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Gas Turbine Prediffuser-Combustor Performance During Operation with Air-Water Mixture

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16. Abstract

In a continuing effort to establish performance changes due to water ingestion into an aircraft gas turbine engine and possible design improvements, an experimental investigation was performed with a model gas turbine prediffuser-combustor sector utilizing a number of mixture and flow conditions in a tunnel operating with a two-phase, air-liquid film-droplet mixture. For given entry conditions into the prediffuser (which can be related to the exit conditions of the core compressor in a bypass engine, and, therefore, also to ingestion conditions at the engine face) the two main issues are (1) the amount of water entering the primary zone of the combustor, and (2) the local reduction in temperature, flame-water interactions, and the vitiation caused by the vaporizing of water. Flow visualization and estimates of water flow and droplet size in the primary zone have been undertaken under cold flow conditions. The amount of water entering the primary zone has been found to be a complex function of (1) the air-water mixture conditions at entry to the prediffuser, and (2) the effects of gravity on the flowfield for given geometry of the prediffuser-combustor, and the flow split between the primary and the coolant streams in the combustor. Combustion tests have been carried out to establish the effects on performance, occurrence of flameout, and recoverability of combustor exit temperature by enhancement of one or both of the fuel equivalence ratio and the oxygen content of air. It was tentatively concluded that the observed effects of the presence of water are a result of the interactive effects of heat transfer to water, vitiation by water vapor on combustion, and the total heat release in the primary zone. The latter being the dominant factor in stabilizing the flame with a large increase in combustor exit temperature, as found in the tests with the addition of both fuel and oxygen to a given burning mixture.

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Any opinions, findings, conclusions or recommendations expressed in this material are those of the author(s) and do not necessarily reflect the views of the Federal Aviation Administration.

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NOMENCLATURE

D_w	mean volumetric drop size, µm
F_{PL}	combustor pressure loss factor
F_{TI}	combustor temperature increase factor
$\dot{m}_a$	mass flow rate of air, kg/s
m	vitiation ratio
n	overall reaction order
N	flow rate of gas into a reactor, mol/s
P	pressure, atm
P_0	stagnation pressure, Pa
Su	flame speed
Su (O_2 = 21)	maximum flame speed
T_{air}	test rig inlet air temperature, ^oC
T_f	fuel temperature, ^oC
T_w	injected water temperature, ^oC
T_0	stagnation temperature, ^oC
V	volume, liter
X_{wd}	mass fraction of water in spray form
X_{wf}	mass fraction of water in film form
X_{O2}	mass fraction of gaseous oxygen
ϕ	fuel equivalence ratio
ϕ_f	local fuel equivalence ratio
ϕ_{O2}	oxygen equivalence ratio
η_c	overall combustor efficiency
η_h	heat transfer efficiency
η_v	heat transfer efficiency for vaporization

Subscripts

a	operation with air only
a+w	operation with air and water
s	stoichiometric
3	prediffuser inlet location
4	combustor exit location

LIST OF ILLUSTRATIONS

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EXECUTIVE SUMMARY

Two of the commonly reported events during the operation of a bypass jet engine in a rain or hail storm are (i) the loss of power and (ii) the difficulties of maintaining a flame in the burner. While the loss of power may also be a consequence of the increased loss in the low and high pressure compressors due to air-water mixture operation, both the loss of power and the possibility of occurrence of flame-out conditions are a direct consequence of the changes in the performance of the prediffuser-combustor due to the presence of water in the air. The air-water mixture entering the prediffuser-combustor section, after exiting the core (or high pressure) compressor, may consist of an air-water vapor mixture that may also include liquid water. The liquid water may be in film form over the casing wall and also, in droplet form in the body of the air flow, depending upon the engine operating (or power setting) conditions, and the mass fraction and droplet size distribution of water entering the engine. Among the engine power settings, the most severe conditions are believed to be the flight idle condition during aircraft descent, and the ground idle condition.

