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COMPARISON OF PRIMARY-ZONE COMBUSTOR LINER WALL TEMPERATURES WITH CALCULATED PREDICTIONS

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16. Abstract

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COMPARISON OF PRIMARY-ZONE COMBUSTOR LINER WALL

TEMPERATURES WITH CALCULATED PREDICTIONS

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SUMMARY

Calculated liner temperatures based on a steady-state radiative and convective heat balance at the liner wall were compared with experimental values. Calculated liner temperatures were approximately 8 percent higher than experimental values. A radiometer was used to experimentally determine the required values of flame temperature and flame emissivity. It was shown that calculated flame temperatures do not adequately predict the actual flame temperature. Use of empirical expressions to determine gray-body flame emissivity indicate that emissivity can be evaluated with reasonable accuracy as compared to experimentally determined flame emissivity data if the expression accounts for the luminosity of the flame. Film cooling effectiveness was calculated from an empirical turbulent mixing expression. A turbulent mixing level of 2 percent was required to correlate the data. The combustor used in the investigation had a rectangular cross section of 15 by 30 centimeters (6 by 12 in.) which simulated a section of an annular combustor designed for a gas turbine engine with a high compressor pressure ratio. Tests were conducted over a range of pressure from 10 to 26.7 atmospheres, at inlet-air temperatures of 589 to 922 K (1060° to 1660° R), and at fuel-air ratios up to 0.019.

INTRODUCTION

Accurate prediction of liner wall temperatures, to be used in the design of combustors for advanced gas turbine engines, are needed to determine the air flow required for adequate liner cooling. Analytical methods currently used to calculate liner wall temperatures are relatively unreliable especially at severe operating conditions which make it difficult to evaluate radiative heat flux from the flame. The purpose of this study was to determine how well liner wall temperatures in the primary zone could be predicted from measured radiative heat flux data.

Methods of calculating liner wall temperature for a specified combustor geometry and a range of operating conditions have been developed in references 1, 2, and 3. The calculations consist of evaluating a one-dimensional heat balance in which radiative and convective heat flux entering and leaving the liner surface are equated. Errors in the determination of flame temperature, flame emissivity, and film cooling effectiveness are shown to be the most serious sources of error in making accurate predictions of liner wall temperature as discussed in reference 3.

Flame temperature is generally estimated by assuming thermodynamic equilibrium and using a value of fuel-air ratio that is determined from calculated air-flow and fuel-flow distribution within the combustor. This calculation generally assumes complete vaporization of the fuel, instantaneous mixing of the fuel and air, negligible recirculation, and complete and instantaneous combustion. All of these assumptions oversimplify actual combustor conditions especially within the primary zone.

The flame emissivity for a nonluminous flame may be estimated by the methods described in reference 2. Methods for predicting flame emissivity in the case of a luminous flame such as that encountered at the elevated combustor pressures required for advanced high compressor pressure-ratio engines are less reliable. Here, the high carbon formation within the primary zone at elevated pressures has a significant influence on flame emissivity as shown in reference 4.

Most film cooling effectiveness calculations are based on experiments conducted in tunnels with low turbulence levels. However, relatively intense turbulent mixing occurs within a gas turbine combustor. Thus, reference 5 which presents a method of relating film cooling effectiveness to level of turbulence is useful in actual combustor applications.

For this study, tests were performed with a short length rectangular combustor segment designated as model 3 that was described in reference 6. The heat transferred to the combustor liner from the flame was determined from experimentally determined flame temperature and emissivity values obtained from radiometric measurements of the flame as discussed in reference 4. Flame properties were obtained from optical observations through sapphire windows located 0.05 meter (2 in.) downstream of the combustor faceplate. Theoretically predicted liner wall temperatures were compared with experimental values. Experimental test conditions encompassed an inlet pressure range from 10 to 26.7 atmospheres; inlet temperature levels from 589 to 922 K (1060° to 1660° R); and values of fuel-air ratio up to 0.019. The experimental liner temperatures were obtained from the average of three thermocouple located in-line with the radiometer.
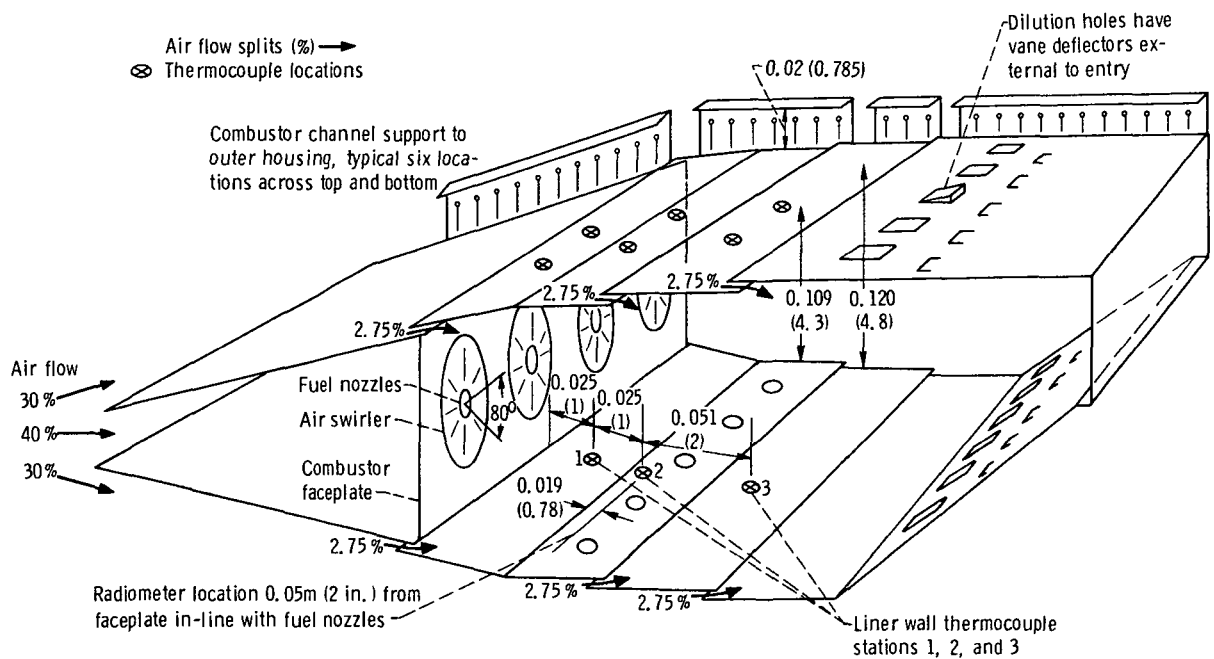
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EXPERIMENTAL METHODS

Experimental Combustor

The combustor used in this investigation was similar to combustor model 3 of reference 6. A schematic of the test combustor is shown in figure 1. The assembled combustor was unique in that each axially segmented panel of the combustor liner was independently supported from the housing wall. The outer housing has a series of channel supports spaced 0.05 meter (2 in.) apart across the width of the combustor from which the combustor liner panels are supported. The structural support member connecting the outside surface of the combustor liner to the housing was immersed in the cool dilution air passage. Saw cuts in this support, perpendicular to the combustor housing, are used to stress relieve the member. Clearance between each combustor portion assures that no buckling of the assembled wall can occur.

