference enthalpy

st = evaluation at static stream conditions

w = evaluation at wall conditions

r = evaluation at a recovery enthalpy condition

Since the gas velocity is much less than Mach 1,

$$H_r = H_{st}\ (=H_g)$$

and we have

$$.22(H_r - H_{st}) = 0$$

so that

$$H_* = .5\ (H_{st} + H_w)$$

$$= .5\ (H_g + H_w)$$

In the report we have thus used:

$$\dot{q}_c = \frac{.023\,(\dot{\omega}/A)^{.8}\ (T_g/T_*)^{.8}\,(\mu_*)^{.2}\,(H_g - H_w)}{(D_e)^{.2}\ (Pr)^{.67}}$$

In Reference IV-1 the correlation:

$$\dot{q} = \frac{.0296\ (\dot{\omega}/A)^{.8}\ (\mu_s)^{.2}\ (H_g - H_w)}{(D_e)^{.2}\ (Pr)^{.67}}$$

may be derived from relations presented in that reference, where properties are evaluated at stream conditions. Similar results would be obtained at condition Z since:

$$.023\ \left(\frac{T_g}{T_*}\right)^{.8}\ (\mu_*)^{.2} \approx .0296\ (\mu_s)^{.2}\ .$$

An "Enthalpy-Temperature Curve," Figure IV-1, is used to determine the difference in enthalpy (ΔH) between the gas at stagnation conditions (H_g), and the cooled wall (H_w). The values of enthalpy are obtained from the output of the same computer program used to determine the equilibrium composition of the gaseous system.

In particular, for specification condition Z′ note from the curve for a_{H_2O} = .41, that:

$H_g \simeq +100$ BTU/lb at gas stream temperature T_g = 5460 R (=5000 F)

and

$H_w \simeq -2300$ BTU/lb at wall temperature T_w = 800 R (=340 F)

These absolute values are considered approximate since they are plotted for 600 psia initial pressure and 41 percent by weight water vapor constant, whereas condition Z′ is at 750 psia with a slightly lower water vapor content. However, the difference in enthalpy (ΔH) is considered sufficiently accurate to be used for heat transfer calculation purposes.

Convection:

$$\dot{q}_c = \frac{.023(\dot{\omega}/A)^{.8}(T_g/T_*)^{.8}(\mu_*)^{.2}\ (H_g-H_w)}{(D_e)^{.2}\ (Pr)^{.67}}$$

Sample Calculation for Spec. Condition Z′

$\dot{\omega}$ = 11.0 lb/sec

T_o = 5000 F = 5460 R

$p \approx$ 750 psia combustor pressure (ref.)

$H_g \cong$ +100 BTU/lb @ T_y = 5460 R (Equilibrium Data)

$H_w \cong$ -2300 BTU/lb @ T_w = 800 R (Equilibrium Data)

D_e = 9 in. dia. = .75 ft combustor diameter

$A = .785\ (D_e)^2 = .785(.75)^2 = .432$ ft.^{2}

$(\dot{\omega}/A) = 11.0/.432 = 25.4$ lb/sec-ft^2

$$H^* = \frac{H_g+H_w}{2} = \frac{100 - 2300}{2} = -\ 1100 \text{ BTU/lb}$$

$T^* = 3600$ R , $\mu^* \cong 4.5 \times 10^{-5}$ lb/ft-sec

$(T/T^*) = (5460/3600) = 1.52$ Pr $\cong$ .75

$H_g - H_w = +\ 100 - (-2300) = 2400$ BTU/lb

$$\dot{q} = \frac{.023(25.4)^{.8}\ (1.52)^{.8}\ (4.5 \times 10^{-5})^{.2}\ (2400)}{(.75)^{.2}\ \ (.75)^{.67}}$$

$= 179$ BTU/sec-ft^2

Radiation Heat Transfer

The radiation heat transfer calculations employ values of gas emissivity found by using the "upper limit" data of Ferriso, et al. (Reference IV-5), which are determined by experimentation plus analysis and extrapolation.

The heat transfer by radiation can be calculated from the simplified equation of Hottel and Egbert found in Reference IV-6:

$$\dot{q}_r = E_o \, \delta \, [E_g T_g^{\,4} - \alpha_{gb} T_w^{\,4}]$$

where:

$\dot{q}_r$ = radiation heat flux from gas to wall, BTU/sec-ft^2

E_e = effective emissivity coefficient for the wall surface

E_g = emissivity of the gas at temperature T_g

T_g = temperature of the gas, ^oR

T_w = temperature of the wall, ^oR

α_{gb} = absorptivity of the gas

δ = Stephan-Boltzmann constant, .48 x 10^{-12} BTU/sec-ft^2-$^oR^4$

Because $T_g >> T_w$, the term $\alpha_{gb} \, T_w^{\,4}$ may be considered negligible and we therefore have:

$$\dot{q}_r \doteq E_e \delta E_g T_g^{\,4} \; .$$

The values of gas emissivity, E_g, were determined by a linear extrapolation of the semi-log plot of Reference IV-5, and for the condition of interest in this application results in values of the order of E_g = .45 to .50 for temperature between 4000 to 5000^{o}F.

For the effective wall surface emissivity coefficient, a value of E_e = .5 is used. This value is considered as a conservative estimate between Hottel's approximation given in Reference IV-6 of:

$$E_o = \frac{E_{gr} + 1}{2}$$

where E_{gr} = emissivity of the wall surface considered as a grey radiator, (this equation was found to be a fair approximation for values of E_g = .80 to .95, the range most frequently encountered in industrial practice), and the other extreme where

$$E_o = E_{gr}$$

for very low values of emissivity, such as E_{gr} = .072 for "commercial-scrapped shiny, but not mirror-like" presented by Hottel in Reference IV-7. Since the maximum copper wall temperature will not exceed 560°F, a value of emissivity of E_{gr} = .57 for a copper plate heated at 1110°F is considered too high for this application, and therefore, the value of E_e = .5 is considered a good practical effective emissivity for design purposes. Radiation heat transfer results are plotted for varying combustion temperature at total pressures of 750 and 375 psia in Figure IV-2.

Sample Calculation:

Total Pressure, P_T = 750 psia

$T_T = 3130^\circ K = 5632^\circ R$

Mole Percent of H_2O = 50%

Partial Pressure Water, P_{H_2O} = 375 psia = 25.5 atm

Characterizing Length, L = .9D = .9(9 in.) = 8.1 in. = 20.6 cm.

PL = (25.5) (20.6) = 525 cm-atm

From Figure 5 of Reference IV-5, the extrapolated value of E_g = .48

$T_g^4 = (5636)^4 = 1000 \times 10^{12}\,{}^\circ R^4$

$E_e = .5$

$\dot{q}_r = E_e \delta E_g T_g^4 = (.5)(.48\times10^{-12})(.48)(1000\times10^{12}) = 115$ BTU/sec-ft^2

TOTAL HEAT FLUX

At specification condition Z', we note that the convective heat flux, $\dot{q}_c$ = 179 added to a radiation rate of $\dot{q}_r$ = 96 found from Figure IV-2, results in a total heat flux of $\dot{q}_r$ = 275 BTU/sec-ft^2. Figure IV-3 is a curve of the heat flux to the combustor walls at all specification envelope conditions.

2. Water Cooling Calculations

The water cooling calculations are based on the forced convection heat transfer correlation of Seider and Tate for tubes, found in Reference IV-8.

Calculations to determine wall temperatures as a function of heat flux with water velocity as a parameter were performed and the results presented in Figures IV-4 and IV-5 for wall gas side and water side temperature respectively.

COOLING CALCULATIONS:

$$\frac{h}{C_p G}\left(\frac{\mu C_p}{K}\right)_b^{2/3}\left(\frac{\mu_w}{\mu_b}\right)^{.14} = \frac{.027}{(GD/\mu_b)^{.2}}$$

Sieder-Tate (P.193 Rohsenow) (Ref. IV-8)

$$\dot{q}_w = h\,(T_w - T_b)$$

From the above eqs. we can derive:

$$\dot{q}_w = \frac{.027\left[\frac{K P_r^{1/3}}{\mu^{.8}}\right]_b}{(D_e)^{.2}}\left(\frac{\mu_b}{\mu_w}\right)^{.14}(\rho V)^{.8}\,(T_w - T_b)$$

where:

$\dot{q}_w$ = water-side (cooling) heat flux, Btu/sec-ft²

K = thermal conductivity of water, Btu/sec-ft-°R

P_r = Prantl No. of water

μ = Viscosity of water, lb/ft-sec

ρ = density of water lb/ft³ = 62.4

D_e = Hydraulic Diameter of Water Passage, Ft.

T_w = Coolant Side Wall Temperature, °F

T_b = Water Bulk Temp, °F

Subscripts

b = refers to bulk temp conditions

w = refers to water coolant side wall temp. conditions.

Sample Calculation

$T_w = 200\ F \qquad T_b = 100\ F$

$T_w - T_b = 200 - 100 = 100$

$(K\,Pr^{1/3}/\mu^{.8})_b = .785 \times 10^{-1}$

$\mu_w = 20.6 \times 10^{-5}$ lb/ft-sec @ 200 F

$\mu_b = 45.9 \times 10^{-5}$ @ 100 F

$(\mu_b/\mu_w)^{.14} = 1.119$

$V = 20$ ft/sec $\qquad \rho = 62.4$ lb/ft^3

$(\rho V)^{.8} = (62.4 \times 20)^{.8} = 300$

$De = 2 \times .25 = .50/12 = 4.16 \times 10^{-2}$ ft, $De^{.2} = .529$

$$\dot{q} = \frac{(.027)\left[.785 \times 10^{-1}\right](1.119)(300)(100)}{.529}$$

$= 134$ Btu/sec-ft^2 See Figure IV-4

WALL TEMPERATURES:

MAT'L: "AMZIRC" — American Metal Climax Designation for Zirconium Copper with .13 - .20 Zr

$K = 90\%$ Pure Copper Thermal Cond.

$= .90\ (226) = 203$ Btu / Hr-ft-°R

$= 56.0 \times 10^{-3}$ Btu/sec-ft-°R

$$\dot{q}_w = (K/x)\ (T_{W_H} - T_{W_C})$$

where:

$\dot{q}_w$ = Heat Flux thru wall, Btu/sec-ft²

K = thermal conductivity, Btu/sec-ft-°R

T_{W_H} = Hot Gas Side Wall Temp, °F

T_{W_C} = Water Coolant Side Wall Temp, °F

$$T_{W_H} - T_{W_C} = \frac{\dot{q}}{(K/x)}$$

$$T_{W_H} = T_{W_C} + \frac{\dot{q}}{(K/x)}$$ See Fig. IV-5

WATER COOLING REQUIREMENTS

$V_w = 20$ ft/sec

$A_w = .785\,(D_{OD}^2 - D_{ID}^2)$
$= .785[(10.5)^2 - (10.0)^2]$
$= .785\,(10.25)$
$= 8.05$ in^2
$= .056$ ft^2

$\dot{w}_w = \rho_w A_w V_w = (62.4)(.056)(20)$
$= 69.9$ lb/sec (×7.2)
$= 503$ gpm

WATER BULK RISE:

$\dot{q}_{max} = 220$ Btu/sec-ft^2

$q_{AVE} = 110$ Assumed L = 4.5 ft

$A_S = \pi D L$ = Surface Area
$= 3.14\,(.75)(4.5)$
$= 10.6$ ft^2

$Q = (\dot{q}_{AVE})(A_S) = 110\,(10.6)$
$= 1165$ Btu/sec Heat Load

$Q = \dot{w} C_p \Delta T_B$

$\Delta T_B = Q/\dot{w} C_p = (1165)/(69.9) \times (1) = 16.7\ \Delta°F.$

Heat Loss from Cooling

For $\dot{q} = 220$ Btu/sec-ft^2 (Cond Z')

$q_{ave} = 110$ $A = 10.6$ ft^2

$Q = 1165$ Btu/sec

At Z' $\dot{w}_{Tot} = 11.0$ lb/sec

$$Q' = \frac{1165}{11.0} = 106 \ \frac{\text{Btu}}{\text{lb}}$$

At 5000 F an enthalpy drop of 106 Btu/lb results in a Temp drop of ≈ 70 °F

Wall Temperature based on Available NASA Water Supply

The available NASA-LRC water supply of 350 gpm at 480 psi corresponds to a water coolant velocity of V = 14 ft/sec in the combustor liner passages. This results in increased wall temperatures and thermal growth for any given heat flux. Figure IV-6 shows the wall temperatures with V = 14 for varying heat flux. Also shown are values for V = 20 ft/sec.

Liner Temperature Response at Various Thermocouple Depths

An analysis has been made to determine the zirconium copper liner wall transient temperature for various fractional wall depths as a parameter. The calculations were performed assuming the wall to be a plate with sudden exposure to a constant heat input on one side with the other side insulated (no water cooling), to simulate a malfunction type condition. The temperature response data of Reference IV-12 was used, where:

$$F_o = \frac{\alpha\theta}{\delta^2}$$

$$T = \left(\frac{k}{\delta\dot{q}}\right)(t-t_o)$$

where:

F_o = Fourier No. = $\frac{\alpha\theta}{\delta^2}$

T = dimensionless temperature

α = thermal diffusivity of body material = $\frac{k}{\rho c}$, $\frac{ft^2}{sec}$

θ = time, sec

δ = plate thickness, ft.

k = thermal conductivity of body material, BTU/sec-ft-^oF

$\dot{q}$ = rate of heat input, BTU/sec-ft^2

t = temperature, ^oF

t_o = initial temperature, ^oF

ρ = depth of body material, lb/ft^3

C_p = specific heat of body material, BTU/lb-^oF

$X = \frac{\lambda}{\delta}$, depth ratio

x = depth, ft.

($x = 0$ at $X = 0$)

($x = \delta$ at $X = 1$)

For zirconium copper

$$K = 200 \text{ BTU/lb-ft-}^{\circ}\text{F}$$
$$= 5.56 \text{ BTU/sec-ft-}^{\circ}\text{F}$$
$$\rho = 559 \text{ lb/ft}^3$$
$$C_p = .0915 \text{ BTU/lb-}^{\circ}\text{F}$$
$$\alpha = \frac{K}{\rho C_p} = 1.09 \times 10^{-3} \text{ ft}^2\text{/sec}$$
$$\delta = .5/12 = .417 \times 10^{-1} \text{ ft}$$

<u>SAMPLE CALCULATION:</u>

At $\theta = 1$ sec, $q = 200$ BTU/dec-ft^2

$$F_o = \frac{\alpha\theta}{\delta^2} = \frac{(1.09\text{x}10^{-3})(1)}{(.174 \times 10^{-2})}$$
$$= .627$$

At $x = 0$, $X = 0$

From Chart 43 of Reference IV-12

$$T = .96$$
$$t - t_o = \frac{\delta}{K} T \dot{q}$$
$$= \frac{(.417\text{x}10^{-1})(.96)(200)}{(5.56 \times 10^{-2})}$$
$$= 144^{\circ}\Delta\text{F}$$
$$t_o = 80\text{F}$$
$$t = 80 + 144 = 224\ ^{\circ}\text{F at } \theta = 1 \text{ sec}$$

The results of these computations are shown in Figures IV-7 and 8. Figure IV-7 shows the temperature response at one second as a function of heat flux, with depth as parameter, and Figure IV-8 shows the temperature response at $\dot{q}$ = 275 as a function of time, again with depth as parameter.