An investigation has been carried out on a sector of an annular prediffuser-combustor of a typical bypass jet engine with a core engine and a supercharger. The main aspects of the investigation are (i) determination of the distribution of water in the prediffuser-combustor when supplied with various types of air-water mixtures, and (ii) determination of the performance of the combustor under burning conditions. The latter has been undertaken with water supplied at low temperature (about 7 C) and typical ground temperature (about 20 C). In addition, in view of the interest in the recoverability of the combustor exit temperature to the value obtained during operation with air, while an air-water mixture is being supplied to the prediffuser, investigations have been conducted on the effects of increasing the fuel equivalence ratio, and also the oxygen content of air, by addition of oxygen to air, both individually and together. In all cases, water has been added to the air flow in the form of (i) film, (ii) droplets, and (iii) a combination of film and droplets, the mass fraction of film being 2.0 per cent and that of droplets varying from 2.0 to about 12.0 per cent. The corrected mass flow of air has been chosen to correspond to either the flight idle or ground idle condition.

The experimental studies have included flow visualization as well as measurements, both under cold flow and burning conditions. The flow visualization has been utilized to observe the distribution of water and water entities (droplets and ligaments) in cold flow tests, and the extent and stability of the flame under burning conditions. The measurements pertain to (a) the water concentration in the primary zone under different air-water mixture supply conditions, and (b) the temperature rise and the pressure loss across the prediffuser-combustor, expressed in terms of the temperature increase factor and the pressure loss factor, respectively.

As a prelude to tests with the (three-dimensional) sector of the annular combustor, extensive cold flow tests have been conducted with two-dimensional prediffuser-combustor models. A summary of those investigations is included in this current report.

In the cold flow tests with the three-dimensional sector, it is observed that the size and spacewise distribution of the liquid entities in the combustor is a complex function of (i) air water mixture conditions in the prediffuser, (ii) the air mass flux and distribution, and (iii) the design parameters of the swirl cups and primary jets. The distribution of water in an annular combustor is strongly affected by gravity on the flowfield in the secondary streams in the outer section of the combustor.

In the burning tests, the main findings are: (1) There is an appreciable drop in the combustor exit temperature with a small mass fraction of water, and only a moderate loss in that temperature as the mass fraction is increased. (2) The effects of gravity on film motion and the resulting changes in combustion performance become substantially increased for the cup that is most displaced in the gravitational direction. (3) Attempts at recovery of exit gas temperature by an increase of fuel equivalence ratio have proved successful with low mass fractions of water in either film form or large droplet spray form, when the water could be expected to enter the primary zone mainly through the primary jets. (4) The effect of enrichment of air with oxygen is always a gain in combustor temperature unless the increase in heat loss to water and vitiation due to vapor generation tends to reduce the gain to a negligible value; and (6) the combined effects of an increase in fuel equivalence ratio, and an enrichment of air with oxygen is to permit a regain of the temperature to the value obtained during operation with air (only), even up to an injection of about 4.0 per cent water.

Based on current results, it can be concluded that during air-water mixture operation, heat loss to water, through heating and vaporization, resulting in a possible decrease in flame temperature in the primary zone, appears to be the dominant process leading to temperature loss across the combustor. The results of the tests performed with modified fuel equivalence ratio, and oxygen-enriched air may also suggest in the case of a practical engine a number of operational changes, for example in the form of engine speed regulation for different power demand settings, that can yield the same benefits as additions of fuel and oxygen.

1. INTRODUCTION

Water ingestion into an aircraft gas turbine engine can occur due to (i) rain, occasionally mixed with hail, entering the engine inlet during flight through a rain storm; and (ii) wheel-generated spray clouds entering the engine inlet during take-off and landing from a runway with puddles of water. It has been shown [references 1 and 2] in the case of a turbofan engine that the core engine compressor (usually the high pressure compressor) exhaust may consist of an air-water vapor-liquid water mixture, and the prediffuser-combustor may have to operate with that mixture.

During water ingestion, water may appear at the entrance to the prediffuser in the form of a film over casing surfaces as well as droplets in the body of the flow. The film flow may vary in different cases with respect to thickness, speed, and stability. Similarly, the variation in geometry, size, and temperature distributions of water droplets can be considerable at the exit from the high pressure compressor, following modifications of the characteristics of the water droplets in the atmosphere through the inlet, the fan, the low, and the high pressure compressors. The changes in the performance of the prediffuser and combustor, under conditions of water ingestion, are therefore complex, both in regard to the number of influencing parameters as well as the manner in which water may affect the flowfield, combustion, and chemistry in the combustor. The flowfield changes occur in the prediffuser and the different streams of the combustor. Combustion, as well as chemical kinetics, is affected due to the reduction in local temperature caused by the presence and vaporization of water, and also, the vitiation by water vapor. This leads to adverse effects in the engine performance, in particular, the engine may surge, or may suffer flameout or may have unsteadiness of the flame in the combustor. In the later case of combustor performance deterioration, it is then of interest to establish the flowfield and combustion characteristics in a three-dimensional, actual combustion chamber operating under simulated water ingestion conditions.