The combustor has an inlet snout open area which is 40 percent of the combustor inlet area. The main portion of the airflow captured by the snout passed through air swirlers, and a small portion (approximately 6 percent of the total flow) was used to film cool the sides of the combustor. The combustor liner walls were film cooled by means of continuous slots, and the dilution air was admitted by means of external scoops. The mass flow distribution in the combustor was calculated by means of a computer program for the analysis of annular combustors as described in reference 7 and the distribution is shown in figure 1.
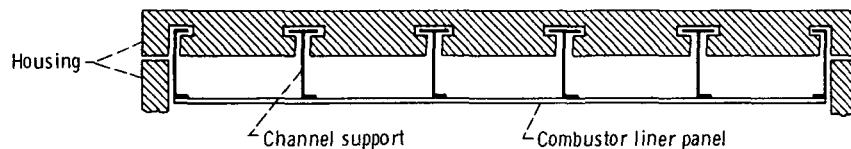
Provision was made in the combustor side plates to allow direct observation of the primary flame zone at various distances downstream of the combustor faceplate. For these runs the observation port was located 0.05 meter (2 in.) downstream. Details of the test facility and combustor instrumentation are presented in appendix A.

Liner Temperature Measurement

The liner skin temperatures were measured by nine Chromel-Alumel thermocouples installed in the combustor wall. The thermocouple sheaths were located in grooves milled into the combustor wall and the thermocouple junction was filled was high temperature braze. A thermocouple including sheath, insulation, and a pair of wires was 0.0015 meter (0.06 in.) in diameter. The hot immersed L/D of the sheath was 16 before the wire was brought out through the cold air passage. The resulting conduction loss from the thermocouple was calculated from the nomograph presented in reference 8. There was also a small correction applied to account for the thermal gradient in the combustor liner wall since the thermocouple junction was on the cold side of the wall to

Air flow splits (%) ⟶
⊗ Thermocouple locations

Combustor channel support to
outer housing, typical six loca-
tions across top and bottom

0.02 (0.785)

Dilution holes have
vane deflectors ex-
ternal to entry

2.75%   2.75%

0.109   0.120
(4.3)   (4.8)

Air flow
30%

40%

30%

2.75%

Fuel nozzles

Air swirler

Combustor
faceplate

80°

0.025
(1)

0.025
(1)

0.051
(2)

1⊗   ⊗2   ⊗3

0.019
(0.78)

Radiometer location 0.05m (2 in.) from
faceplate in-line with fuel nozzles

2.75%   2.75%

Liner wall thermocouple
stations 1, 2, and 3

(a) Cutaway view of combustor. (Dimensions in m (in.) unless indicated otherwise.)

Housing

Channel support   Combustor liner panel

(b) Typical end view of housing to illustrate combustor liner assembly.

Figure 1. - Schematic of combustor liner assembly. Combustor width, 0.30 meter (12 in.), length, 0.32 meter (12.5 in.); maximum combustor housing height, 0.15 meter (6 in.).

eliminate radiation corrections. The corrections applied were of the order of 6 to 17 K (11° to 31° R) in this investigation.

Three thermocouples were located at three stations; 0.025 meter (1 in.), 0.05 meter (2 in.), and 0.10 meter (4 in.) downstream of the combustor faceplate. Two thermocouples were located on the upper liner wall and one on the lower liner wall at each station. One of the two thermocouples located on the upper wall at a station was placed in-line with a fuel nozzle and the other between fuel nozzles. The thermocouple on the lower wall was in-line with a fuel nozzle. The three thermocouples located 0.051 meter (2 in.) downstream of the combustor faceplate at station 2 shown in figure 1 gave consistently higher temperature readings than the other six thermocouples which were closer to film cooling slots. The experimental liner temperatures presented are the average of these three highest thermocouple readings in-line with the radiometer view ports.

A correction was also applied to the thermocouple indications to account for the liner channel support. This correction can be assumed to be similar to that for a cooling fin. The temperature of the wall with a fin was determined from the following expression:

$$T'_w - T_w = \frac{T_c - T_w}{\cosh(ml)} \tag{1}$$

where $m = 2h/kb$. Symbols are defined in appendix B. The fin thickness was 0.002 meter (0.08 in.); the fin length was 0.014 meter (0.54 in.); the material was Hastelloy with a conductivity of 0.2175 J/(sec)(cm)(°C) (0.003494 Btu/(ft)(sec)(°F)); and the film coefficient was determined from the flow conditions within the passage bounded by the fin, the combustor housing, and the liner wall. The conduction path along the liner to the fin was 0.001 meter (0.05 in.) for the two thermocouples in-line with the fuel nozzles and 0.030 meter (1.25 in.) for the thermocouples between the fuel nozzles. At an operating pressure of 10 atmospheres a correction factor of about 22 K (40° R) was indicated, and at 20 atmospheres the correction factor was 6 K (11° R).

Flame Temperature and Emissivity Measurement

Measurements of optical properties of the flame were obtained with a radiometer by observation through sapphire windows located 0.05 meter (2 in.) downstream of the combustor faceplate in-line with the three liner wall thermocouples at station 2. Two radiometric determinations were taken: in the first, the effective flame temperature was calculated from a narrow band region where the flame could be assumed to be a blackbody; and in the second, the total radiance was calculated from the observed flame

radiance. An indium-antimonide detector was used for the narrow band spectrum (2.608- to 2.789-μm half-width) and an unimmersed bolometer thermal detector was used for the wide spectral band (0.25- to 6-μm half-width, cut-off due to sapphire windows). Details of the radiometer and calculation procedure are presented in reference 4. The equivalent gray-body emissivity was calculated from the Stephan-Boltzmann equation by using the values of flame temperature and total flame radiance.

Combustor Test Conditions

The combustor was operated by burning ASTM A-1 jet fuel over a range of fuel-air ratios. Liner temperatures are compared at a fuel-air ratio of 0.019. The combustor was operated at the test conditions given in table I.

TABLE I. - COMBUSTOR TEST CONDITIONS

Test con-dition	Inlet pres-sure, atm	Inlet temperature		Reference velocity		Mass flow	
		K	$^{\circ}$R	m/sec	ft/sec	kg/sec	lb/sec
A	10	589	1060	21.3	70	6.05	13.36
B	20	589	1060	21.3	70	12.10	26.72
C	26.7	589	1060	21.3	70	16.16	35.67
D[a]	10	756	1360	27.4	90	6.05	13.36
E[a]	10	922	1660	33.5	110	6.05	13.36

[a]Preheating of inlet air above 589 K (1060° R) required use of an in-line combustor thus vitiating inlet air stream.