From Figure IV-7 we see that at a depth of X = 2/3, x = .333 in (distance of .167 inches from insulated side) there is a temperature rise of 110 Δ^{o}F after 1 second exposure. This compares to a rise of 200 Δ^{o}F at X = 0 (hot surface). A thermocouple embedded a distance of .167 inches from the cold side (X = 2/3) is therefore considered adequate.

THERMOCOUPLE INSTALLATION

To secure the 1/16" diameter sheathed thermocouple in place, a thermally conductive cement is to be used. This is preferred rather than a true soldering operation in order to avoid heating the entire liner after machining.

The cement specified is "Eccobond Solder 57C," an epoxide based room temperature curing conductive adhesive. It is used in applications where conventional hot soldering is impractical. The thermal conductivity given by the manufacturer (Emerson and Cuming, Inc.) is k > 200 BTU/ft^2/hr/^{o}F/in.

3. Air Cooling Analysis

An analysis was performed to determine the suitability of using heater supply air as a combustor wall coolant, prior to injection into the main combustion chamber.

The analysis, evaluated at condition Z', indicated that air was not an appropriate coolant choice for this application. The analysis follows.

AIR COOLING ANALYSIS: Condition. 2'

$\dot{q} \cong 150$ Btu/sec for $T_{WH} = 1400$ F (Inconel X-750)
& D = 10 inch I.D.

$$\Delta T_w = \frac{\dot{q}}{K/x}$$

where:

ΔT_w = temp diff. across wall

K = thermal cond. of inconel at approx 1200 °F $= 3.3\times10^{-3}$ $\frac{\text{Btu-ft}}{\text{sec-ft}^2\text{-°R}}$

$x = \frac{.078}{12} = .0065 = 6.5\times10^{-3}$

$$\Delta T_w = \frac{180}{3.3\times10^{-3}/6.5\times10^{-3}} = \frac{180}{.51} = 350$$

$$S_{tt} = \frac{E\alpha\Delta T_w}{1.4}$$

$E = 30\times10^6$ psi
$\alpha = 8\times10^{-6}$ in/in-°F

where:

S_{tt} = Thermal stress from temp diff. across wall, psi

α = Coeff. of thermal expansion, in/in-°F

$$S_{tt} = \frac{(30\times10^6)(8\times10^{-6})(350)}{1.4} = 60{,}000 \text{ psi}$$

$S_{yield} \cong 120{,}000$ psi for Heat Treated Inconel X

$$P_{cr} = \frac{2E}{1-\nu^2}\left(\frac{t}{d}\right)^3 \text{ psi for long cylinder}$$

$$= \frac{2(30\times10^6)}{(1-.09)}(.078/10)^3 = (6.6\times10^7)(.475\times10^{-6})$$

$$= 31 \text{ psi}$$

$$S.F. = \frac{120\,000}{60\,000} = 2.0$$

Calc of Velocity required to maintain
T_{W_H} at 1400°F & T_{W_C} = 1050 °F

$$\dot{q} = \frac{.023}{D_e^{.2}} \left[\frac{C_p \mu^{.2}}{P_r^{2/3}} \right]_f (\rho V)^{.8} (T_{W_C} - T_B)$$

where:

$\dot{q}$ = heat flux from wall to air coolant
D_e = Hydraulic dia, (ft)
C_p = Specific heat Btu/lb-°F
μ = viscosity lb/ft-sec
P_r = Prantl No.
ρ = density of Air, lb/ft³
V = Velocity of Air
T_{W_C} = Air Coolant side Wall Temp, °F
T_B = Air Bulk Temp

Subscript:
f = denotes evaluation at film temp
where $T_f = \frac{T_{W_C} + T_B}{2}$

By Iteration we Can Obtain $\dot{q} \approx 150$ Btu/sec-ft²
Object is to find Velocity Required

$$V^{.8} = \frac{(\dot{q})(D_e)^{.2}}{(.023)(\rho)^{.8}\left[\frac{C_p \mu^{.2}}{P_r^{2/3}}\right]_f (T_{w_c} - T_B)}$$

$\dot{q} = 150$

$D_e = \frac{.125}{12} = 1.04 \times 10^{-2}$ for annulus = .062 in where $D_e = 2 \times$ annulus width.

$D_e^{.2} = .40$

$\rho = \frac{P \times 144}{R \times T_B} = \frac{750 \times 144}{53.3 \times 660} = 3.07$ lb/ft³

$(\rho)^{.8} = 2.45$

$T_{w_c} = 1050$ F

$T_B = 200$ F

$T_{w_c} - T_B = 850$

$T_f = \frac{1050 + 200}{2} = 625$ °F

$C_{p_f} = .25$ Btu/lb-°F

$\mu_f = 2.0 \times 10^{-5}$ lb/ft-sec

$(\mu_f)^{.2} = 1.15 \times 10^{-1}$

$P_{r_f} = .68$

$(P_r)_f^{2/3} = .77$

$\left[\frac{C_p \mu^{.2}}{P_{r_f}^{2/3}}\right]_f = \frac{(.25)(.115)}{(.77)} = .374 \times 10^{-1}$

T_w.

$$(V)^{.8} = \frac{(150)(.40)}{(.023)(2.45)[.374\times10^{-1}](850)}$$

$$(V)^{.8} = 33.6$$

$$V = (33.6)^{1.25}$$

$$V = 80 \text{ Ft/sec}$$

Annulus Calc Check

$$\dot{w} = \rho A V$$

where A = Annulus Flow Area ft^2

$$A = \pi D t$$

where D = Annulus Diameter ft. = .84
t = annulus width ft

$$A = \frac{\dot{w}}{\rho V} =$$

$\dot{w}$ = air flow for Cond Z
= 4.73 lb/sec

$$A = \frac{4.73}{3.07 \times 80}$$

$$= 1.93\times10^{-2} \; ft^2$$

$$t = \frac{A}{\pi D} = \frac{1.93\times10^{-2}}{3.14\,(.84)} = .73\times10^{-2} \; ft$$

= .088 inches – this is very small! difficult to control

Pressure Drop

$$\Delta P_f = f\left(\frac{L}{D}\right)\frac{\rho V^2}{2g \times 144}$$

where:

ΔP_f = pressure drop from friction along liner, psi

f = friction factor = .03 from Moody for e/D = .004

L = Liner Length = $\frac{60}{12}$ in = 5 ft

$D_e = \frac{.088 \times 2}{12} = .0147$ ft hydr. dia

$(f)\frac{L}{D} = \frac{.03(5)}{.0147} = 10.2$

$\rho = 3.07$ lb/ft^2 at $p = 750$ psi, $T = 660$ R

$V = 80$ ft/sec

$$\frac{\rho V^2}{2g \times 144} = \frac{3.07(80)^2}{9.28 \times 10^3} = 2.12$$

$$\Delta P_f = 10.2(2.12) = 21.6 \text{ psi}$$

BUCKLING SAFETY FACTOR

$$S.F. = \frac{P_{CR}}{\Delta P_f} = \frac{31}{21.6} = 1.44$$ this is too low

Stresses Induced by Thermal Restraint

$$S_{tr} = E\alpha\Delta T = (30 \times 10^6)(8 \times 10^{-6})(100) = 24{,}000 \text{ psi}/100^\circ\text{F restraint}$$

BULK RISE OF AIR

$Q = \dot{w} C_p \Delta T_B$

$q_{ave} = \frac{150}{2} = 75$ Btu/sec ft²

$C_p = .25$

$\dot{w} = 4.73$ lb/sec AIR

$A_s = \pi D L = (3.14)(.83)(5)$

$= 13.0\ ft^2$

$Q = 75(13) = 980$ Btu/sec

$\Delta T_B = \frac{980}{(.25)(4.73)} = 810\ °\Delta R$ Very High

This type of bulk rise would result in greatly increased pressure drop accross liner over that estimated for $T_{B_{ave}} = 200$ F calculated

CONCLUSION - Air Cooling Analysis

Combination of unfavorable characteristics using air cooling - at condition Z', makes air cooling undesirable. These adverse conditions are high bulk temp. rise, low buckling safety factors from high pressure drop through liner cooling passage.

Lower velocities result in higher surface temps with drop in yield strength and stress safety factor

Increased velocity results in still thinner annulars spacing (< .085"), higher liner pressure drops and lower buckling safety factor - prohibitive.

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Use of thinner walls (less than .078") with ribbed supports is undesirable since thermal restraint stress is quite high, being S_{tr} = 24,000 psi / 100°F restraint.

It is considered too risky to mix oxygen with the air for use as a coolant since this would introduce problems in cleanliness of the air lines, liner, manifolds, etc because of the large quantities of oxygen req'd (over 50%) in the mixture. Leakage of this oxygen enriched, pressurized air through combustor liner into combustor could cause problems.

Conclusion was to use water cooled design.

B.1. LINER STRESS ANALYSIS:

PRESSURE INDUCED STRESS:

MAXIMUM PRESSURE INDUCED STRESS OCCURS AT START & END OF EACH RUN, WITH FULL WATER PRESSURE ACTING EXTERNALLY ON CYLINDRICAL LINER TO INDUCE COMPRESSIVE STRESS.

$$S_c = \frac{pd}{2t} \text{ psi (tangential stress)}$$

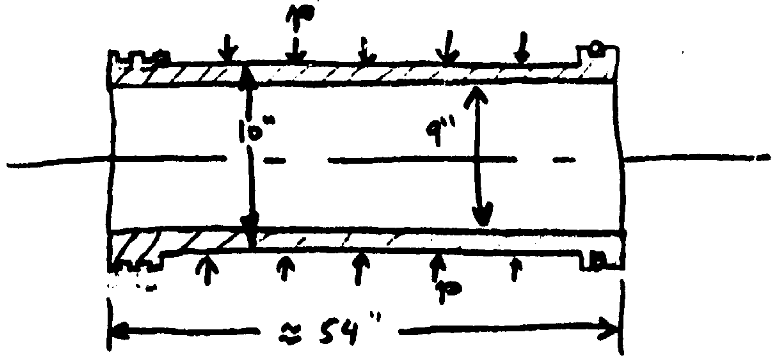
where:

S_c = Compressive Stress. psi

p = water pressure, = 1200 psi

d = liner diameter, taken at O.D. to be conservative = 10 in.

t = liner thickness = .50 in



$$S = \frac{1200(10)}{2(.5)} = 12{,}000 \text{ psi compression}$$

THERMAL STRESS:

The liner is designed with a piston type seal so that it can expand longitudinally, therefore longitudinal thermal stress is minimal. Thermal stress from temp. difference accross wall must be evaluated.

$$S_{tt} = \frac{E \alpha \Delta T_w}{2(1-\nu)} \text{ psi}$$

{ Compression on Hot Face (I.D.)
Tension on Cold Face (O.D.) }

Roark, "Formulas for Stress & Strain" 3rd Ed. p.335 Case 5.

where:

S_{tt} = Thermal Stress induced by temp. difference accross wall, psi

E = Modulus of Elasticity, lb/in²

α = Expansion Coefficient in/in-°R

ΔT_W = Temperature Difference accross Wall of liner, °R

ν = Poisson's Ratio

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Sample Calculation

$\mu = .3$

$E = 19.0 \times 10^6$ lb/in²

$\alpha = 9.3 \times 10^{-6}$ in/in-°R

$\Delta T = 150$ ΔR at $\dot{q} = 200$ Btu/sec-ft², $t = .50$ in

$$S_{tt} = \frac{(19 \times 10^6)(9.3 \times 10^{-6})(150)}{2\,(1-.3)}$$

$= 18,900$ psi { compr. on hot face / tension on cold face }

See Figure IV-9 for Curve of Thermal Stress

MAX THERMAL STRESS:

At Condition Z' (from Fig. IV-9)

$\dot{q} = 275$ Btu/sec-ft²

$S_{tt} = 26,000$ psi

$S_{yield} = 35,000$ psi for Zirconium Copper FORGING 11" DIA "WARM WORKED" & AGED. ZIRCONIUM COPPER WILL MAINTAIN THIS YIELD STRENGTH TO 800 °F BEFORE DROP IN VALUE OCCURS.

$$S.F. = \frac{35,000}{26,000} = 1.35$$

A Safety Factor, S.F. = 2.0 is obtained when $S_{tt} = 17,500$ psi which corresponds to $\dot{q} = 185$ Btu/sec-ft²

Combined Pressure & Thermally Induced Stress

$$S_c = S_p \pm S_{tt}$$

where: S_c = Combined Stress

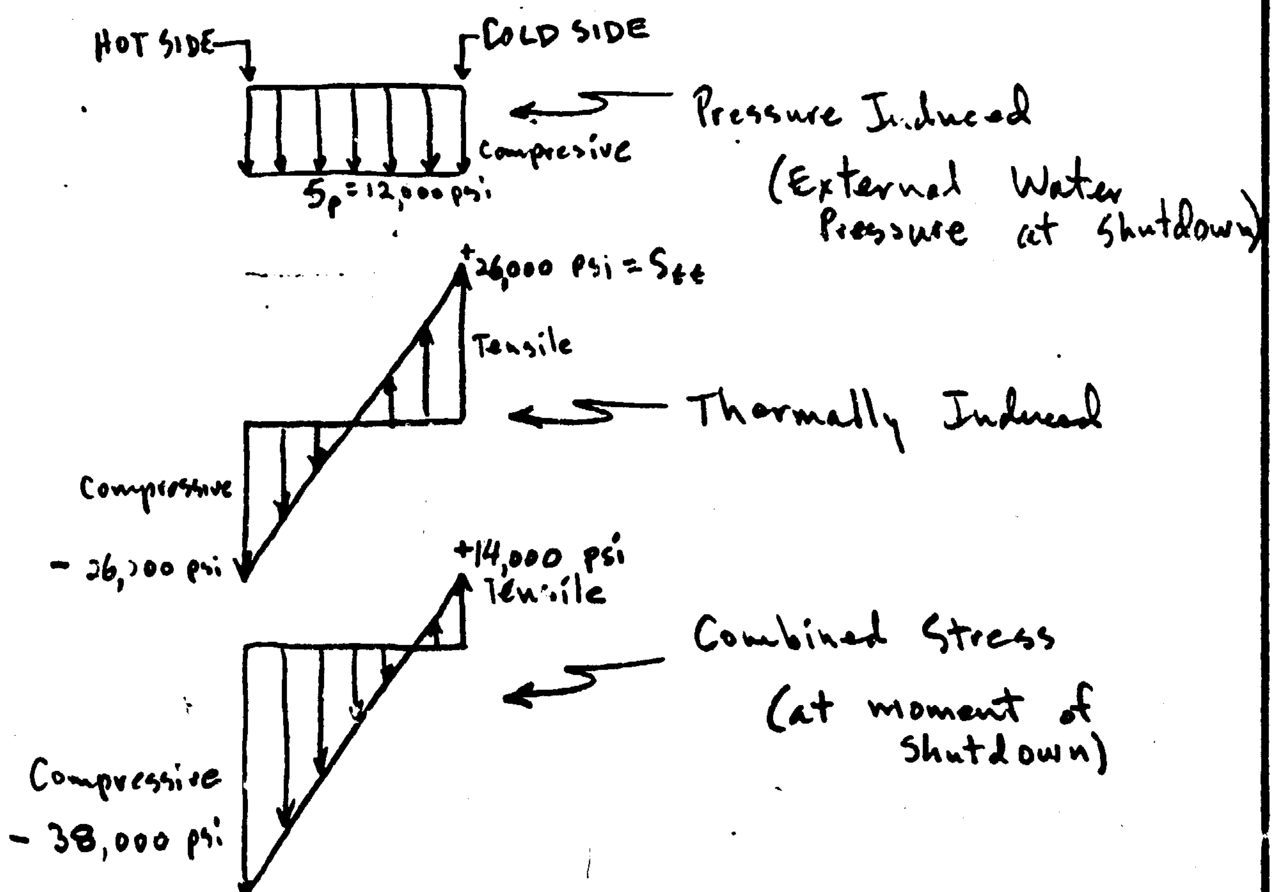
S_p = Pressure Induced Stress

S_{tt} = Thermally Induced Stress

At Z':

$S_{tt} = 26{,}000$ psi { compressive on hot side (−); tensile on cold side (+) }

$S_p = 12{,}000$ psi compressive at start of shutdown



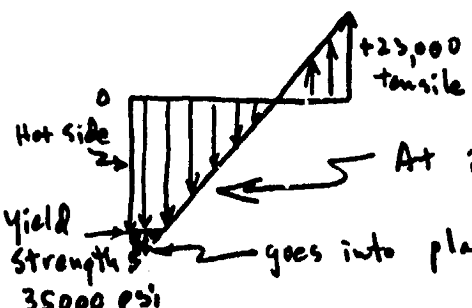
It is significant that the water pressure induces a compressive stress, since this allows the liner to withstand thermal stresses to 35,000 psi in tension corresponding to $\dot{q} = 370$ Btu/sec-ft^2 (S.F. = 1) as shown below:

For instance: Assuming a test run where $\dot{q} = 370$ and P = 1200 psi water pressure. During the test the water pressure = gas pressure, so that the combined stress is 35000 tensile and 35000 compressive. At moment of initiation of shutdown, assume wall temperature differential stays "as is" and gas pressure drops instantaneously to zero. Then combined stresses are:

$S_{tensile} = 35000 - 12000 = 23{,}000$ psi and

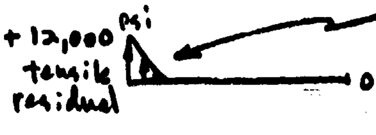
$S_{compressive} = 35{,}000 + 12000 = 47{,}000$ psi (calculated).