1.1 BACKGROUND

An air-water mixture ingested into an engine may be characterized by the following: (1) pressure and temperature of the air; (2) temperature, mass fraction and physical state of the water; (3) saturation with respect to water vapor at the given mixture temperature; (4) difference in the temperature of the air and water, and (5) difference in velocity between the air and water droplets. In general, during flight operations in a rain storm, the air-water mixture may be assumed to be fully saturated with respect to the water vapor. The water content by weight, during a rain storm, can vary from a small value such as 0.5 per cent to a large value such as 18.0 per cent. A water mass fraction of 18.0 per cent would correspond to a storm conditions, and under such extreme conditions there may also be hail and snow ingested into an engine. There is a large variation in the mean volumetric diameter of the water droplets ingested into an engine during a rain storm. The droplet sizes may be of the order of 100 to 1,500 microns, although 3,000 micron size droplets have also been reported. The large droplets can be expected to move independently of the air in direction, but the small droplets will follow the air streamlines to some extent.

Investigations have been conducted at Purdue University over several years in the subject of water ingestion into engines [references 1 through 7]. In an investigation carried out by Tsuchiya and Murthy [references 1,3 and 4], an analysis of the effects of water ingestion into a compressor was carried out leading to the development of a predictive code, the PURDU-WINCOF code. A series of tests were also carried out on a small, test compressor with air-water droplet mixtures. The experimental results were compared with predictions, and it was

concluded that the basic effects of water ingestion into compressors arose through blockage, distortion, and heat and mass transfer processes, with the changes in blade aerodynamic performance being relatively small. In subsequent work carried out by Haykin and Murthy [reference 2], an attempt was made to establish the transient performance of a generic, high bypass ratio, two-spool gas turbine engine under a variety of water ingestion and power setting conditions, and with a temperature sensor, that is utilized for providing input to the engine control, becoming flooded by water and recording lower temperature than the local gas phase temperature. The principal tools utilized in the investigation were the PURDU-WINCOF code and an engine simulation code. It was shown in general that engine performance is degraded during operation with water ingestion and the amount of degradation is a nonlinear function of inlet water mass fraction; controllability of the engine with respect to operator-initiated power settings changes is affected by water ingestion; and errors in the temperature sensor providing an input to engine control lead to instability in engine operation, eventually causing a limiting condition or parameter to be exceeded. Later, an attempt was made by Murthy and Mullican [reference 5] to establish the time-dependent nature of effects of water ingestion in an axial-flow compressor, and a generic high bypass ratio engine with a control. In view of the large effects arising in the air compression system and the prediffuser-combustor unit during water ingestion, attention was focused on those effects and the resulting changes in engine performance. It was established that under all conditions of operation, whether water ingestion was steady or not, that water ingestion caused a fan-compressor unit to operate in a time-dependent fashion with periodic features, particularly with respect to the state of water in the span and the film in the casing clearance space, at the exit of the unit.

Finally, an investigation was carried out by Minster and Murthy [references 6 and 7] in which a number of two-dimensional prediffuser-combustor model configurations were studied with air-water mixture flow in a two-dimensional, two-phase flow tunnel. The two-dimensional, diffuser combustor configurations employed in the tests are illustrated in figure 1. Each of the test models included a diffuser section, a dome section with three fuel injector swirl cups, a main chamber which contained the primary and secondary zones, two (outer and inner) sections through which the primary and dilution (or secondary) jets flowed, and means for discharge of flow from the different sections. Model I differed from Models II and III with respect to the area ratio between the inner and the outer streams. Models II and III differed from each other only in the geometry of the diffuser. For Model I, the injector cups had no net swirl, while Models II and III used injector cups with swirl. The air-water mixture conditions corresponded to those obtained at the core compressor exit of a generic turbofan engine that is exposed to water ingestion at the inlet of the engine [reference 5]. Through flow visualization and detailed measurements, the redistribution of water, entering the diffuser in film and droplet form, was established in the diffuser, at the dome, and in the primary zone of the models. An attempt was made to establish the effects of evaporation of such water in the primary zone on flame speed and flame stability based on a proposal of Odgers [reference 8] for dealing with combustion under conditions of appreciable vitiation.