ANALYTICAL CALCULATION OF LINER WALL TEMPERATURE

The method predicting liner wall temperature was based on a one-dimensional steady-state heat balance in which the radiative and convective heat fluxes entering and leaving the wall are equated at the liner surface (refs. 1 and 2). Thus, the following equation was used:

$$R_1 + C_1 = R_2 + C_2 \qquad (2)$$

The liner wall temperature was determined from equation (2) by using an iterative process to solve for wall temperature. The assumptions and methods that were used to evaluate each term in equation (2) are described in the following sections.

Internal Radiation R_1

The radiant heat flux was calculated from

$$R_1 = \sigma\left(\frac{1 + e_w}{2}\right)\left(e_f T_f^4 - \alpha_f T_w^4\right) \tag{3}$$

In this program, both the flame temperature T_f and the emissivity e_f were determined from radiometric flame measurements at each specified test condition. The term α_f was calculated from the relation $\alpha_f/e_f = (T_f/T_w)^{1.5}$. At high levels of flame emissivity the expression gives values of α_f greater than unity. The present program used this approximation for α_f, however, a limit of unity was imposed. The remaining term e_w, assumed constant for all wall temperatures, was assigned a value of 0.685.

Internal Convection C_1

The internal convective heat transfer is calculated from

$$C_1 = h_1(T_{ad, w} - T_w) \tag{4}$$

The equation used to determine the film coefficient h_1 was evaluated somewhat differently in this program than in the usual case for an annular or cylindrical configuration since the combustor liner was composed of a series of flat plates. Assuming the boundary layer to be turbulent over the whole length, a relation of the following form was used to evaluate the Nusselt number (ref. 9).

$$\overline{Nu_x} = 0.037(Re_x)^{0.8}(Pr)^{0.33} \tag{5}$$

The properties of the gas at the exit of the slot were calculated at the adiabatic wall temperature. The adiabatic wall temperature was related to the film cooling effectiveness by

$$h = \frac{T_f - T_{ad, w}}{T_f - T_c} \tag{6}$$

In this program the film cooling effectiveness was determined from reference 5 as

$$h = \frac{1}{1 + C_m \frac{x}{Ms}} \tag{7}$$

where C_m is the turbulent mixing coefficient. The equivalent adiabatic temperature was obtained from equation (6) using the experimentally determined flame temperature and coolant air temperature.

External Radiation R_2

The external radiation heat flux was calculated from the following expression

$$R_2 = \sigma e' \left(T_w^4 - T_c^4\right) \tag{8}$$

where

$$e' = \frac{e_w e_c}{e_c + e_w(1 - e_c) \frac{A_w}{A_c}} \tag{9}$$

The casing temperature was assumed equal to the inlet air temperature. A term including the view factor was omitted since the view factor was taken to be one. The surface area of the liner wall was equal to the surface area of the casing so that $A_w/A_c = 1$. The emissivity of the liner wall was assumed constant and equal to 0.685. The emissivity of the casing was assumed constant and equal to 0.80. A factor $(1 - \alpha)$ was added for the prediction of liner wall temperatures at test conditions D and E to account for the effects of vitiation of the inlet air above 589 K $(1060^O$ R).

External Convection C_2

The external heat transfer is calculated from

$$C_2 = h_2(T_w - T_c) \tag{10}$$

The Nusselt number used to evaluate the heat transfer coefficient h_2 was determined from equation (4) using the bulk properties of the cool gas stream. The effect of the inlet diffuser passage was neglected, and the Reynolds number was evaluated from the

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leading edge where the combustor walls become parallel. The heat transfer coefficient C_2 was also increased by a factor corresponding to 14 percent to account for the effect of the liner channel supports (see Experimental Methods) assuming an infinite fin such as discussed in reference 9.

RESULTS AND DISCUSSION

A rectangular combustor segment with a cross section of 15 by 30 centimeters (6 by 12 in.) was used to simulate a section of an annular combustor. The combustor was operated at pressure levels to 26.7 atmospheres and inlet air temperatures to 922 K

TABLE II. - VARIABLES REQUIRED TO EVALUATE THE CONVECTIVE HEAT TRANSFER TERMS

C_1 term	
Primary mass flow, $\dot{m}_t$	0.34
Coolant mass flow, $\dot{m}_t$	0.0275
Primary zone height, cm (in.)	10.9 (4.3)
Distance downstream from coolant slot, cm (in.)	5.1 (2)
C_2 term	
Stream mass flow, $\dot{m}_t$	0.2725
Stream passage height, cm (in.)	2.0 (0.785)
Distance downstream from end of diffuser, cm (in.)	1.9 (0.75)

TABLE III. - EXPERIMENTAL DATA

Test condition	Flame temperature		Flame emissivity	Liner wall temperature							
				Top wall		Bottom wall	Uncorrected average		Corrected average		
	K	°R		Between nozzles, K	In-line with nozzles, K	In-line with nozzles, K	K	°R	K	°R	
A	2180	3924	0.32	690	657	761	702	1264	723	1301	
B	2190	3942	.60	706	736	758	733	1319	755	1359	
C	2256	4060	.855	770	733	792	765	1377	790	1422	
D	{2417	4350	.30	861	883	889	878	1580	885	1593	
	{2397	4311	.30	897	889	963	916	1649	923	1661	
E	2556	4600	.27	1061	1033	1122	1072	1930	1095	1971	

(1660° R) over a range of test conditions as previously shown in table I. The values of the variables required in the heat balance as represented by equation (2) are listed in table II. Experimental flame temperatures, flame emissivity, and liner wall temperatures taken at a position 0.05 meter (2 in.) downstream of the fuel nozzles are presented in table III.