Since 47,000 psi exceeds yield point we have local plastic strain.



At initiation of shutdown

goes into plastic strain



After shutdown completed we have 12,000 psi max residual stress.

Thermal Growth of Liner

$\delta = \alpha \Delta T_b L$ in.

where δ = thermal elonation, inches
α = coefficient of expansion in/in-°F
ΔT_b = mean increase in wall temperature from initial temp = 80 F
L = Liner Length, inches

$\alpha = 9.3 \times 10^{-6}$ in/in-°F for Zirconium Copper

$\Delta T_b = (T_{Wc} - T_b) + \left(\frac{T_{WH} + T_{Wc}}{2}\right)$

where T_b = initial wall temp, °F
T_{Wc} = Water-Side Wall Temp, °F
T_{WH} = Gas-Side Wall Temp, °F

$T_b = 80$ °F

$T_{Wc} = 350$ °F at $\dot{q} = 275$ Cond Z'

$T_{WH} = 560$ °F at $\dot{q} = 275$ Cond Z'

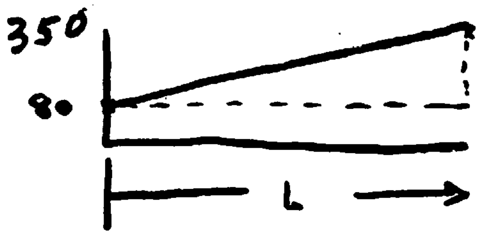
Assuming most <u>conservative</u> case where $\dot{q} = 275$ along entire length of liner:

$\Delta T_b = (350 - 80) + \left(\frac{560 - 350}{2}\right) = 270 + \frac{210}{2}$
$= 365$ °ΔF

L = 54"

$$\delta = (9.3 \times 10^{-6})(365)(54)$$
$$= .183 \text{ inches expansion}$$

For Linear Temp Rise:



$\delta = \frac{.183}{2} = .091$ in → This is most likely expansion at condition 2'

Allow for excess $\dot{q}$ to $\dot{q} = 400$ at full temp for entire length. (Very Conservative)

$$\Delta T_s = (370-80) + \frac{670-370}{2} = 290 + 200 = 490$$

$\delta = (9.3 \times 10^{-6})(490)(54) = .246$ in expansion

Let expansion capability = 3/8 to 1/2 inch. on detailed design.

Radial Expansion of the Combustor Liner

Analyses have been made to determine the radial thermal expansion of the liner and the corresponding clearances required. Radial expansion results from heating of the copper liner from 80 F initial temperature to a final average wall temperature. At specification condition Z', with nominal water coolant passage velocity, V = 14 ft/sec, and heat flux, $\dot{q}$ = 275 BTU/sec-ft^2, we have an approximate average copper temperature rise of ΔT=365 ΔF.

Radial Expansion of Liner at Instrumentation Port Locations

Access ports for instrumentation are located at Section B-B as shown on GASL drawing HE 1702 (NASA drawing No. L522238). A clearance space must be provided between the inner and outer liners to accomodate thermal radial expansion of the inner liner. The average temperature rise of ΔT = 365 ΔF was used to determine the expansion.

The diametral radial thermal expansion is approximated by:

$$\delta_d = \alpha \, \Delta T_b \, D$$

where:

δ_d = diametral growth, inches

α = coefficient of thermal expansion, in/in-$^\circ$F

ΔT_b = average bulk temperature rise of copper liner, Δ°F

D = diameter of liner, inches

$$\delta_d = (9.3 \times 10^{-6})\ (365)\ (10.0)$$

$$= .034 \text{ inches (diametral growth)}$$

The design will provide for .040 inch diametral clearance at Section B-B.

Radial Expansion of Liner Downstream Flange

The liner downstream sealing flange is shown on GASL drawing HE 1700 (NASA drawing No. L-522236).

To determine the radial expansion at the downstream sealing flange, the temperature profile through the flange must be known.

A graphical method found in Reference IV-10, was used to approximate (for design purposes) the temperature profile. The method requires sketching into the flange region a net of isothermal-adiabatic lines, carrying out the construction by trial and error until the criterion that these lines meet at right angles at all points of intersection is satisfied.

The results of this approximation are shown in Figure IV-10 for a heat flux to the liner wall assumed constant at $\dot{q}$ = 275 BTU/sec-ft^2 and with the water flow area restricted so as to obtain a coolant velocity of V_w = 40 ft/sec in this region. The coolant film coefficient in this curved-entrance type flow region is approximated from the method of Jeschke for helical flow found in Reference IV-11:

$$\frac{h'}{h} = 1 + 3.5 \frac{D}{D_{He}}$$

where:

h'/h = ratio of coolant heat transfer coefficient in helical flow (curved) to that in a straight pipe

D = pipe diameter (equivalent diameter, D_e, used), inches

D_{He} = helix diameter (2 x corner radius used), inches

To obtain V = 40 ft/sec, coolant passage must be reduced to:

$$t = .25 \left(\frac{14}{20}\right) = .088 \text{ in}$$

$$D_e = 2 \ (.088) = .176 \text{ in}$$

$$D_{He} = 2 \times .25 = .50$$

$$\frac{h'}{h} = 1 + 3.5 \frac{(.176)}{(.50)} = 1 + 1.23$$

$$\approx 2.0$$

For the value $h'/h \approx 2$, we can use the water cooling characteristic curves of Figure IV-4 for V = 80 to obtain approximate wall temperatures at $\dot{q} = 275$, $V \approx 80$

$$T_{\substack{WALL\\COLD\ SIDE}} = 160^{o}F$$

$$T_{\substack{WALL\\HOT\ SIDE}} = 360^{o}F$$

$$T_{\substack{WALL\\AVERAGE}} = \frac{160+360}{2} = 260^{o}F$$

Observation of Figure IV-10 indicates that an isotherm of $T = 260^{o}F$ passes through the flange (and O-ring) section, and this value is used as the average temperature for determining a copper bulk temperature rise.

$$T_B = 260 - 80 = 180\ \Delta^{o}F$$

the radial expansion is approximated as

$$\delta_d = \alpha\ \Delta T_B D$$

$$= (9.3 \times 10^{-6})\ (180)\ (11.0)$$

$$= .018 \text{ inches (diametral)}$$

Therefore, in order to provide for radial expansion at the liner downstream sealing flange, a radial clearance (diametral) of 0.018 can be provided. Static O-ring sealing can be attained with this clearance.

BUCKLING ANALYSIS:

$$P_{cr} = \frac{2E}{1-\nu^2}\left(\frac{t}{d}\right)^3 \quad psi$$

where

P_{cr} = Critical buckling pressure for Cylinder with high L/D, psi

E = Modulus of Elasticity, psi

t = wall thickness, in

d = diameter, in

$E = 19.3 \times 10^6$

$\nu = .3$ (Poisson's Ratio)

t = .5 in

D = 10 in

$$P_{cr} = \frac{2(19.3\times10^6)}{1-(.3)^2}\left(\frac{.5}{10}\right)^3$$

$$= 5300 \text{ psi}$$

P_w = 1200 psi max water pressure, external,

Buckling Safety Factor,

$$S.F. = \frac{5300}{1200} = 4.42 \quad \text{good}$$

Values of SF = 3 to 4 are acceptable

IV.B. 2. Pressure Enclosure:

$$t = \frac{PR}{SE - .6P}$$

Circumferential Stress
ASME Code - Section VIII
Para. UG-27

where: t = min required shell thickness, in.
P = design pressure, psi
R = Inside radius, in
S = maximum allowable stress, psi
E = joint efficiency

P = 1200 psi
R = 8 inches
S = 15,000 psi
E = 1 (no joints, or radiographed butt joints)

$$t = \frac{(1200)(8)}{15000(1) - .6(1200)} = \frac{9600}{15000 - 720} = \frac{9600}{14,280}$$

= .673 inches req'd

Present Configuration has t > .673 at all locations.

Openings have reinforcing studded outlets

(For Final Design of Pressure Enclosure, see "Stress Analysis Report for GAS-L Vessel" by National Forge Co., which follows.)

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NATIONAL FORGE CO.

BY Winn DATE 6/1/69 SUBJECT ________ SHEET NO. 1 OF ____
CHKD. BY ____ DATE ____ JOB NO. ____
APPROVED ____ ENG. NO. ____

STRESS ANALYSIS REPORT

FOR GASL VESSEL

CUSTOMER ORDER NO. 3594

N.F. SHOP ORDER 53-7117

(I.) DESIGN BASIS

(A) APPLICABLE DOCUMENTS:

GASL SPEC. NO. 3982. REVISION: A.

ASME BOILER & PRESSURE VESSEL CODE

SECTION VIII. DIV. 1.

(B) DESIGN DATA:

(1) DESIGN PRESSURE. 1,200 PSIG AT 650°F

(2) TEST PRESSURE: 1,800 PSIG AT AMB.

(3) MATERIAL OF VESSEL

SA-266-II.

TENSILE STR. MIN. 70,000 PSI

ALLOWABLE STRESS 17,500 PSI AT

-20°F - 650°F.

(4) MATERIAL OF BLIND FLANGES.

SA 105- II.

TENSILE STR. (MIN) 70,000 PSI

ALLOWABLE STRESS. (MAX) 17,500 PSI
AT -20°F - 650°F

(5) MATERIAL OF STUDS & NUTS.

NUTS: A 194 - 2H

STUDS: A 193 - B7

ALLOWABLE STRESS (MAX) 20,000 PSI
AT -20°F - 650°F

(II) VESSEL BODY DESIGN

(A) MINIMUM THICKNESS SECTION VIII, UG27.(c)(1)

$$t = \frac{PR}{SE - .6P}$$

WHERE

$P = 1200$ PSIG

$R = 7''$

$S = 17,500$ PSI

$E = 1$

$$\therefore t = \frac{1200 \times 7}{17500 \times 1 - .6 \times 1200}$$

$$= .72''$$

AT THE LARGE END, $R' = 8.75''$

$$\therefore t' = \frac{1200 \times 8.75}{17500 \times 1 - .6 \times 1200}$$

$$= .9''$$

ACTUALLY THIS VESSEL IS GOING TO BE MADE OF 27.75" O.D. ITS ACTUAL THICKNESS WOULD BE $\frac{1}{2}(27.75 - 17.5) = 5\frac{1}{8}''$ (MIN.)

THE MINIMUM SHELL THICKNESS CAN ALSO BE CHECKED BY FORMULA OF APPENDIX I OF CODE SECTION VIII.

$$Z = \frac{SE + P}{SE - P}$$

$$= \frac{17500 \times 1 + 1200}{17500 \times 1 - 1200}$$

$$\doteq 1.15$$

$$R = 8.75''$$

$$t = R(Z^{\frac{1}{2}} - 1)$$

$$= 8.75 \times (1.15^{\frac{1}{2}} - 1)$$

$$= .64'' \quad << 5\tfrac{1}{8}'' \text{ THICKNESS.}$$

(B) STRESS INTENSITY.

THE VESSEL IS GOING TO BE MADE OF 27.75" OD REQUIRED BY THE CUSTOMER, THE STRESS LEVEL WOULD BE VERY LOW. CODE SECTION VIII DOES NOT REQUIRE STRESS INTENSITY CALCULATION. YET, WE STILL CAN ESTIMATE IT AS FOLLOWS:

AT THE LARGER VOID END, ID = 17.5".

$$w = \frac{OD}{ID}$$

$$= \frac{27.75}{17.5}$$

$$= 1.585$$

$$w^2 = 2.52$$

$$\therefore \sigma_t = \frac{w^2+1}{w^2-1} \cdot P$$

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CHKD. BY____ DATE____ JOB NO.____
APPROVED____ INQ. NO.____

$$\therefore \sigma_t = \frac{2.52+1}{2.52-1} \times 1{,}200$$

$$= 2{,}770 \text{ PSI}$$

$$\sigma_l = \frac{1}{W^2-1} P$$

$$= \frac{1}{2.52-1} \times 1{,}200$$

$$= 790 \text{ PSI}$$

$$\sigma_r = -1{,}200 \text{ PSI.}$$

$$S = 2770 + 1200$$

$$= 3{,}970 \text{ PSI.}$$

APPARENTLY. THE STRESS LEVEL IS VERY LOW.

(III) BLIND FLANGES DESIGN

(A) LARGER VOID END.

CODE SECTION' VIII UG-34 (2).

$$t = d\sqrt{\frac{C \cdot P}{S} + 1.78 \frac{W \cdot h_G}{S d^3}}.$$

WHERE

$$d = G$$

$$= 18.203''$$

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APPROVED______ INQ. NO.______

$$\overline{W} = \frac{\pi}{4}(18.203)^2 \times 1{,}200$$

$$= 3.13 \times 10^5 \text{ lbs}$$

$$h_G = \frac{1}{2}(24.25 - 18.203)$$

$$= 3.024$$

$$C = .3$$

$$\therefore t_1 = 18.203\sqrt{\frac{.3 \times 1200}{17500} + 1.78\,\frac{3.13 \times 10^5 \times 3.024}{17500 \times 18.203^3}}$$

$$= 18.203\sqrt{.0206 + .0160}$$

$$= 3.48''$$

USE 4" THICK

THE CUSTOMER NEED IT BE $\frac{1}{2}$ THICKER THAN THAT REQUIRED BY CODE.