Through visualization of the flowfield in the diffuser at the dome section, two notable features were observed: (1) the shape and characteristic size of water entities that entered the injector cups, and (2) the flow split. In the case of film injection, the water entered the injector cups, following break-away from the walls, mostly in the form of large entities of indefinite shape. In the case of spray injection, and also in the case of a mixture of film and spray injection, there was a greater tendency for water to enter the injector cups in the form of droplets of relatively small diameter. The amount of water in the outer and the inner stream also differed from each other between the case of film injection and the two cases of injection of spray primarily due to the low

momentum film becoming diverted by the spray. As the air flow was increased, it was pointed out that the total amount of water entering through the cups tended to increase.

Downstream of the injector cups of Model I, the water was in the form of a fine mist consisting of a very small diameter droplets moving at a high velocity. In the region of the primary jets, the small droplets were observed to slow down and coalesce with the droplets in the primary jets. This resulted in larger droplets occurring in the region adjacent to the primary jets. Downstream of the injector cups of Models II and III, the water was present in the form of a spray with some angular momentum in the case of swirl cups. Moving downstream of the cups, the droplets underwent substantial collisions with one another as well as the droplets that fell off the boundary walls. This resulted in coalescence of the water droplets into fairly large drops and, often, non-spherical entities.

The flow through the primary jets into the primary zone was found to depend on the air flow and water split at the dome, and the nature of the flow issuing out of the injector cups. The water entities in the primary jet were generally large and tended to break up following impact of opposing jets.

Concerning the redistribution of water in the prediffuser-combustor models, figures 2 through 5 present data from measurements of the fraction of water that entered the primary zone as a function of mass fraction of water injected in the three models under different conditions.

Film Injection Case (figures 3 and 5): The flow rate of water into the primary zones of all three models increased with an increase in the mass fraction of injected water. At low air flow rates, the amount of water entering the primary zone of Model III was slightly larger than that of Model II, while at higher air flow rates, it was slightly less than that of Model II. This was attributed to the influence of the diffuser configurations on the length of the attached film. The film remained attached to the top wall of Model III (dump diffuser configuration) over a greater length than Model II (stepped diffuser). Thus, at low air flow rate conditions, a larger fraction of the detached film impacted the cups, and at higher flow rates, the detached film tended to flow over the dome section into the secondary stream. The length of film attached to the top surface of Model I (single divergence diffuser) was longer than that of Model II. The stepped diffuser configuration Model II had the shortest length of attached film of the three Models.

Spray Injection Case (figures 2 and 4): For both low and high air flow rates, and small and large water droplets, the amount of water entering the primary zones of all three models increased nearly linearly as the mass fraction of water (X_{wd}) increased. The increase was less for high air flow rates. At low air flow rates, there was no recognizable difference between the water entry in Models II and III. The flow rate of water into the primary zone of Model I was significantly larger than that of Models II and III. The fraction of injected water entering the primary zone of Models II and III decreased slightly with an increase in X_{wd}, whereas in Model I it increased. At high air flow rates, the results of all three models displayed similar trends except that, again, the amount of water entering the primary zone of Model I was larger than that of Models II and III.

Spray and Film Injection Case (figure 3 and 5): The combination of film and spray injection into the test models resulted in a 10 to 40 per cent increase in the water entry into the primary zone (compared to the individual injection of film or spray). For all three models, the rate of water entry into the primary zone increased significantly with an increase in air flow rate. Also, as the mass fraction of spray increased, the fraction of injected water entering the primary zone leveled off and slightly decreased. At low air flow rates and small water droplets, the flow rate of water

into the primary zones of Models II and III remained nearly constant whereas it increased for Model I with an increase in X_{wd}.