Evaluation of Major Variables

Determination of flame temperature. - Since flame temperature is one of the most difficult variables to evaluate in the calculation of liner temperature it is usually calculated from ideal mixing and thermodynamic theory. Based on this approach a plot of
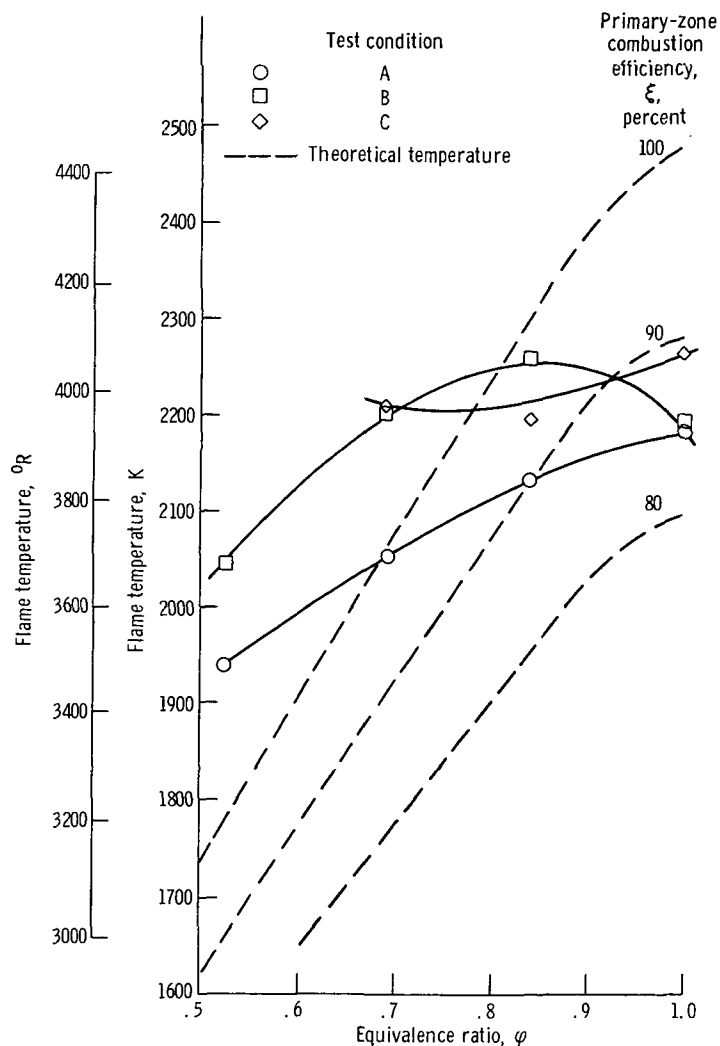
Figure 2. - Comparison of theoretical and experimental flame temperature for inlet temperature of 589 K (1060° R). $T_f = T_2 + \xi \Delta T$.

theoretical flame temperature for a range of primary zone equivalence ratios is presented in figure 2. Three levels of combustion efficiency which correspond to an 80, 90, and 100 percent theoretical temperature rise are shown for comparitive purposes at a combustor pressure of 10 atmospheres.

Experimental flame temperature data from model 3 at test conditions A, B, and C at a combustor inlet temperature of 589 K (1060° R) are also shown in figure 2. The primary-zone equivalence ratio for these data were calculated from the air flow distribution calculated in accord with the procedure outlined in reference 7 and assuming that the fuel mixed with the available air instantaneously. In general at the leaner equivalence ratio of 0.7 which corresponds to an overall fuel air ratio of less than 0.013, the theoretical flame temperature is less than experimental values. At the higher equivalence ratios the theoretical temperature is higher than experimentally observed.

Experimental measurement of temperature within a combustor is difficult due to the lack of knowledge of the degree of mixing of fuel and air, the degree of vaporization, the extent of recirculation of hot gases, and the reliability of instrumentation techniques. As a result, very little data are available for combustor design applications. Previous experience with eleven additional combustor configurations from reference 4 indicate that there is no clear cut trend of flame temperature with equivalence ratio. Many combustors indicated an increasing flame temperature with increasing equivalence ratio and other combustors indicated an opposite trend. These variations can be attributed not only to primary-zone variables as noted but also to a possible shift in the location of the flame. In this study and that of reference 4 radiometric observation was made at a fixed location 0.05 meter (2 in.) downstream of the combustor faceplate.

In reference 10, a two-color pyrometric method was used to experimentally determine the flame temperature in a 0.36-meter (14-in.) diameter can-type combustor at distances of 0.14, 0.19, and 0.30 meter (5.6, 7.5, and 12 in.) downstream of the fuel nozzle tip. The observed flame temperature was generally constant over this range of observation. Operation over a wide range of fuel flows indicated also that the observed flame temperature remained relatively constant at a value of about 1920±31 K (3456±55° R).

Comparison of the theoretically determined flame temperature with experimental values from figure 2 and reference 10 indicate that (1) a wide variation in effective flame temperature is not observed with variation in primary-zone equivalence ratio; thus, indicating that at least in a liquid fuel pressure-atomizing combustor burning takes place over a comparatively narrow range of effective equivalence ratios; and (2) the present method of calculating flame temperatures does not adequately predict the actual effective flame temperature for this type of fuel injection technique.

Determination of flame emissivity. - Flame emissivity as shown in figure 3 increased with increasing compressor pressure ratio. Two empirical equations are plotted for comparative purposes (refs. 1 and 2). The empirical expression of refer-
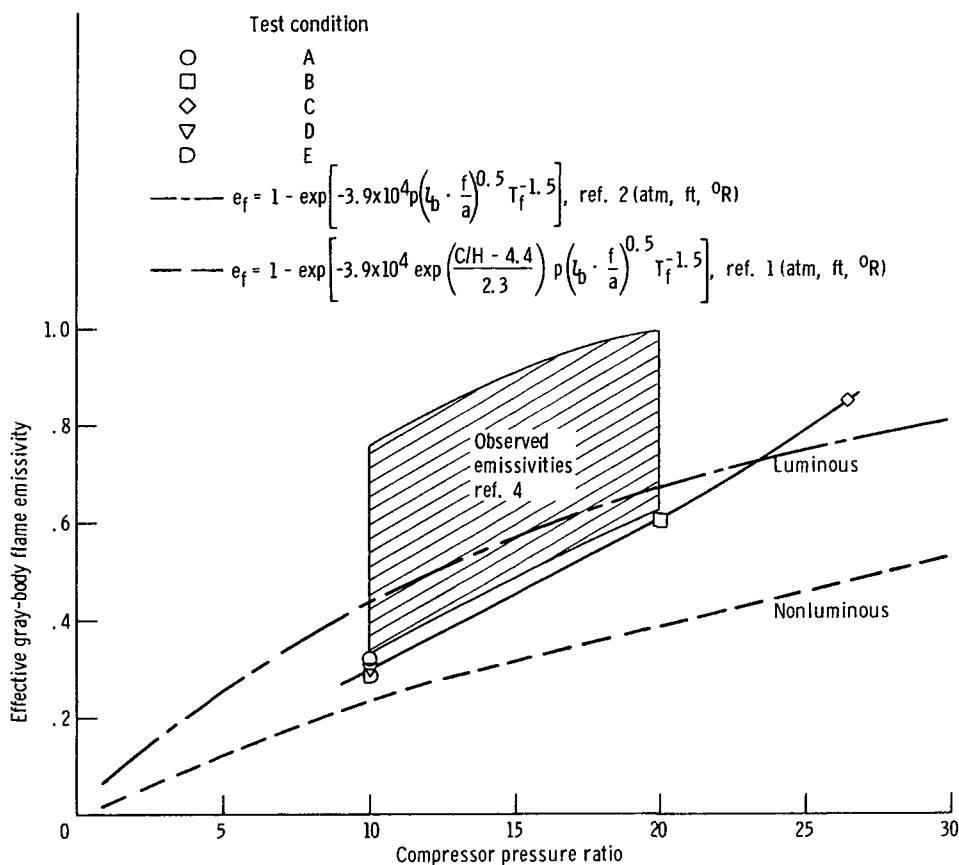
Test condition