∴ USE $4\frac{1}{2}''$ THICK FORGED PLATE.

(B) SMALLER VOID END.

AT THIS END

$$d = G$$

$$= 15.703''$$

C-2

NATIONAL FORGE CO.

BY____ DATE____ SUBJECT____ SHEET NO. 7 OF____
CHKD. BY____ DATE____ JOB NO.____
APPROVED____ ENG. NO.____

$$W = \frac{\pi}{4}(15.703)^2 \times 1200$$

$$= 2.33 \times 10^5 \text{ lbs}$$

$$h_G = \frac{1}{2}(24.25 - 15.703)$$

$$= 4.274''$$

$$C = .3.$$

$$\therefore t_2 = 15.703\sqrt{\frac{.3 \times 1200}{17500} + 1.78\,\frac{2.33 \times 10^5 \times 4.274}{17500 \times 15.703^2}}$$

$$= 15.703\sqrt{.0206 + .0262}$$

$$= 3.4''$$

USE $3\frac{1}{2}''$ FORGED PLATE.

(C) BLIND FOR 5" DIA OPENINGS

$$d = G$$

$$= 6.53''.$$

$$W = \frac{\pi}{4}(6.53)^2 \times 1200$$

$$= 4.04 \times 10^4 \text{ lbs.}$$

$$h_G = 2.23''$$

$c = .3$.

$$t_3 = 6.53\sqrt{\frac{.3 \times 1200}{17500} + 1.78\,\frac{4.04 \times 10^4 \times 2.23}{17500 \times 6.53^3}}$$

$$= 6.53\sqrt{.0206 + .0327}$$

$$= 1.51''$$

COMMERCIAL BLIND OF 900 lbs CATEGORY FOR 5" DIA HOLE HAS A THICKNESS OF 2" AT THE RIM & 2¼" AT CENTER. THIS IS GOOD FOR THE PURPOSE.

(D) BLIND FOR 4" DIA OPENINGS

$d = G$
$= 5.468''$

$$W = \frac{\pi}{4}(5.468)^2 \times 1200$$

$$= 2.82 \times 10^4 \text{ lbs.}$$

$h_G = 1.89$.

$c = .3$

$$t_4 = 5.468\sqrt{\frac{.3\times1200}{17500} + 1.78\,\frac{2.82\times10^4\times1.89}{17500\times5.468^3}}$$

$$= 5.468\sqrt{.0206+.0332}$$

$$= 1.27''$$

COMMERCIAL BLIND OF 900 lbs CATEGORY FOR 4" OPENING HAS A THICKNESS OF 1.75" AT THE RIM & 2" THICKNESS AT CENTER. THIS IS GOOD FOR THE PURPOSE.

(IV) STUDS DESIGN.

(A) FOR BOTH ENDS. — "O" RING SEALING.

$$W_{m_1} = \frac{\pi}{4} G^2 P$$

(1) LARGER VOID END.

$$W_{m_1} = 3.13\times10^5 \text{ lbs.}$$

$$A_m = \frac{3.13\times10^5}{20{,}000}$$

$$= 15.65 \text{ IN}^2.$$

BUT, $A_b = 20 \times 1.78$ $(1\frac{1}{8} - 8UN)$

$= 35.5\ \square''$

$\therefore A_b >> A_m$ O.K.

(2) SMALLER VOID END

$W_{m_1} = 2.33 \times 10^5$ lbs.

$$A_m = \frac{2.33 \times 10^5}{20,000}$$

$= 11.65\ \square''$

BUT, $A_b = 20 \times 1.78$

$= 35.5\ \square''$

$\therefore A_b >> A_m$ O.K.

(B) FOR 5" DIA OPENING (GASKET: FLEXITALLIC $5\frac{3}{4} \times 7\frac{1}{16} \times .175$)

$$W_{m_1} = \frac{\pi}{4} G^2 p + 2\pi b G m P$$

$$= 4.04 \times 10^4 + 2\pi b G m P$$

WHERE

$N = .78''$

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CHKD. BY ______ DATE ______ JOB NO. ______
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$b_0 = .39''$

$b = \frac{\sqrt{b_0}}{2}$

$= .311$

$m = 3.$

$\therefore W_{m_1} = 4.04 \times 10^4 + 2\pi \times .311 \times 6.63 \times 3 \times 1200$

$= 4.04 \times 10^4 + 4.62 \times 10^4$

$= 8.66 \times 10^4$ lbs.

$A_m = \frac{8.66 \times 10^4}{20{,}000}$

$= 4.33 \square''.$

$A_b = 8 \times 1.000$

$= 8 \square''.$

$A_b > A_m.$ OK.

DURING TEST. $P_i = 18.00$ PSIG

$W'_{m_1} = 8.66 \times 10^4 \times 1.5$

$= 1.3 \times 10^5$ lbs.

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BY ______ DATE ______ SUBJECT ______ SHEET NO. 12 OF ______
CHKD. BY ______ DATE ______ JOB NO. ______
APPROVED ______ INQ. NO. ______

$$A_m' = \frac{1.3\times10^5}{20{,}000}$$

$$= 6.5 \ \square''$$

$$A_b > A_m' \qquad \text{O.K.}$$

(C) FOR 4" DIA OPENING (GASKET FLEXITALLIC $4\frac{7}{8}\times6\frac{3}{16}\times.125$)

$$W_{m_1} = 2.82\times10^4 + 2\pi b G m P$$

WHERE $N = .728''$

$$b_o = .364''$$

$$b = \frac{\sqrt{.364}}{2}$$

$$= .301$$

$$m = 3$$

$$\therefore W_{m_1} = 2.82\times10^4 + 2\pi\times.301\times5.468\times3\times1200$$

$$= 2.82\times10^4 + 3.71\times10^4$$

$$= 6.53\times10^4$$

$$A_m = \frac{6.53\times10^4}{20{,}000}$$

$$= 3.27 \ \square''$$

$A_b = 8 \times .79$

$= 6.33 \ \square''$

$\therefore A_b > A_m$ O.K.

DURING TEST $P_i = 1{,}800$ PSIG

$\therefore W'_{m1} = 6.53 \times 10^4 \times 1.5$

$= 9.78 \times 10^4$

$\therefore A'_m = \dfrac{9.78 \times 10^4}{20000}$

$\doteq 4.89 \ \square''$

$A_b > A'_m$ O.K.

(V) CONCLUSIONS:

(1) THE VESSEL BODY IS OVER DESIGNED SO THAT, AFTER BEING FLATTED THE OUTER SURFACE OF 4" & 5" OPENINGS, THE THICKNESS AVAILABLE IS STILL MORE THAN REQUIRED BY CODE, NAMELY, .72" AT BORE OF 14" ID, & .9" AT BORE OF $17\frac{1}{2}$" ID.

NATIONAL FORGE CO.

BY ______ DATE ______ SUBJECT ______ SHEET NO. 14 OF ______
CHKD. BY ______ DATE ______ JOB NO. ______
APPROVED ______ INQ. NO. ______

IT IS NECESSARY THAT THE STUDS SHOULD NOT PENATRATE INTO THE VESSEL WALL BEYOND THE MINIMUM THICKNESS LIMIT.

(2) BLIND FLANGES AND STUDS COMPLY WITH THE CODE STANDARD.

(4) THE DESIGN IS CONSIDERED TO BE SAFE.

BY FW DATE 2-21-67 SUBJECT Bolt Torque SHEET NO. 1 OF 3
CHKD. BY ____ DATE ____ JOB NO. 53-7117-30

P = 1800 PSI

(R1-9N) $b = b_o = 1/4''$ $N = 0.5''$

(R1-9N) O.D = $7\,5/16''$

I.D = $6\,5/16''$

R1-9L O.D = $6\,3/16''$

I.D = $5\,3/16''$

$b = b_o = 1/4$ $N = 0.5''$

Cover for 5" Dia Penetration

R1-9N $G = OD - 2b = 6.8125''$

R1-9L $G = OD - 2b = 5.6875''$

R1-9N $N = 8,\ 1\tfrac{1}{4}''\phi$ M.D = 1.0945"

R1-9L $N = 8\ \ 1\tfrac{1}{8}''\phi$ M.D = .9695

$W_{m_1} = H + H_p = .785\,G^2P + (2b \times 3.14\,G\,m\,P)$

R1-9N

$W_{m_1} = .785(6.8125)^2(1800) + (2 \times \tfrac{1}{4} \times 3.14 \times 6.8125 \times 3 \times 1800)$

$W_{m_1} = 66{,}400 + 57{,}900 = 124{,}300^{\#}$

$T = .2DW = \dfrac{.2(1.0945)(124{,}300)}{8} = 3400$ lb in = 283 lbft

Load Per Bolt: 15,538#

Stress Area: (1¼-8) 1.0 in²

$\sigma = \dfrac{15{,}538}{1} = 15{,}538$ psi Tension

COVER FOR 4" DIA PENETRATION

R1-9L

$W_{m_1} = .785(5.6875)^2 1800 + (2(\frac{1}{4}) \times 3.14 \times 5.6875 \times 3 \times 1800)$

$W_{m_1} = 45{,}500 + 48{,}400$

$= 93{,}900^{\#}$

$T = \frac{.2(.9695)\ 93{,}900}{8} = 2{,}275$ lb in $= 190$ lb ft

LOAD PER BOLT = $11{,}700^{\#}$

STRESS AREA: $(1\frac{1}{8}-8)$ $.790 \, in^2$

$\sigma = \frac{11{,}700}{.790} = 14{,}850$ psi TENSION

END COVERS

GASKET: "O" RING $m=0$

$G = 16.025 - 2(.161) = 15.703$
$P = 1800$ psi

$W_{m_1} = H + H_p = 0.785G^2P + (2b \times 3.14\, G\, m\, P)$

$= .785(15.703)^2(1800) + 0$

$W_{m_1} = 348,000^{\#}$

(20) 1 5/8-8 bolts

LOAD PER BOLT = 17,400#

STRESS AREA: 1.695 in²

$\sigma = \frac{17,400}{1.695} = 10,270$ psi TENSION

TORQUE:

$T = .2(1.47)\,17,400 = 5,120$ lb in $= 427$ lbft

C. PRESSURE LOSSES

C.1. PRESSURE DROP ACCROSS LINER (WATER)

(SEE SKETCH - NEXT PAGE)

FRICTIONAL:

$$\Delta P_f = f\left(\frac{L}{D_e}\right)\frac{\rho V^2}{2g \times 144} \quad \text{PSI}$$

WHERE:

ΔP_f = FRICTIONAL PRESSURE DROP, PSI

f = FRICTION FACTOR

L = LINER LENGTH, IN

D_e = HYDRAULIC DIA. (FOR AN ANNULUS $D_e \approx 2 \times$ WIDTH), IN

ρ = WATER DENSITY, LB/FT³

V = VELOCITY, FT/SEC

g = 32.2 FT/SEC²

$$R_e = \frac{\rho V D_e}{\mu}$$

WHERE:

R_e = REYNOLDS NO.

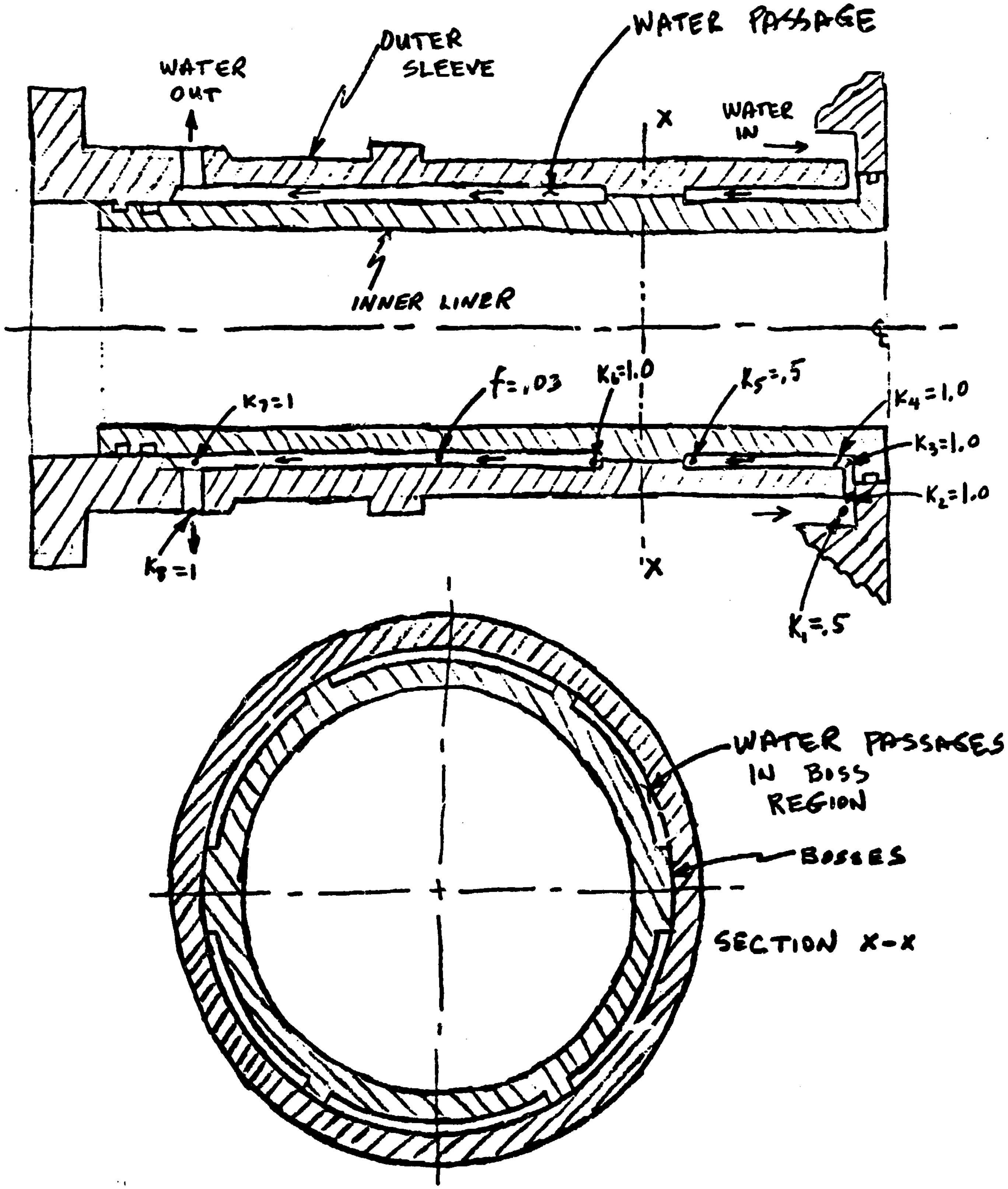
μ = VISCOSITY OF WATER, LB/FT-SEC

$L = 54$ IN

$D_e = 2(.25) = .5$ IN

$\rho = 62.4$ LB/FT³

$\mu = 45.9 \times 10^{-5}$ LB/FT-SEC



WATER PASSAGE CROSS-SECTIONAL VIEW

$$R_e = \frac{\rho V D_e}{\mu} = \frac{(62.4)(14)(.5/12)}{45.9 \times 10^{-5}} = .8 \times 10^5$$