The flowfield in the primary zone could be characterized, by processing the video picture frames, in terms of the total quantity of water entering the zone and the droplet distribution in terms of size. It was shown that vitiation of air with water vapor corresponding to the estimates of water droplets that could undergo vaporization in the primary zone of the test models was adequate to lead to a reduction in the laminar flame speed and flame stability limits. Specifically, it was found that in regard to Models II and III, there did not seem to arise a significant effect of vitiation at conditions of low air flow rates (0.45 kg/s), while at high air flow rates such as 0.91 kg/s, the effect of vitiation on the selected combustion parameters seemed to be substantial.

1.1.1 Diffuser Entry Conditions.

The entry conditions of the air-water mixture at the prediffuser-combustor section are determined by the initial state of the mixture in the atmosphere, and the modifications of the state in the inlet and the compressor. The air-water mixture at the inlet of the prediffuser can be characterized by the same state properties as at entry to the engine inlet, namely, water mass fraction, water droplet size distribution and temperature and velocities of air and water.

It has been found that water exiting a compressor operating under conditions of water ingestion appears in two forms: a film that flows along the casing of the compressor in the blade tip-casing clearance and a spray of various droplet sizes in the span of the blades. The generation of film and spray can be attributed to the cumulative effects of the following different processes occurring in the compressor: droplet break-up and coalescence, centrifugal action towards the casing, and heat and mass transfer. Work input in the compressor and heat and mass transfer processes between the water and air give rise to a substantial increase in the vapor content of the air at the compressor exit, although the air (at high pressure) does not become saturated with water vapor. Although the aerodynamic performance of the prediffuser-combustor section may not be affected to a large extent by the presence of water vapor, the presence of water, in both liquid and vapor form, can be expected to affect the combustion processes in the combustor [references 2 and 8].

An earlier investigation conducted at Purdue University [reference 5] established, as stated earlier, that the performance of turbomachinery becomes time-dependent during water ingestion. Hence, the entry conditions to the prediffuser can be expected to be time-dependent. However, this aspect of the problem is not addressed in the current investigation.

1.1.2 Nature of Problems in the Prediffuser Section During Water Ingestion.

During water ingestion into an engine, the flowfield entering the prediffuser consists of a water film flow at the outer wall and an air-water droplet mixture flow in the span. The water film arises as a result of the centrifuging action of the rotating compressor on the water droplets, and is in motion due to the shearing action of the adjacent air-water mixture flow. The thickness and velocity of the film undergo changes along the flow as a result of gravitational action, diffusion of the flowfield and deposition of droplets from flow in the span. Eventually, when the film velocity becomes sufficiently small, the film detaches from the diffuser wall and falls into the body of the air flow. The path line and speed of the film-generated ligaments and droplets depend upon (1) droplet and ligament size, (2) mass fraction of spray that exists in the air-water mixture prior to film break-up, (3) air flow velocity, and (4) gravitational action on the droplets.

In nearly all cases, it can be expected that some portion of the detached film will impact on the dome section, with some of the water being bypassed into the inner and outer streams and entering the primary zone through the primary jets. Several important features pertaining to the prediffuser flow include: (1) the development of the film at the wall and its subsequent detachment; (2) the division of flow into primary and secondary streams at the dome of the combustor with changes in air-water mixture composition, and (3) the amount of water entering the primary zone as a fraction of the total water in film and droplet form.

1.1.3. Nature of Problems in the Combustor Section During Water Ingestion.

In certain circumstances, the addition of water into a combustor is desirable for obtaining additional mass for thrust augmentation, and reduction in flame temperature which leads to a reduction in NOX formation [reference 9]. However, it is crucial to note that there is a difference between the ingestion and the injection cases in that the ingested water entering the combustor is in the form of droplets and ligaments of arbitrary size and shape, while water added by design is injected at a specific location, and in fine spray form and generally at elevated temperatures.

The nature of the problems that occur in the combustion chamber during water ingestion include: combustor performance deterioration in terms of increased loss of stagnation pressure, reduction in combustor exit temperature, and combustion efficiency; heat transfer to the liquid phase, and the resulting mass transfer to the gas phase; and local and global quenching of the flame. The combustor of the type investigated here operates with unmixed fuel and air, that mix and burn in the primary zone, roughly the region between the dome with the injector cups and the vicinity of the primary jets. The effects of the presence of water along with air are also more significant in the primary zone than elsewhere from the point of view of combustion-related processes.