○	A
□	B
◇	C
▽	D
◻	E

$$e_f = 1 - \exp\left[-3.9 \times 10^4 p\left(l_b \cdot \frac{f}{a}\right)^{0.5} T_f^{-1.5}\right], \text{ ref. 2 (atm, ft, }^\circ R)$$

$$e_f = 1 - \exp\left[-3.9 \times 10^4 \exp\left(\frac{C/H - 4.4}{2.3}\right) p\left(l_b \cdot \frac{f}{a}\right)^{0.5} T_f^{-1.5}\right], \text{ ref. 1 (atm, ft, }^\circ R)$$

Figure 3. – Comparison of empirical and experimental flame emissivity for an equivalence ratio of 1.0.

ence 2 is considered satisfactory for a distillate fuel producing a nonluminous flame at pressures up to 5 atmospheres. The equation in reference 1 is a modification of the equation in reference 2 which corrects for smoke produced within the flame by adding a factor which accounts for the effect of the hydrogen-carbon ratio of the fuel and variation in combustor operating pressure levels. The flame temperature required in the equations of references 1 and 2 was calculated from the theoretical temperature rise and the inlet-air temperature. The theoretical temperature rise was calculated by assuming an equivalence ratio of 1 (i. e., stoichiometric fuel-air ratio). The inlet-air temperature was calculated from the compressor pressure ratio assuming a cycle efficiency of 80 percent.

Also shown in figure 3 is a general broad band of emissivities for various inlet pressures which were obtained during the investigation reported in reference 4. These combustor configurations are representative of very clean to very smoky operation. The emissivity data from the present investigation are shown by the data points in figure 3 and represent a condition corresponding to an overall fuel-air ratio of approximately 0.019 (the primary air and fuel distribution is such that this corresponds approximately to a primary-zone equivalence ratio of one). It should be noted that test condition A

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corresponds roughly to an inlet temperature compatible with a compressor pressure ratio of 10.

Comparisons of the various curves indicate that, as expected, the nonluminous correlation predicts values too low compared to experimental data. The expression from reference 1 which accounts for the luminosity of the flame predicts a value which agrees to within 37 percent of the experimental values of emissivity at a pressure ratio of 10 (i. e., 0.42 predicted compared to 0.32 measured). The observed flame emissivities from reference 4 indicate that in a smoky combustor configuration it is not unreasonable to assume that the flame behaves as a blackbody radiator as the pressure ratio is increased to 20 atmospheres.

Determination of film cooling effectiveness. - It has been shown in reference 5 that the film cooling effectiveness can be related to a turbulent mixing coefficient as indicated by equation (7). The film cooling effectiveness, as obtained from reference 5 and shown in figure 4 is presented in terms of a downstream distance parameter for various levels of turbulent mixing levels from 0.01 to 0.20. The value of x/Ms for the present configuration corresponds to a value of 5.75. As shown in figure 4 it would be reasonable to expect the experimental film cooling effectiveness to vary from 0.46 to 0.95 for a range of the turbulent mixing parameter from 0.20 to 0.01. A variation of this magnitude could result in a wide variation in predicted liner temperature level.

Measured values of the turbulent mixing coefficient are very difficult to obtain in combustor tests because of the uncertainties involved in measurement technique. For example, hot wire anemometry measurements are generally limited to low temperature conditions. In this program the value of the turbulent mixing coefficient was determined primarily by means of a curve fitting technique. Various values of C_m were selected and the liner wall temperature calculated. A reasonable value of C_m which correlated the liner temperature data was used.

Effect of Empirically Determined Variables on Calculated Liner Wall Temperatures

Over the range of experimental conditions a series of calculations were made using equation (2) to determine effects of flame temperature, flame emissivity, and the film cooling turbulent mixing coefficient on calculating liner wall temperature.

Effect of flame temperature. - In figure 5, the liner wall temperature is shown for a range of flame temperatures corresponding approximately to equivalence ratios up to 1.0. In solving equation (2), one value of flame emissivity (0.5), one value of inlet-air temperature (589 K (1060° R)), and two pressure levels (10 and 20 atm) were chosen. The turbulent mixing coefficient was assumed constant at a value of 0.02. As the flame temperature increased, the liner temperature is shown to rise at an increasing rate. As the pressure increased, lower wall temperatures were predicted. Lower liner wall
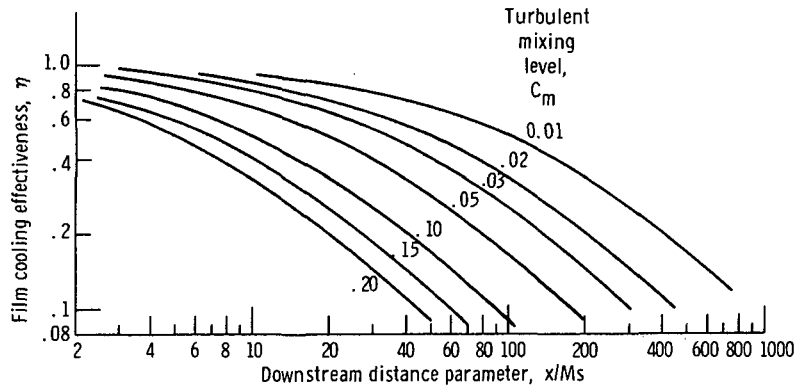
Figure 4. - Turbulent mixing correlation: effect of mixing coefficient C_m on film cooling effectiveness η (ref. 5).
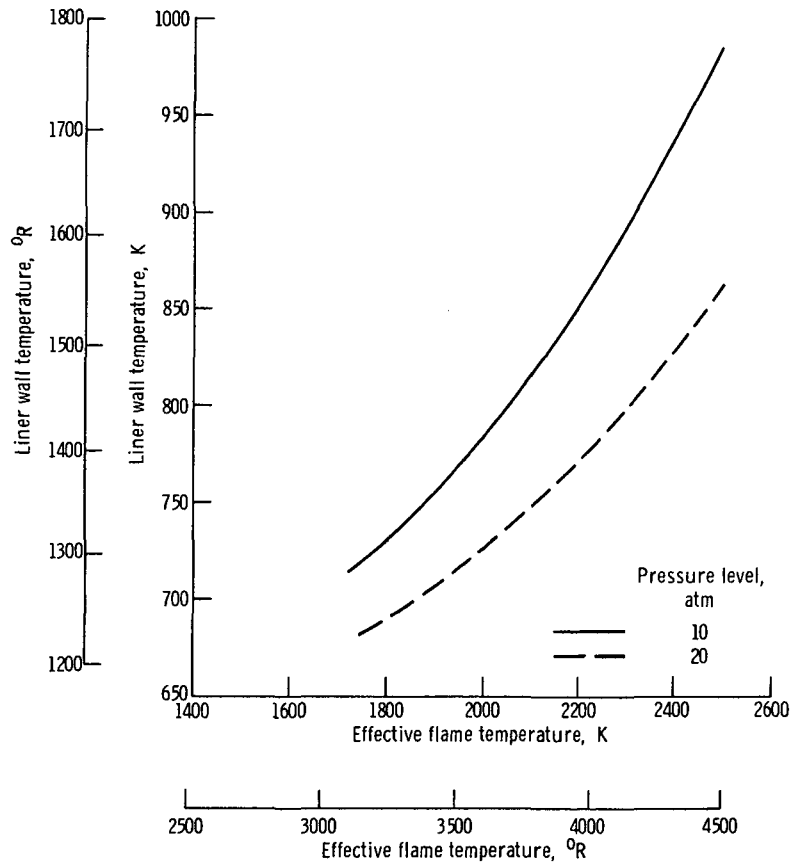