USE: $f \approx .03$

FOR $V = 14$ FT/SEC

$$\Delta P_f = f\left(\frac{L}{D_e}\right)\frac{\rho V^2}{2g \times 144}$$

$$= (.03)\left(\frac{54}{.5}\right)\left[\frac{(62.4)(14)^2}{9.28 \times 10^3}\right] = 3.24\,(1.32)$$

$$= 4.3 \text{ PSI}$$

CONTRACTION & ENLARGEMENT LOSSES

$$\Delta P_K = [\Sigma K]\,\frac{\rho V^2}{2g},$$

WHERE:

K = PRESSURE LOSS FACTOR

The pressure drop at the entry flange, where V = 40, is due to a contraction, $K_1 = .5$, two right angles, $K_2 = 1.0$, $K_3 = 1.0$, and an expansion $K_4 = 1.0$, so that

$$\Delta P_{Flange} = [\Sigma K] \frac{\rho V^2}{2g}$$

where K = pressure loss factor

$$\Delta P_{Flange} = [.5 + 1.0 + 1.0 + 1.0] \left[\frac{(62.4)\ (40)^2}{9.28 \times 10^3} \right]$$

$$= (3.5)\ (10.8)$$

$$= 40 \text{ psi}$$

The pressure drop in the boss region, with a slot height of .188, consists of a contraction loss, a friction loss and an expansion loss. For a water capacity of 350 gpm, the water velocity past the bosses is

$$V = \frac{\dot{w}}{\rho A_w}$$

where:

A_w = water flow area between bosses (ft^2)

A_w = (πD) blockage length) t

$$= \{ (3.14)\ (10.188) - [(4)(1) + (2)(2.5)]\}\ (.188)$$

$$= (32.0-9)\ (.188)$$

$$= 4.33 \text{ in}^2 = .030 \text{ ft}^2$$

Therefore:

$$V = \left(\frac{350}{7.2} \right) \left(\frac{1}{62.4} \right) \left(\frac{1}{.030} \right)$$

$$= 26.9 \text{ ft/sec}$$

The boss region entry plus exit losses are

$$\Delta P = \Sigma K \left(\frac{\rho V^2}{2g}\right) = (K_5+K_6)\frac{\rho V^2}{2g} = (.5+1.0)\ \frac{(62.4)\ (26.0)^2}{9.28\times 10^3}$$

$$= (1.5)\ (4.55) = 6.83 \text{ psi}$$

The frictional loss in the boss area is:

$$\Delta P_f = f\left(\frac{\ell}{D}\right)\frac{\rho V^2}{2g}$$

$$= (.03)\ \left(\frac{3}{.188}\right)\ (4.55)$$

$$= .48\ (4.55)$$

$$= 2.2 \text{ psi}$$

The pressure drop through the boss area is then

$$\Delta P_{BOSS} = 6.8 + 2.2 = 9.0 \text{ psi}$$

The pressure drop at the liner exit consists of a right angle $K_7 = 1$ and an enlargement $K_8 = 1$ at $V = 14$ ft/sec

$$\Delta P_{exit} = \Sigma K \frac{\rho V^2}{2g} = (K_7+K_8)\left(\frac{\rho V^2}{2g \times 144}\right) = (1+1)\ (1.32)$$

$$= 2.6 \text{ psi}$$

The total liner pressure drop is therefore:

$$\Delta P_{\substack{total\\liner}} = \Delta P_{\substack{entry\\flange}} + \Delta P_{\substack{length\\friction}} + \Delta P_{\substack{boss\\region}} + \Delta P_{exit}$$

$$= 40.0 + 4.3 + 9.0 + 2.6$$

$$= 55.9 \approx 56 \text{ psi}$$

IV. C. 2. PRESS. DROP IN WATER FEED LINES

4" Sched 40 piping - flanges

ID = 4.026 in

$$A = \frac{12.73}{144} = .0885 \text{ ft}^2$$

$$V = \frac{\dot{w}}{\rho A} = \frac{69.9}{62.4(.0885)} = 12.7 \text{ ft/sec}$$

$$\frac{\rho V^2}{2g \times 144} = \frac{(62.4)(12.7)^2}{9.28 \times 10^3} = 1.08$$

Say K = 1 inlet - rt ∠

K = 1 outlet rt ∠

$K_{tot} = 2$

$$\Delta P_{feed} = 2(1.08) \approx 2 \text{ psi}$$

TOTAL WATER PRESS. DROP FROM ENCLOSURE INLET TO OUTLET

$$\Delta P_{TOT} = \Delta P_{\substack{liner \\ friction}} + \Delta P_{\substack{liner \\ inl \,\&\, outl}} + \Delta P_{\substack{enclosure \\ inl. \,\&\, outl. \\ fittings}}$$

$$= 8.8 + 12.2 + 2.0$$

$= 23$ psi for $V = 20$ ft/sec along liner

$\dot{w} = 503$ gpm

Annulus = .25 in width

Note that 4" - sched 40 inlet piping size is suitable for any future increase in total flow capacity since present $\Delta P_{\substack{inl \\ \& outl}} = 2$ psi only.

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PRESSURE DROP IN 4" sched 40 pipe lines

$$\Delta P = f\left(\frac{L}{D}\right)\frac{\rho V^2}{2g \times 144} \quad \text{psi}$$

Let L = 100 ft = 1200 in

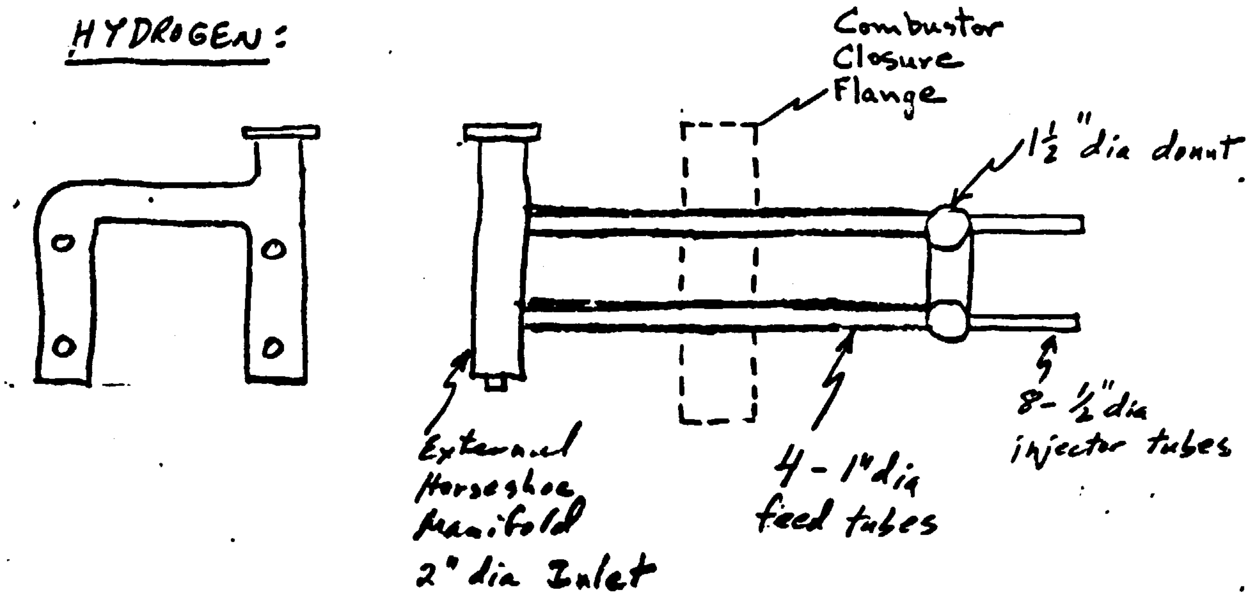
$$f\,\frac{L}{D} = (.03)\left(\frac{1200}{4.026}\right) = 8.94$$

$$\Delta P = \left(f\,\frac{L}{D}\right)\left(\frac{\rho V^2}{2g \times 144}\right) = (8.94)(1.08)$$

= 9.6 psi / 100 ft pipe.

IV.C. 3. PROPELLANT FEED & INJECTOR LINE REQUIREMENTS

HYDROGEN:



$\dot{\omega}_{H_2} = 1.5$ lb/sec @ 1200 psia

$$\rho_{H_2} = \frac{P \times 144}{R \times T} = \frac{1200 \times 144}{768 \times 560} = .40$$

Inlet (Manifold) Use 2" sched 80 pipe

$A = .0205\ ft^2$

$$V = \frac{\dot{\omega}}{\rho A} = \frac{1.5}{(.40)(.0205)} = 183 \text{ ft/sec}$$

Feed Tubes (4): Use 1" OD × .094 wall tube

ID = .812 in

$A = .0036\ ft^2$

$$V = \frac{\dot{\omega}}{\rho A} = \frac{.375}{.4(.0036)} = 261 \text{ Ft/sec}$$

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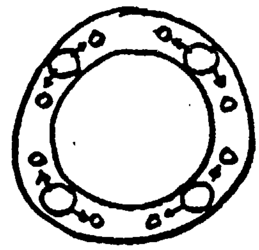
Internal Donut Manifold:

Use 1½ dia tube x .125 wall

$\dot{w} = \frac{1.5}{8} = .188$ lb/sec

$A = .785(1.25)^2 = 1.23$ in^2

$= .853 \times 10^{-2}$ ft^2



$V = \frac{\dot{w}}{\rho A} = \frac{.188}{(.40)(.853 \times 10^{-2})} = 55$ ft/sec (max) to V=0 (min)

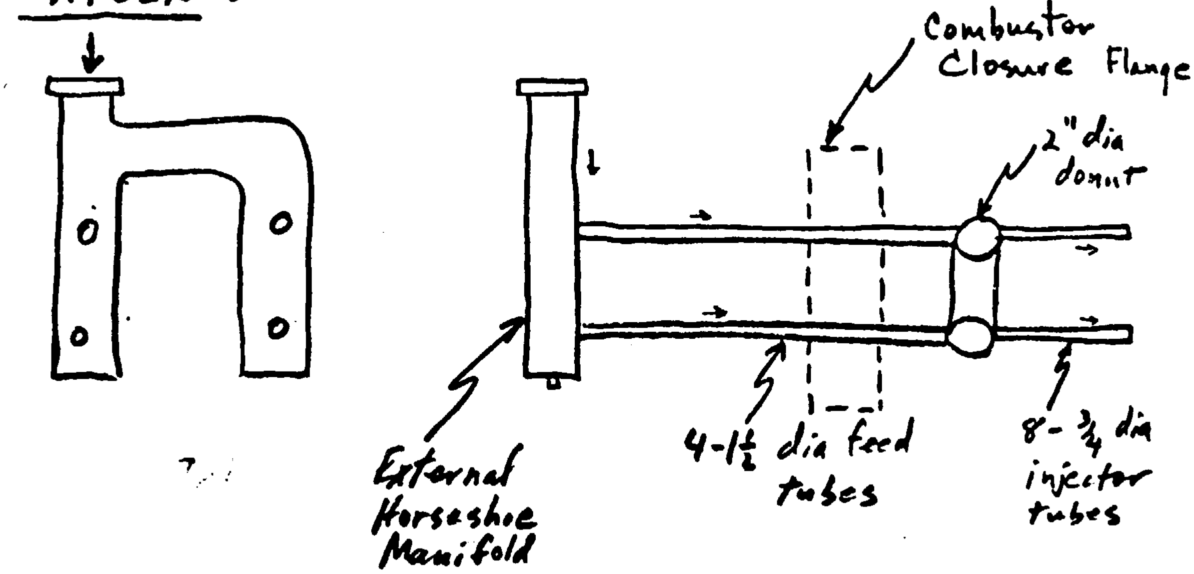
Injector Tubes: (8)

Use .50 dia x .042 wall tubes

$A = \frac{.785(.416)^2}{144} = .94 \times 10^{-3}$ ft^2

$V = \frac{.188}{(.40)(.94 \times 10^{-3})} = 500$ ft/sec

OXYGEN :



Inlet

$\mathring{\omega}_{O_2} = 16$ lb/sec

Use 3" Sched 80 Pipe

OD = 3.500
I.D = 2.900
A = .046 ft²

$$\rho = \frac{P \times 144}{RT} = \frac{1200 \times 144}{48.3 \times 560} = 6.38 \text{ lb/ft}^3$$

$$V_{O_2} = \frac{\mathring{\omega}}{\rho A} = \frac{16.0}{(6.38)(.046)}$$

$= 54.7$ ft/sec

Feed Tubes:

Use 1½ dia - 4 tubes
.125 wall

$A = .00855$ ft²

$$V = \frac{4.0}{6.38 \times .00855} = 73.5 \text{ f/sec}$$

Internal Donut   Use 2" tube × .125 wall

$$A = \frac{.785\,(1.75)^2}{144} = 1.67 \times 10^{-2} \ ft^2$$

$$V = \frac{\dot{w}}{\rho A} = \frac{2.0}{6.38 \times (1.67 \times 10^{-2})} = 18.8 \ ft/sec$$

Injector Tubes (8):   Use 3/4" × .062 wall

$A = .00213 \ ft^2$ ID = .625 in

$$V = \frac{2.0}{(6.38)(.00213)} = 147 \ ft/sec$$

D. WATER COOLED PLUG FOR ACCESS PORT

See Sketch, p. D-2

$\dot{q}_{MAX} = 250$ Btu/sec-ft²

IV.D.1 Calculation of Gas-Side Surface Temp.:

$K = 210$ Btu/HR-ft² (ETP Copper)

$= .058$ Btu/sec-ft²

$x = \frac{.188}{12} = .0157$

$K/x = 3.7$

$\Delta T_w = \dot{q}/(K/x) = 250/3.7 = 67.5$ °R

Assume Velocity at Wall is

$V \cong 40$ ft/sec – radial (estimate)

$T_{w_c} = 210$ °F

$T_{w_H} = T_{w_c} + \Delta T_w = 210 + 68$

$= 278$ °F

WATER COOLING REQUIREMENTS:

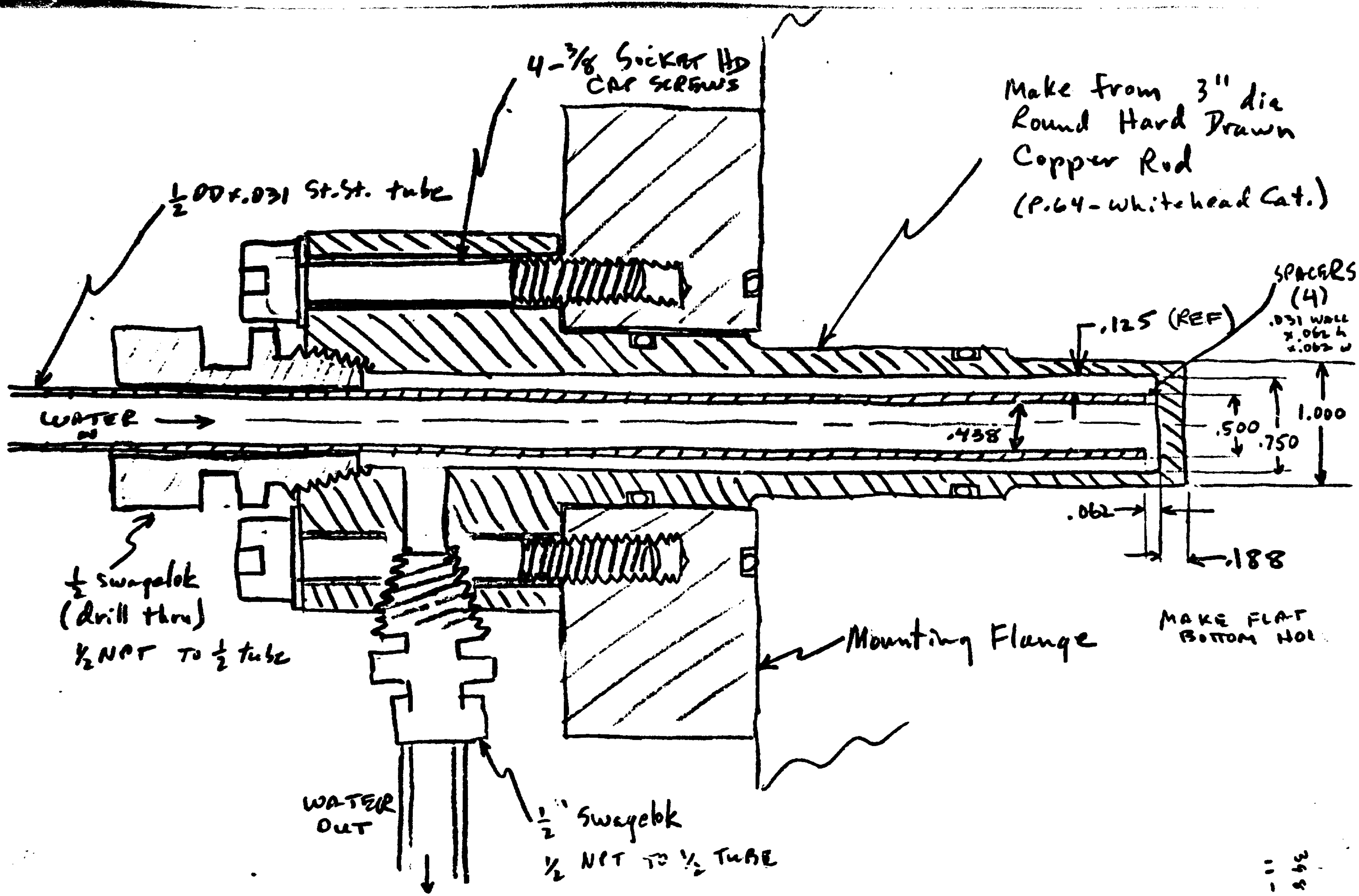
Assume $\dot{W}_w = 10$ gpm $= 1.39$ lb/sec

$V = \frac{\dot{w}}{\rho A}$

$A_{gap} = \pi D t = \frac{3.14(.50)(.062)}{144} = .676 \times 10^{-3}$

$V_{gap} = \frac{1.39}{62.4(.676 \times 10^{-3})} = 33$ ft/sec OK – will have high film coefficient at gap because of 180° u-turn

$\therefore \dot{W} = 10$ gpm is OK.



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WATER COOLED PLUG, ACCESS PORT

WATER VELOCITY

IN COOLING GAP: $V \approx 33$ ft/sec (p. 12)

INLET TUBE:

$$A = \frac{.785\,(.438)^2}{144} = 1.05 \times 10^{-3}\ ft^2$$

$$V_{IN} = \frac{\dot{w}}{\rho A} = \frac{1.39}{62.4\,(1.05 \times 10^{-3})} = 21.2\ ft/sec$$ low enough (good)

OUTLET TUBE:

$$A = \frac{.785\,(.75^2 - .50^2)}{144} = \frac{.785\,(.564 - .250)}{144} = \frac{.246}{144}$$

$$= 1.72 \times 10^{-3}\ ft^2$$

$$V = \frac{1.39}{62.4\,(1.72 \times 10^{-3})} = 12.9\ ft/sec$$ low - good!

D.2. PRESSURE DROP

AT GAP THERE IS A CONTRACTION IN AREA, 2 RT ANGLE TURNS, & AN EXPANSION

ASSUME $K = 1 + 2 + 1 = 4$

$$\Delta P_{gap} = K \frac{\rho V^2}{2g \times 144} = (4)\left[\frac{(62.4)(40)^2}{9.255 \times 10^3}\right] = (4)(10.8)$$

$$= 43.2\ PSIA$$

FOR .500 OD x .438 ID Tube

$$\Delta P = f\left(\frac{L}{D}\right)\left(\frac{\rho V^2}{2g\,144}\right)$$

$V = 21.2$ ft/sec

$\frac{\rho V^2}{2g \cdot 144} = \frac{62.4 (21.2)^2}{9.28 \times 10^3} = 3.02$ psi

For 1 ft $\quad f\left(\frac{L}{D}\right) = .03\left(\frac{12}{.438}\right) = .82$

$\Delta P_{tube} = (.82)(3.02) = 2.5$ psi / ft tube length.

D.3. STRESSES

THERMAL STRESS

$$S_t \cong \frac{E \alpha \Delta T}{2(1-\nu)} = \frac{(17 \times 10^6)(9.8 \times 10^{-6})(68)}{(1.4)}$$

$= 8,000$ psi $\quad$ O.K.

STRESS FROM PRESSURE LOAD ON END CAP

$$S_{r\,max} = \frac{3W}{8\pi m t^2}\left[(3m+1)\right]$$

ROARK, P.154 CASE 1 TABLE X
FORM. FOR STRESS & STRAIN, 3RD ED.

where

S_r = max radial stress psi

W = Total Load = $p \times A = p \times (\pi D^2)$ lb.

$m = \frac{1}{\mu} = \frac{1}{\text{Poisson's Ratio}}$

t = plate thickness, in.

Since equation is for simply supported edges & we have some partial support, the calculation will be conservative.

$A = .785(.75)^2 = .485 \text{ in}^2$

$W = 1200(.485) = 582\#$

$m = \frac{1}{\mu} = \frac{1}{.30} = 3.33$

$t^2 = (.188)^2 = .035$

$S_r = \frac{(3)(582)}{(8)(3.14)(3.33)(.035)}\left[3(3.33) + 1\right]$ (11.0)

$= 6,600 \text{ psi}$

$S_{combined} = S_t + S_r = 8,000 + 6,600 = 14,600 \text{ psi}$

$S_{yield} = \approx 40,000$ psi for hard drawn Cu.

SAFETY FACTOR, $S.F. = \frac{40,000}{14,600} \cong 2.7$

STRESS IN CYLINDRICAL SECTION

$S_{max \, tangential} = \frac{pd}{2t} = \frac{(1200)(1.00)}{2(.125)} = 4,800 \text{ psi}$

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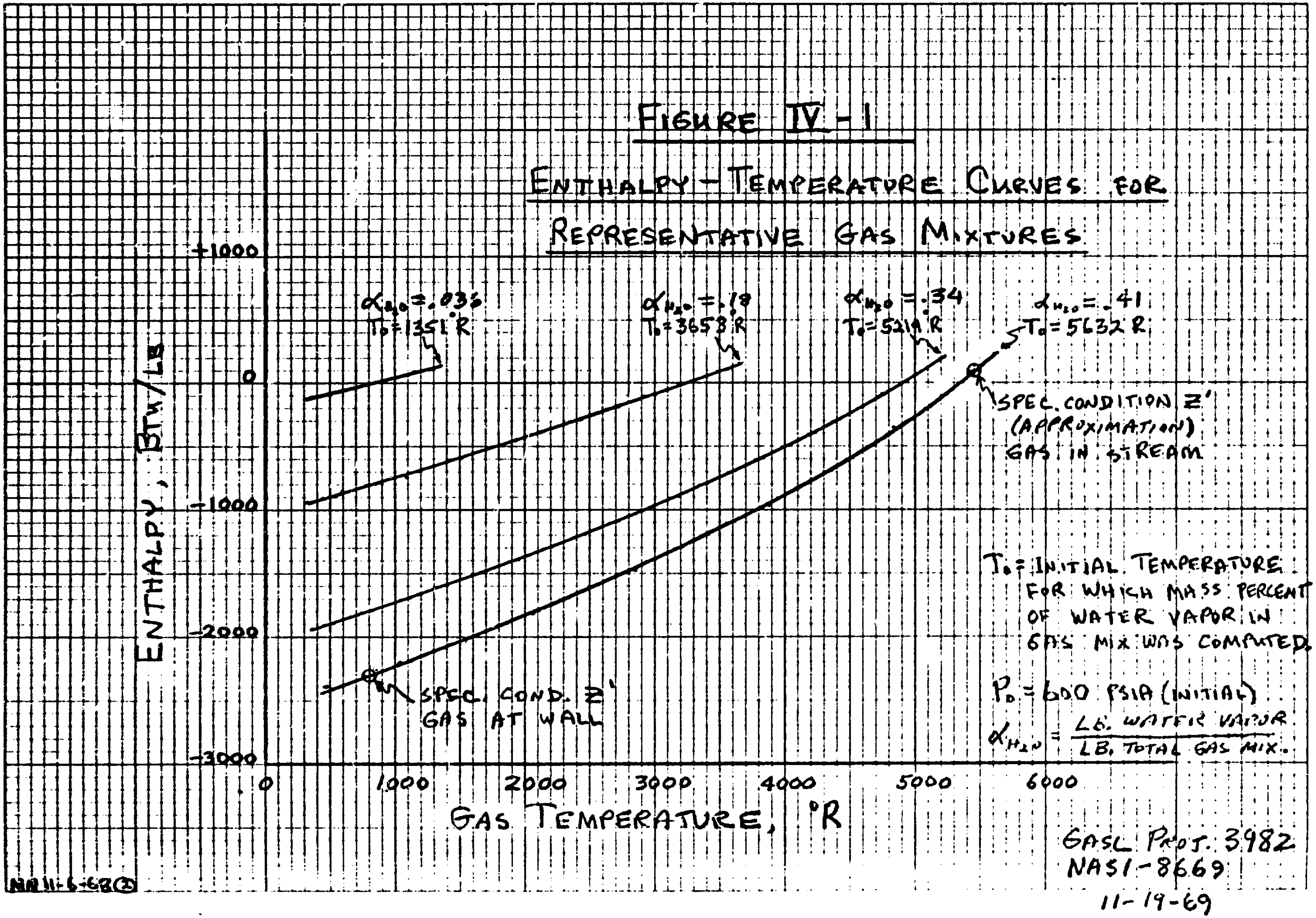

FIGURE IV - 1

ENTHALPY - TEMPERATURE CURVES FOR REPRESENTATIVE GAS MIXTURES

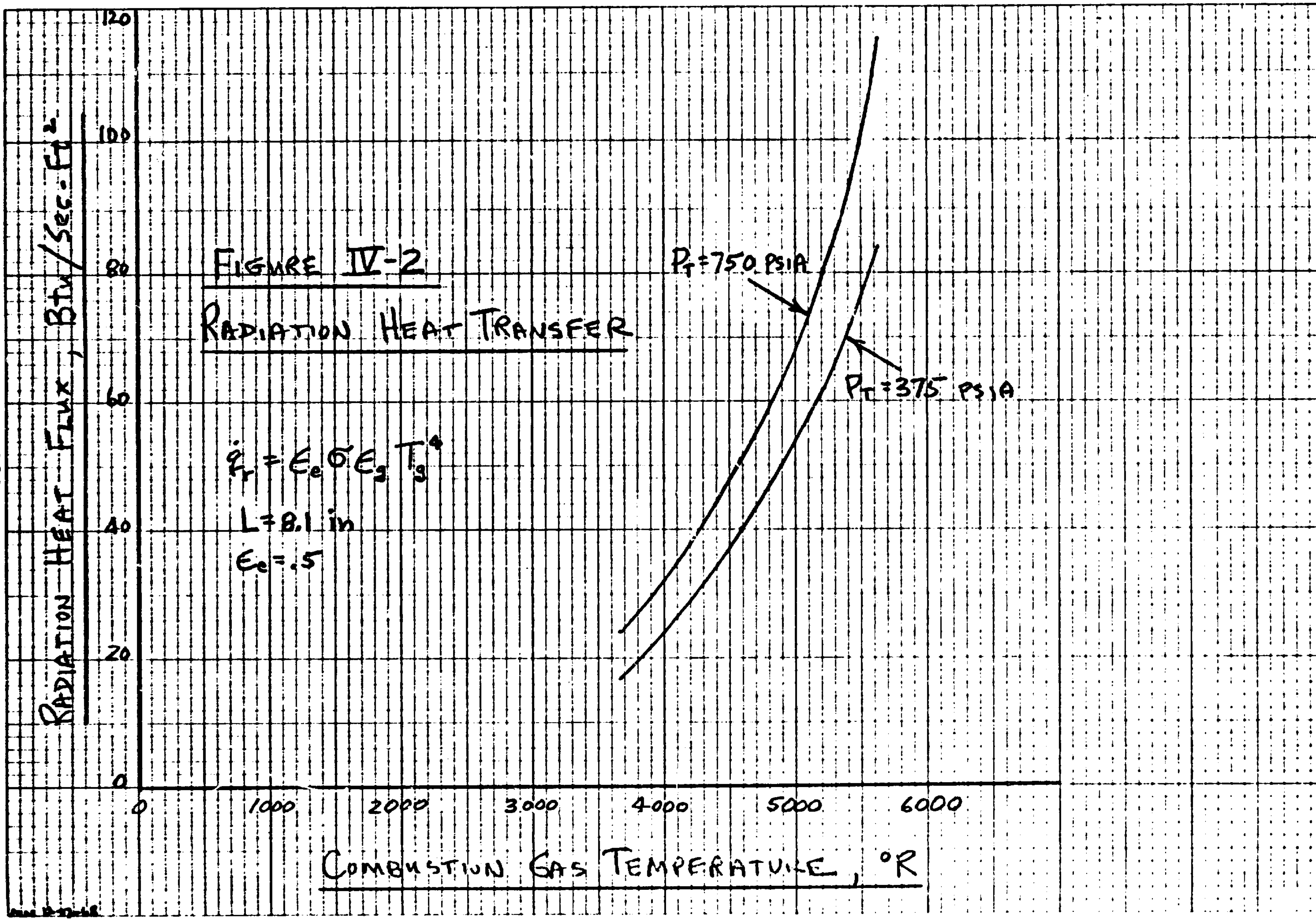

FIGURE IV-2
RADIATION HEAT TRANSFER
$\dot{q}_r = \epsilon_e \sigma \epsilon_g T_g^4$
L = 8.1 in
ϵ_e = .5
P_T = 750 PSIA
P_T = 375 PSIA
RADIATION HEAT FLUX, Btu/Sec-Ft²
COMBUSTION GAS TEMPERATURE, °R
0
20
40
60
80
100
120
1000
2000
3000
4000
5000
6000

3982-2c

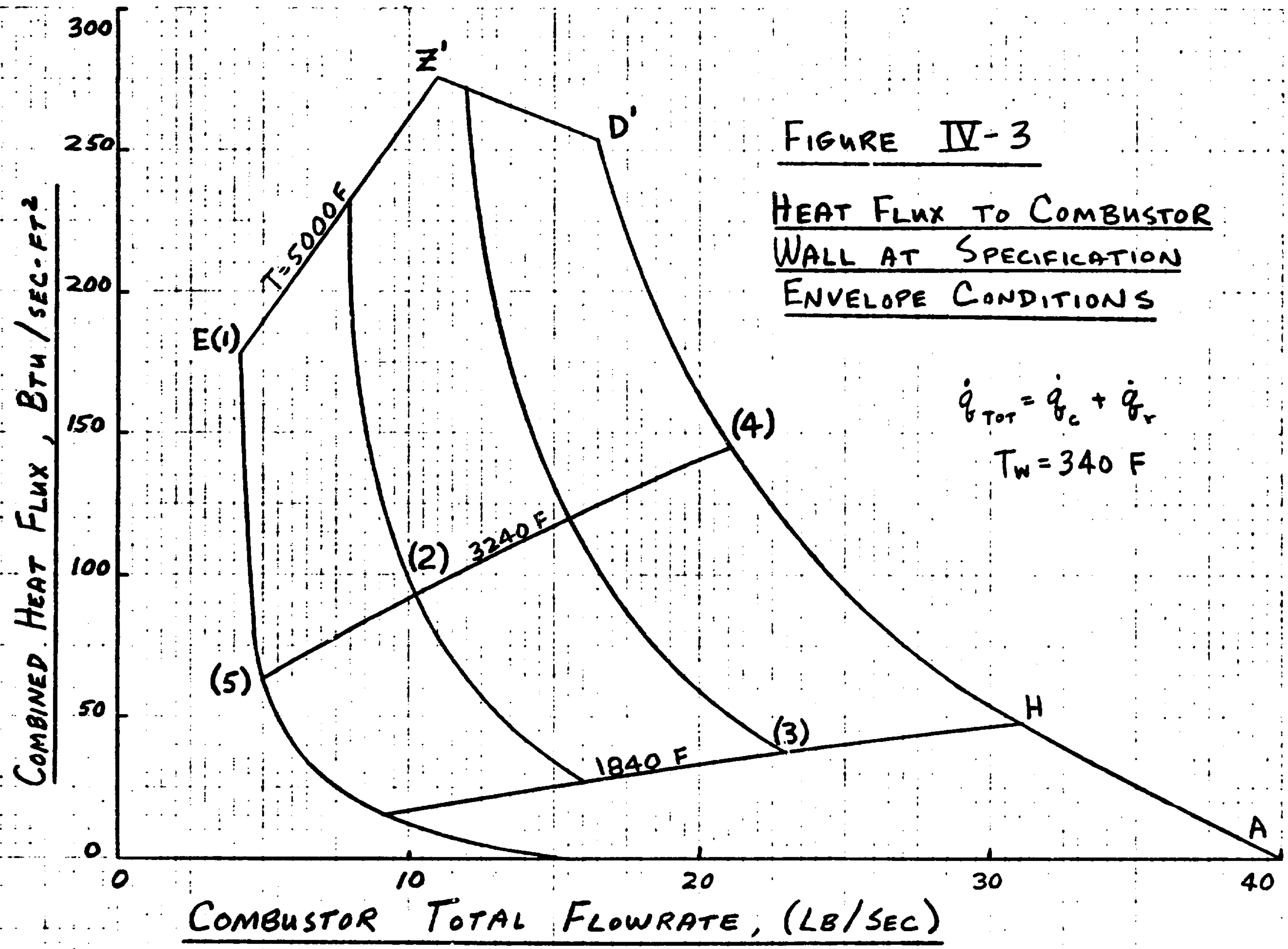

FIGURE IV-3

HEAT FLUX TO COMBUSTOR WALL AT SPECIFICATION ENVELOPE CONDITIONS

12-17-68 REV. MN

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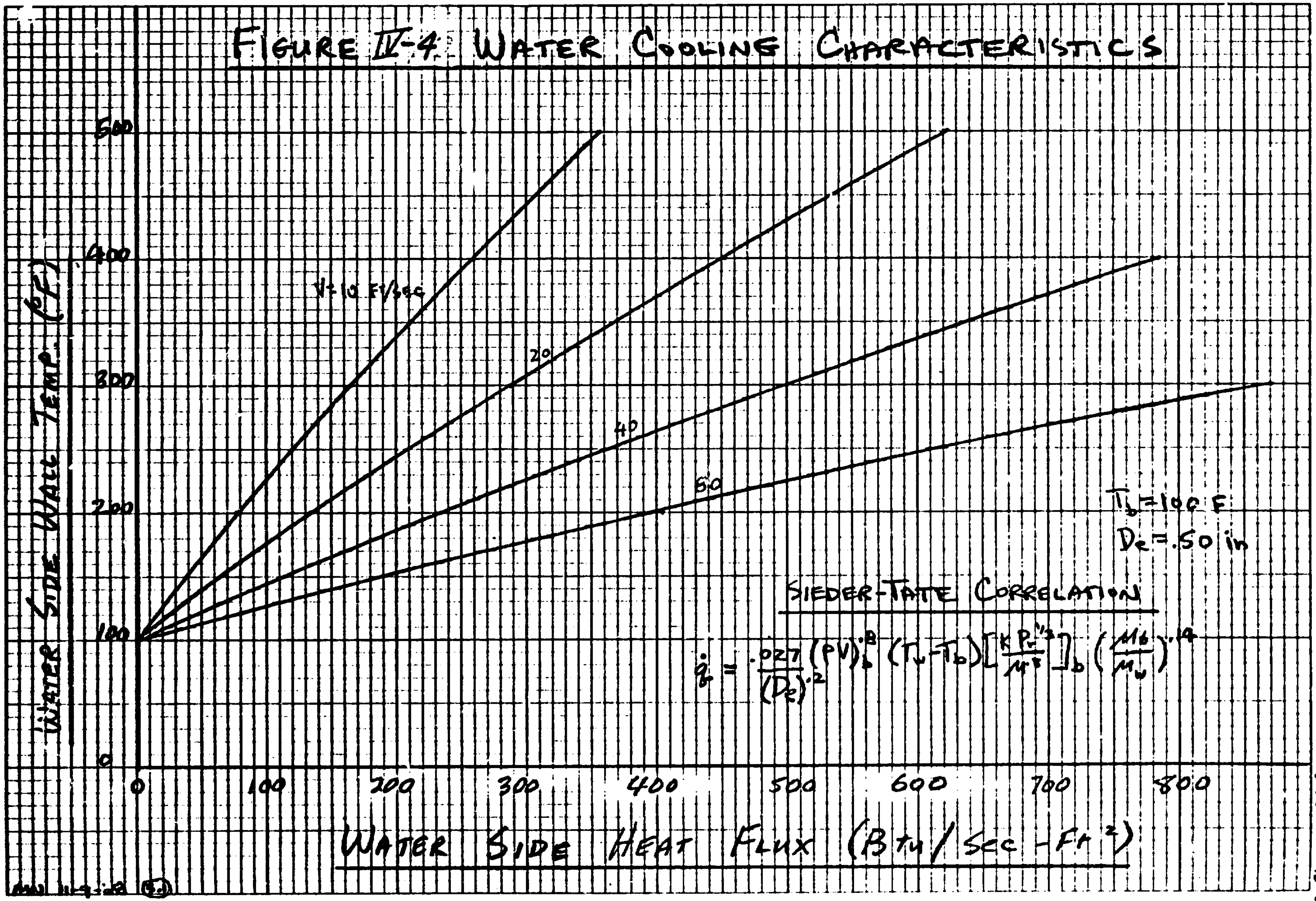


Figure IV-4 Water Cooling Characteristics

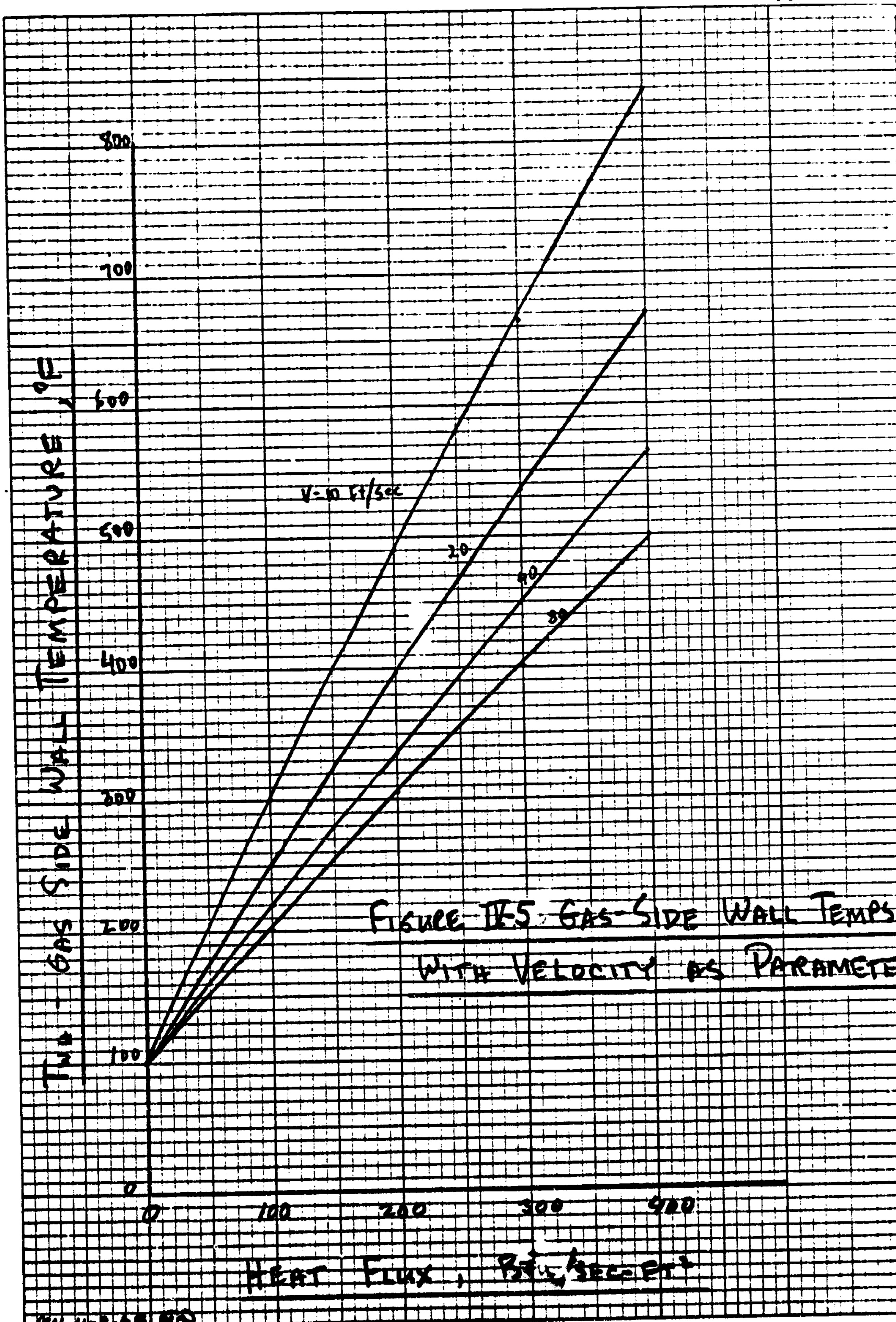


FIGURE IV-5 GAS-SIDE WALL TEMPS. WITH VELOCITY AS PARAMETER

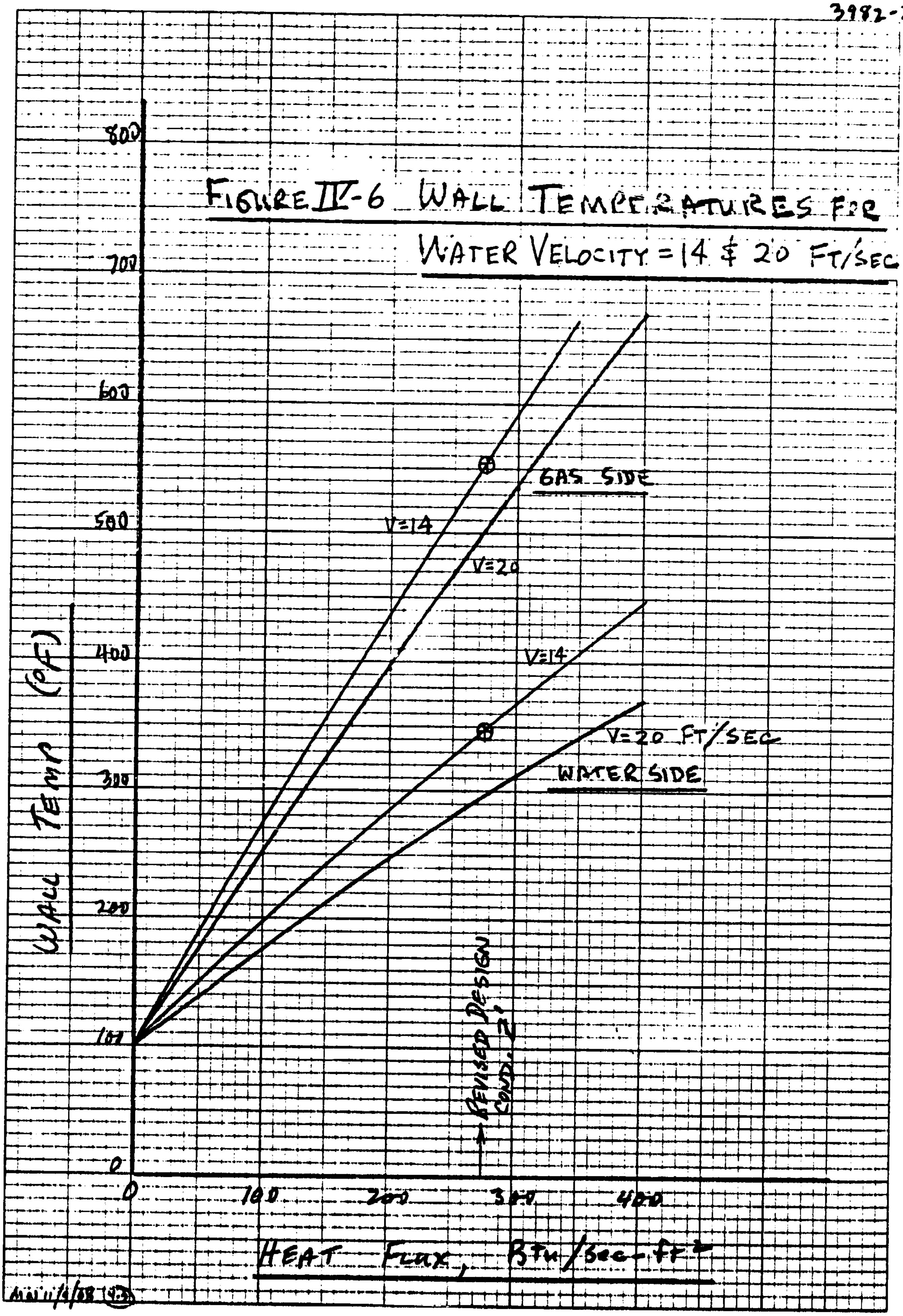
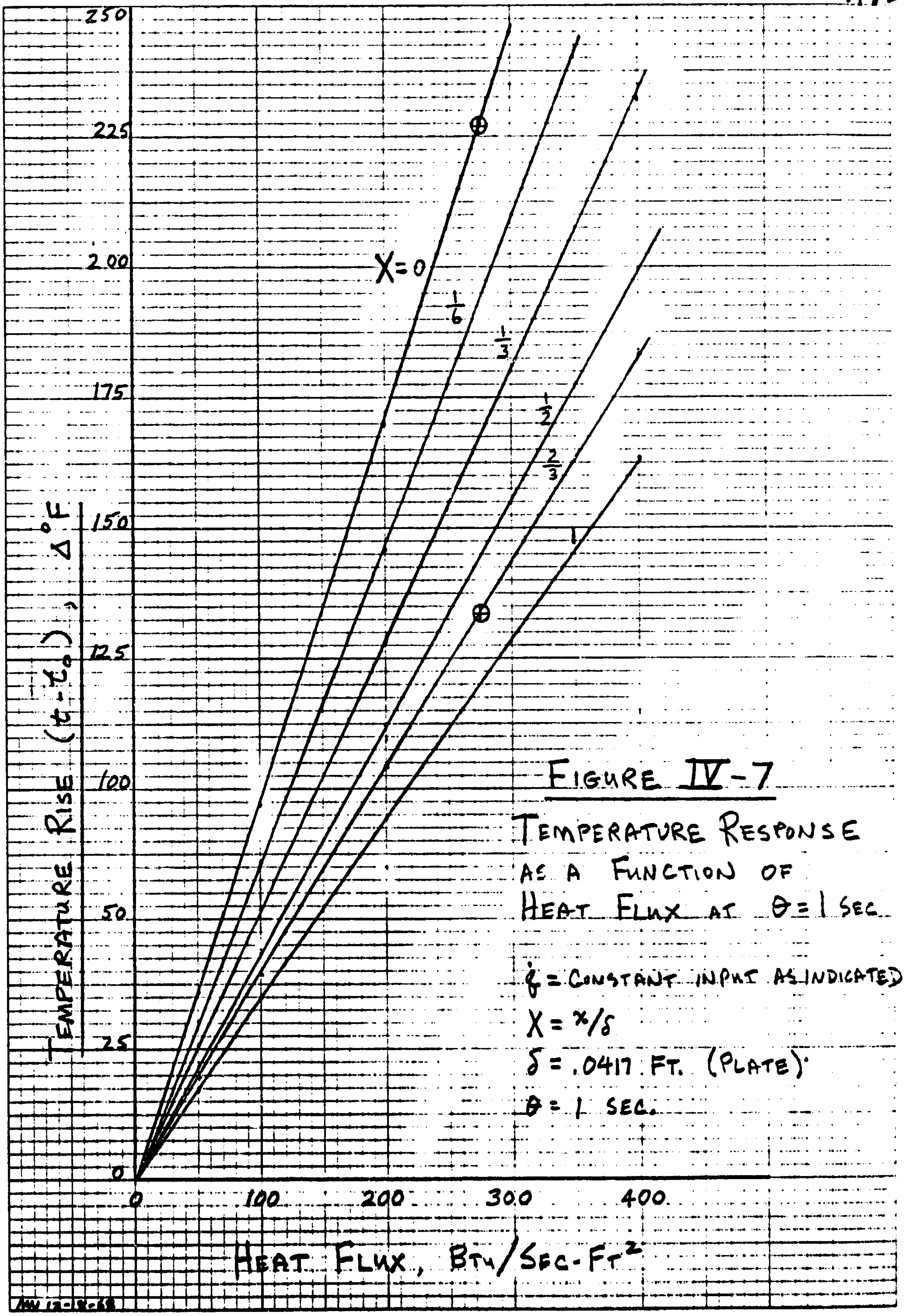