Figure 5. - Effect of flame temperature on calculated liner wall temperature. Flame emissivity, 0.5; turbulent mixing coefficient, 0.02; inlet-air temperature, 589 K (1060° R).

temperatures at the higher pressure level were attributed to improvements in the convective heat transfer coefficient.

Effect of flame emissivity. - In figure 6, the liner wall temperature is shown for a range of emissivities. One value of flame temperature (2472 K (4450° R)), one value of inlet air temperature 589 K (1060° R), and two pressure levels (10 and 20 atm) are included in figure 6. The turbulent mixing coefficient was assumed constant at a value of 0.02. Increasing the emissivity level of the flame produces an increase in the calcu-
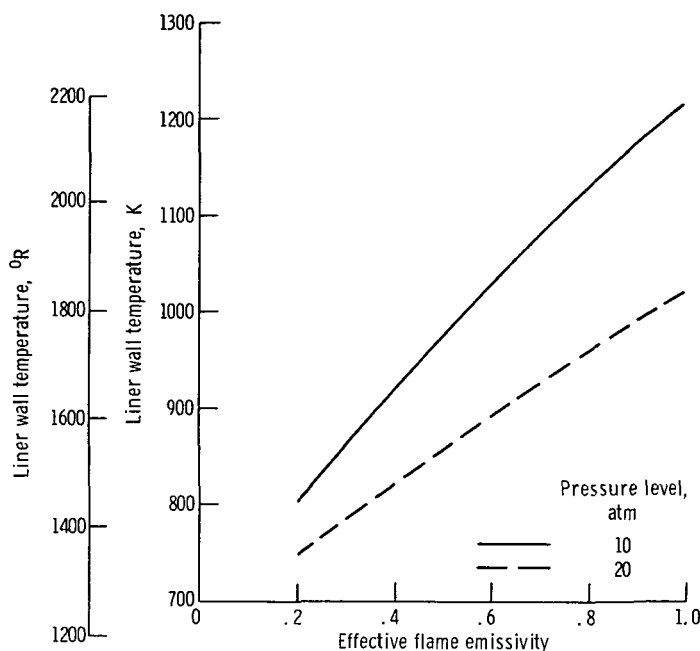
Figure 6. - Effect of flame emissivity on calculated liner wall temperature. Flame temperature, 2472 K (4450° R); turbulent mixing coefficient, 0.02; inlet-air temperature, 589 K (1060° R).

lated liner temperature. As the pressure increased, lower liner wall temperatures resulted due to the improvement in the convective heat transfer coefficient.

Effect of turbulent mixing coefficient. - In figure 7, the calculated liner wall temperature is shown for a range of turbulent mixing parameters. One value of flame temperature (2472 K (4450° R)), one value of flame emissivity (0.5), one value of inlet air temperature (589 K (1060° R)), and two pressure levels (10 and 20 atm) are included in figure 7. The turbulent mixing coefficient was varied from 0.02 to 0.16. As reported in reference 5, ranges experienced for combustors have been as low as 0.03 and as high as 0.15. The liner wall temperature level is shown to increase appreciably as the mixing coefficient is increased.

Error analysis. - Parametric variations of effective flame temperature, flame emissivity, and turbulent mixing coefficients are presented in figures 5, 6, and 7.
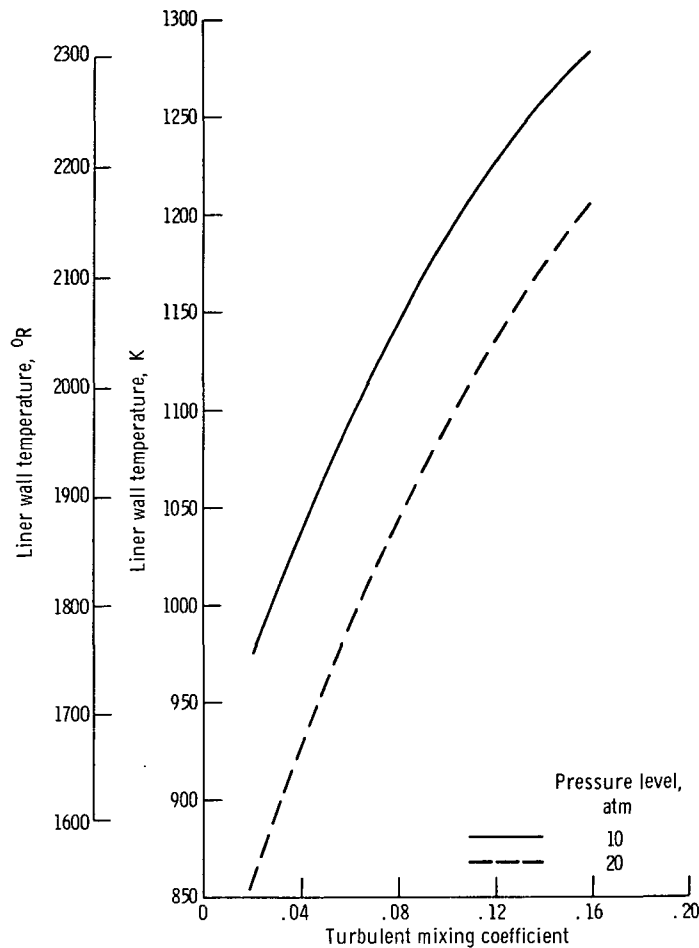
Figure 7. - Effect of turbulent mixing coefficient on calculated liner wall
temperature. Flame temperature, 2472 K (4450° R); flame emissivity,
0. 5; inlet-air temperature, 589 K (1060° R).

Based on an effective flame temperature of 2472 K (4450° R), an emissivity of 0. 5, and
a turbulent mixing coefficient of 0. 15 a comparison was made of the effect of individually
decreasing each of these variables by 10 percent for an inlet temperature level of 589 K
(1060° R) and pressure of 10 atmospheres. Thus, a reduction of 10 percent in flame
temperature dropped the predicted liner temperature from 972 to 861 K or about
11. 6 percent; the emissivity reduced the predicted liner temperature from 975 to 948 K
or about 2. 8 percent; and the turbulent mixing coefficient reduced the predicted liner
temperature from 1268 to 1248 K or about 1. 6 percent. This could give a possible ac-
cumulated error of 16. 0 percent in calculated liner temperature. It should be noted that
it is often not possible to estimate flame temperature, flame emissivity, and turbulent
mixing coefficient to within 10 percent during actual combustor operation. The effects
in liner wall temperature due to the variations of flame temperature, flame emissivity,
and turbulent mixing parameters point out the relative importance of selecting appro-

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priate values (in particular flame temperature) for design calculations in the combustor primary zone.

Comparison of Experimental and Calculated Liner Wall Temperatures

Averages of the three liner wall temperature measurements at station 2 are shown in figure 8. The data are presented for an inlet air temperature of 589 K (1060° R), fuel-air ratio of 0.019, reference velocity of 21.3 meters per second (70 ft/sec), and a range of pressures to 26.7 atmospheres. The experimental liner temperatures are compared with values calculated from equation (2) using experimentally determined flame temperature and emissivity and a value of C_m of 0.02.

The convective heat transfer term C_1 in equation (2) for this case - where the liner wall temperature is dominated by radiation from the flame - results in heat being transferred from the wall to the film cooling stream. The heat removed from the wall is based on the assumption that the gases are at some adiabatic gas temperature which can be estimated and that there is no profile. The value of C_m used to determine the adiabatic gas temperature depends on the heat transferred from the surface. Since details of the gradients within the primary zone were not obtained it was assumed that the value of C_m could be represented by a constant at all test conditions. A value of C_m corresponding to 0.02 was selected to correlate the experimental data with equation (2). The low value of C_m required to correlate the data, as compared with the combustors reported in reference 5 (C_m in the range of 0.03 to 0.15), reflects the probability that there is an effect due to the gas temperature profile on C_m within the combustor.
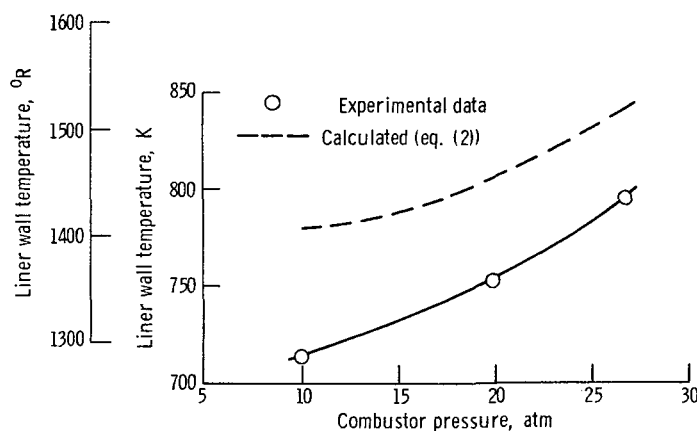

Figure 8. - Comparison of experimental and calculated liner wall temperatures over a range of inlet-air pressures. Inlet-air temperature, 589 K (1060° R); combustor reference velocity, 21.3 m/sec (70 ft/sec); fuel-air ratio, 0.019.

As shown in figure 8 the experimental liner wall temperature increased with in-
creasing pressure. Experimental liner wall temperatures and values calculated from
equation (2) agreed within 8 percent over a pressure range of 10 to 26.7 atmospheres.
As previously shown in figure 6 as pressure increased liner wall temperature decreased
for constant flame conditions; whereas, in actual combustor operation flame conditions
changed and an increase in liner temperature was observed. This is due to the fact that
while liner cooling by convection is improved the heat transfer due to flame radiation is
increased with the net results that the liner temperature is higher.

In figure 9, the effect of inlet-air temperature on liner temperature for a constant
pressure of 10 atmospheres, fuel-air ratio of 0.019, and reference velocities from
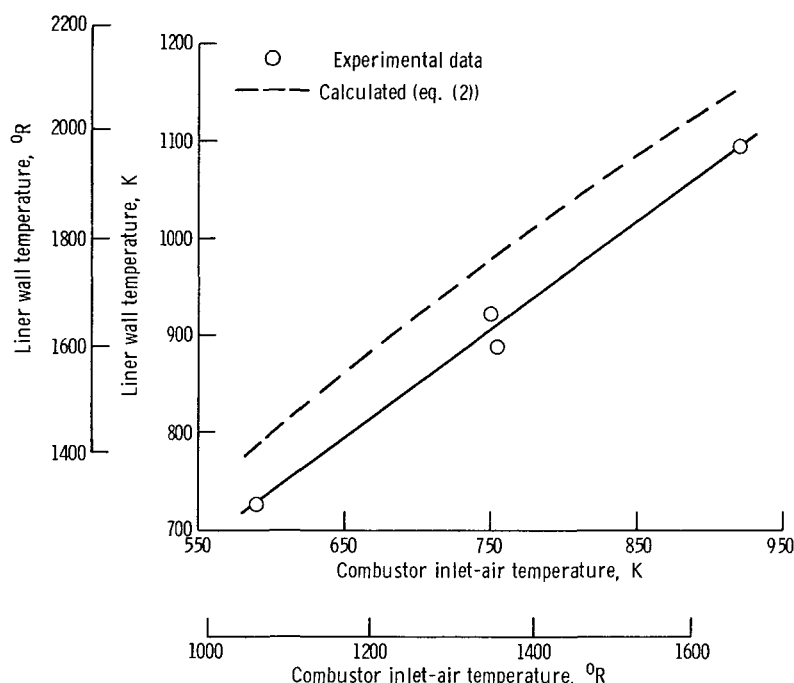
Figure 9. - Comparison of experimental and calculated liner wall temperatures
over a range of inlet-air temperatures. Inlet pressure, 10 atm; total air-flow,
6.06 kg/sec (13.36 lb/sec); fuel-air ratio, 0.019.

21.3 to 33.3 meters per second (70 to 110 ft/sec) is shown. The mass flow through the
combustor was held constant and the reference velocity allowed to vary so that the cool-
ing film Reynolds number remained relatively constant. The liner wall temperature is
shown to be strongly dependent on the combustor inlet-air temperature level. Agree-
ment between experimental and liner wall temperatures calculated from equation (2) was
within 7 percent over the inlet-air temperature range from 589 to 922 K (1060° to
1660° F). The overall trend of increasing liner temperature with increasing inlet air
temperature agreed with experimental results.

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These results indicate that the model used to predict liner wall temperature is quite sensitive to the flame temperature, flame emissivity, and C_m coefficient as shown in figures 5 to 7. Using experimentally determined values, good agreement (within 8 percent) is shown to be feasible.