FIGURE IV-6 WALL TEMPERATURES FOR
WATER VELOCITY = 14 & 20 FT/SEC
800
700
600
500
400
300
200
100
0
WALL TEMP (°F)
GAS SIDE
V=14
V=20
V=14
V=20 FT/SEC
WATER SIDE
REVISED DESIGN COND. 2
0
100
200
300
400
HEAT FLUX, BTU/SEC-FT²



250
225
200
175
150
125
100
50
25
0
0
100
200
300
400
Temperature Rise (t - t₀), Δ°F
Heat Flux, Btu/Sec-Ft²
X = 0
1/6
1/3
1/2
2/3
1
Figure IV-7
Temperature Response as a Function of Heat Flux at θ = 1 Sec
q̇ = Constant input as indicated
X = x/δ
δ = .0417 Ft. (Plate)
θ = 1 Sec.

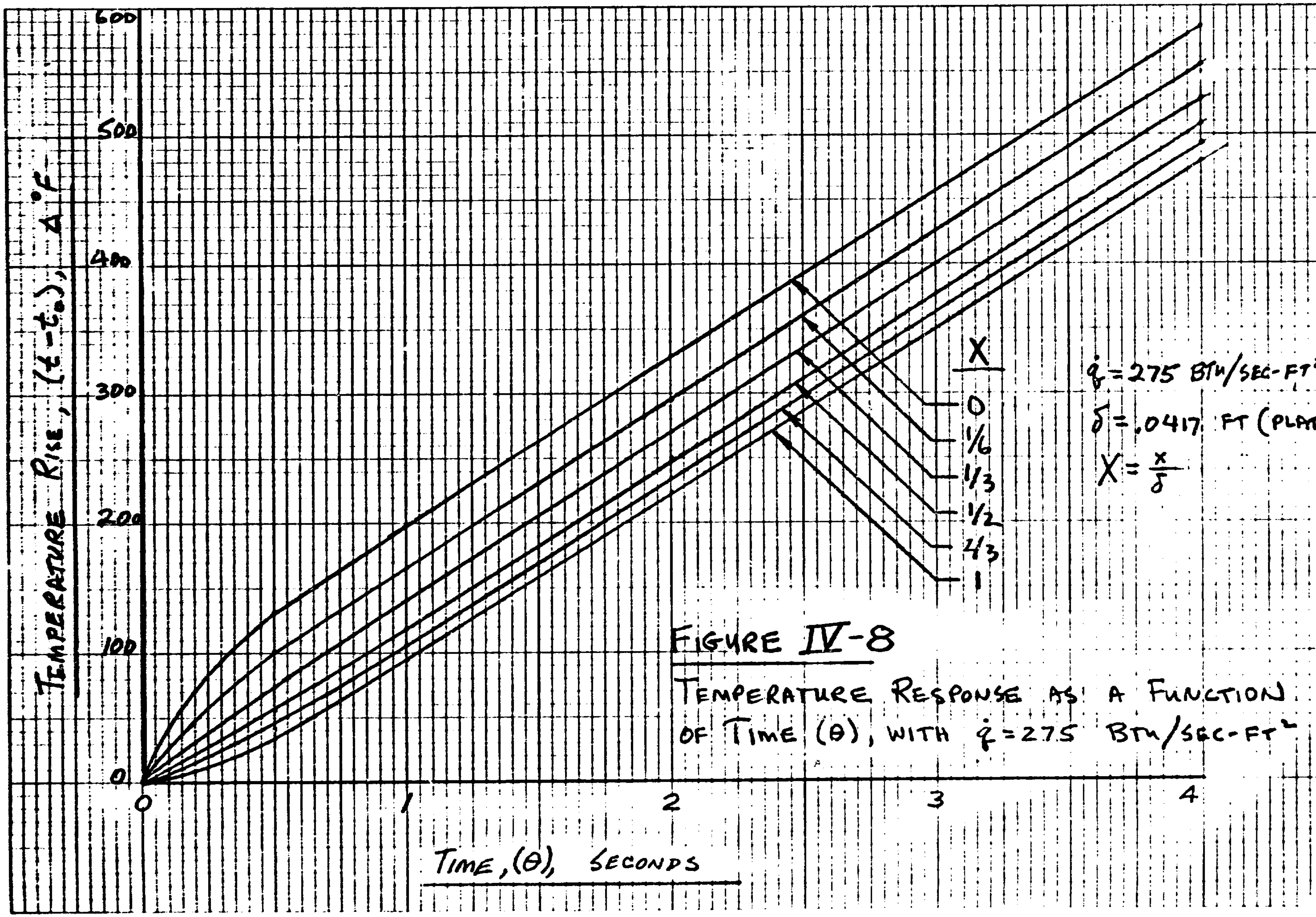


FIGURE IV-8

TEMPERATURE RESPONSE AS A FUNCTION OF TIME (θ), WITH $\dot{q} = 275$ BTU/SEC-FT^2

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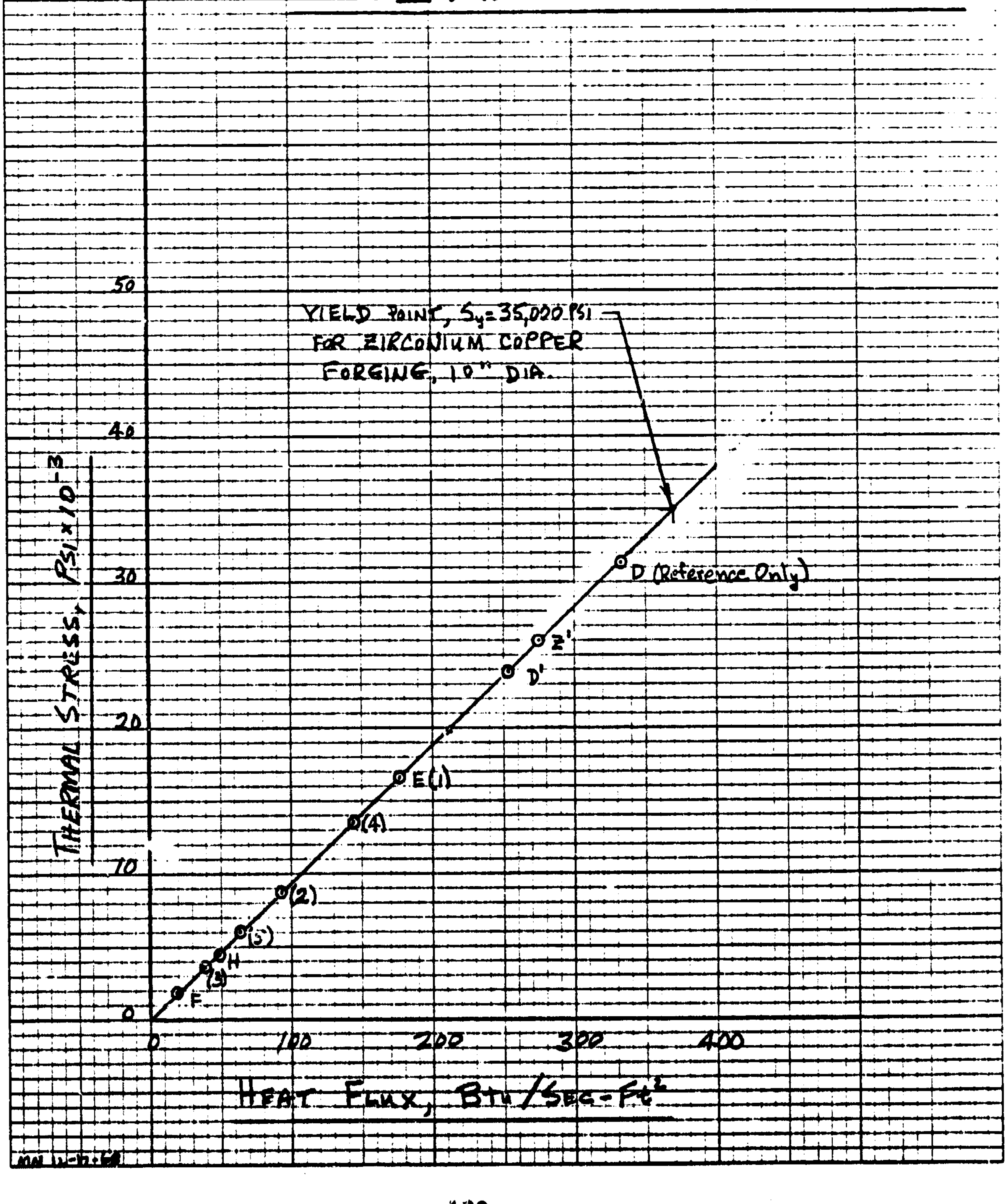
FIGURE IV-9 THERMAL STRESS ACROSS LINER



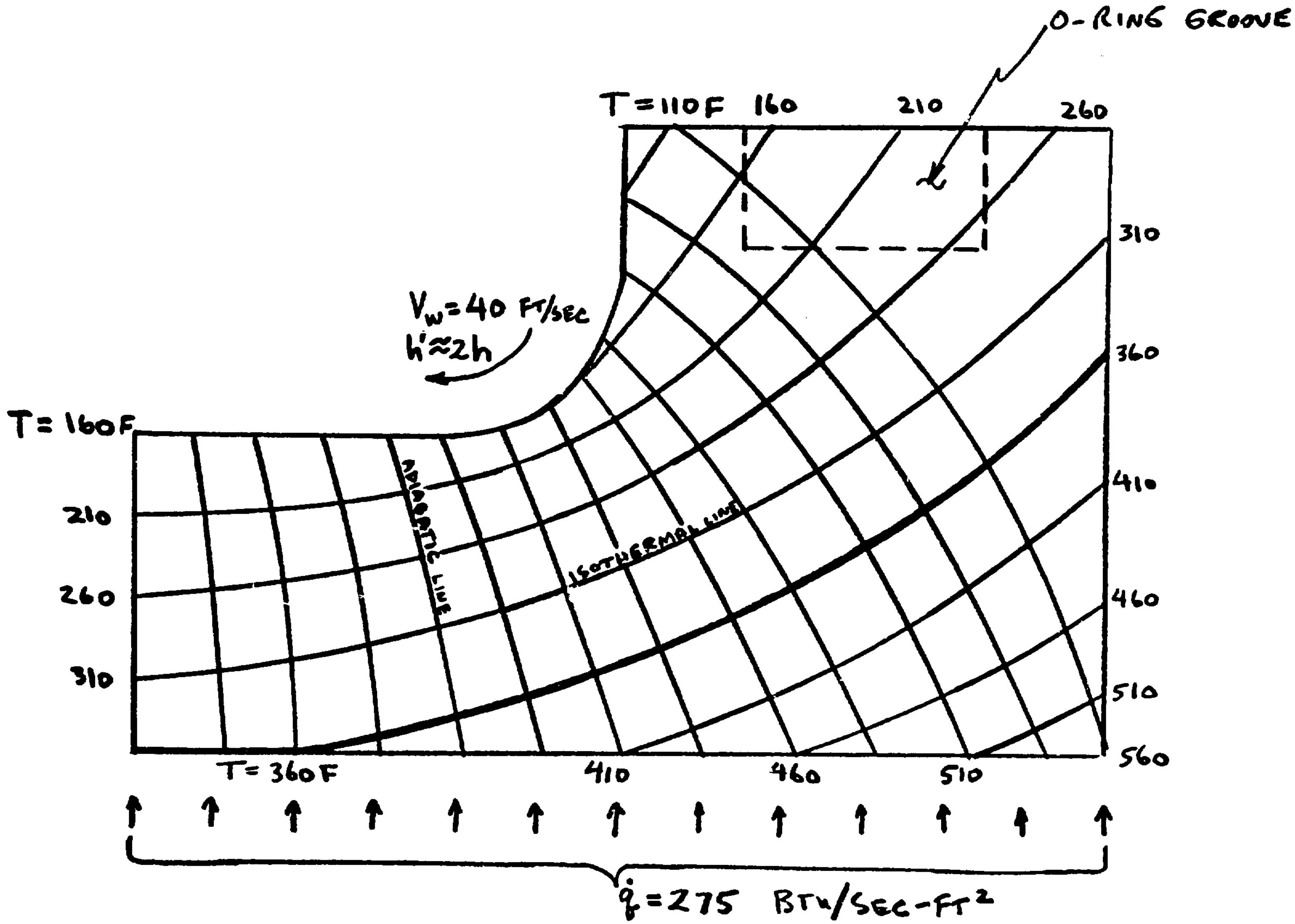


FIGURE IV-10. APPROXIMATE TEMPERATURE PROFILE OF LINER FLANGE

2-15-68