SUMMARY OF RESULTS

Combustor liner wall temperatures were measured and compared with calculated values. Comparison was made at a fuel-air ratio of 0.019; at inlet total pressures of 10, 20, and 26.7 atmospheres with a constant inlet-air temperature of 589 K (1060° R); and at inlet-air temperatures of 589, 756, and 922 K (1060°, 1360°, and 1660° R) with a constant inlet total pressure of 10 atmospheres. The following results were obtained:

1. Calculated combustor liner temperatures using experimentally determined flame temperature, flame emissivity, and a constant value for the turbulent mixing coefficient of 0.02 in the heat transfer model were approximately 8 percent higher than experimental values.

2. Evaluation of measured flame temperature indicated that a narrow range of flame temperatures were observed experimentally; whereas, a wide range of flame temperatures would be expected from theory.

3. Use of empirical expressions to determine gray-body flame emissivity indicate that emissivity can be evaluated with reasonable accuracy (i.e., 0.42 predicted compared to 0.32 measured at 10 atm) providing the expression accounts for the luminosity of the flame.

4. A turbulent mixing coefficient of 0.02 was required to correlate the data. A constant value of 0.02 was shown to be applicable over a wide range of combustor operating conditions.

Lewis Research Center,
    National Aeronautics and Space Administration,
        Cleveland, Ohio, October 30, 1972,
            501-24.

APPENDIX A

COMBUSTOR TEST FACILITY AND INSTRUMENTATION

The test combustor was mounted in a closed-duct facility shown in figure 10 and tested at inlet-air pressures up to 26.7 atmospheres and temperatures to 922 K (1660° R). Combustion air drawn from the laboratory high pressure supply system was indirectly heated up to 589 K (1060° R) in a counterflow U-tube heat exchanger. The temperature of the air flowing out of the heat exchanger was automatically controlled by mixing the heated air with varying amounts of cold bypassed air. To obtain the higher temperature levels required further heating by means of a natural gas fired vitiating heater mounted downstream of the heat exchanger. Airflow through the heat exchanger and bypass flow system and the total pressure of the combustor inlet airflow were regulated by remote-controlled valves.

Combustor instrumentation stations are indicated in figure 10. The inlet-air temperatures were measured at station 1 with eight Chromel-Alumel thermocouples. Inlet
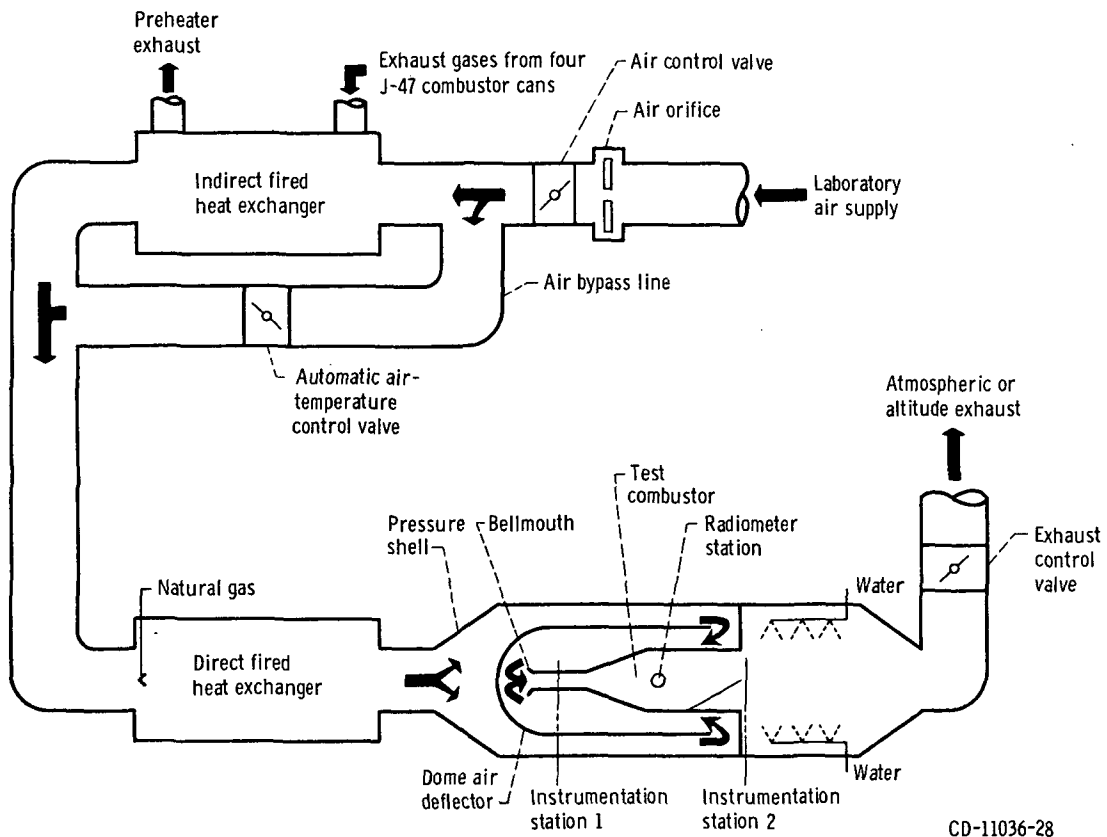

Figure 10. - Test facility and auxiliary equipment.

total pressures were measured at the same station by four stationary rakes consisting of three total-pressure tubes each. The total-pressure tubes were connected to differential pressure strain gage transducers that were balanced by wall static pressure taps located at the top and bottom of the duct. Combustor outlet temperatures and pressures were measured with a traversing exhaust probe mounted at station 2. The probe consisted of twelve elements; five aspirating platinum-platinum 13 percent rhodium total temperature thermocouples; five total pressure probes, and two wedge-shaped static pressure probes.

APPENDIX B

SYMBOLS

A   area

b   fin thickness

C   convective heat transfer component

C_m turbulent mixing coefficient

C/H   carbon-hydrogen ratio

e   emissivity

e'   emissivity term defined by eq. (9)

f/a   fuel-air ratio

h   heat transfer coefficient

k   thermal conductivity

L/D   length of immersed sheath/diameter of sheath

l width of cooling slot

l_b mean beam path length

M   mass flux ratio, $\rho_c U_c / \rho_h U_h$

$\dot{m}$ mass flow

Nu   Nusselt number

Pr   Prandtl number

p   combustor pressure, atm

R   radiation heat transfer component

Re   Reynolds number

s   equivalent slot height, A_s/width of cooling slot

T   temperature

T'_w wall temperature as defined by eq. (1)

U   gas velocity

η film cooling effectiveness

ξ primary-zone combustion efficiency

ρ mass density

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σ Stefan-Boltzmann constant

φ equivalence ratio

Subscripts:

ad, w   adiabatic wall

c   coldstream

f   flame

h   hot stream

t   total

w   wall

x   distance

1   flame side of combustor firewall

2   coolant side of combustor firewall

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