Given the mass flux of air and water, and the fuel equivalence ratio at a certain instant in time within the combustor, a limiting case may be visualized in which there is adequate heat release, and all of the water is in a discrete state that permits instantaneous evaporation of water, without the possibility of any other interactions between air and water or among the water entities. In this limiting case there arise a reduction in gas phase temperature, ΔT_{0a}, and a change in vitiation ratio due to water vapor addition, Δm.

In a given gas turbine combustor, such a limiting case may be associated with a certain regime of operations, characterized by the mass flux of air and the fuel equivalence ratio, and the mass fraction of water admitted in spray and film forms at a certain temperature. However, the same values of ΔT_{0a}, and Δm may be obtained in a number of other regimes of operation representing combinations of mass flux of air, fuel equivalence ratio, mass fraction of water in spray and film forms, and temperature of water.

1.1.4 Recoverability of Combustor Exit Temperature

One of the main concerns in the performance of a gas turbine combustor is the combustor exit temperature which is a significant parameter in the operation of a gas turbine engine. It is, therefore, important to establish if the reduced combustor exit temperature obtained during operation with an air-water mixture can be increased to the value that would be obtained during operation with air, under otherwise identical conditions. Among several methods of increasing the combustor exhaust temperature, two that are considered of special interest are increasing the fuel equivalence ratio, and enrichment of air with oxygen. One may also consider the possibility

of combining the enrichment of air with oxygen with increasing the fuel equivalence ratio. It is clear that there are combustion-related and practical limits to the increase in fuel equivalence ratio as well as addition of oxygen. The fuel equivalence ratio may affect the total amount of fuel burnt. At the same time the combustion efficiency is modified by the fuel equivalence ratio. The enrichment of air with oxygen may lead to an improvement in combustion efficiency and some increase in heat released. In the case where enrichment of air is combined with increase in fuel equivalence ratio, one may expect a definite increase in heat release.

The process of increasing the combustor exit temperature from a reduced value obtained during air-water mixture operation to the value obtained during air operation of the combustor may be referred to as the process of combustor exit temperature recovery. The 'recovery tests', discussed in Chapter 2.8 and onwards, refer to the tests conducted to establish the recoverability of combustor exhaust temperature.

It may also be pertinent to point out that the investigations with additional fuel and oxygen provide important opportunities for examining the effects of the presence of water during the combustion process.

1.2. COMBUSTION IN THE PRESENCE OF WATER

The effects of addition of water may be divided into several parts: (i) Direct effects on mixing and the formation of the recirculation zones upstream of the primary jets; (ii) heat and mass transfer causing gas phase temperature reduction, and vitiation with water vapor; (iii) chemical-mechanical interactions between the flame and discrete entities of water; and (iv) direct effects on reaction kinetics. The operational parameters affecting the performance of the combustor, are the following: (i) the state properties and mass flow of air; (ii) the temperature and equivalence ratio of fuel; and (iii) the temperature, the physical characteristics, and the motion (relative to air and flame) of water entities. The parameters may be chosen and identified, independently of one another, in a given operation or a test. However, the flow, heat and mass transfer, and combustion processes in the primary zone of the combustor depend on the parameters variously in a highly complex fashion. Furthermore, the processes themselves are interactive. The processes (namely, physical, flame-related, and chemical) occur in different parts of the combustor with several characteristic length and time scales in the gaseous and the liquid phases.

During operation with an air-water mixture, the characteristic parameters affecting the combustor performance may be chosen as follows within the combustor:

1. the size distribution of water entities,

2. the total mass and volume of water present at an instant in time,

3. the velocity distributions of the entities in relation to air velocity and flame speed,

4. the separation distance distribution for the entities, and

5. the residence time distribution of the entities in relation to the residence time of the gas phase.

Another parameter affecting the performance is the difference in temperature between the gaseous products and the liquid phase, the gas temperature being a function of the fuel equivalence ratio and the heat release effectiveness. Although no detailed investigations have

been conducted in the past in burners with discrete entities of water, a background exists in literature on heat and mass transfer, flame straining, droplet deformation and break-up, and vitiation [references 10 through 19].

Odgers [reference 13] developed a methodology for determining the possibility of a flameout due solely to the effect of vitiation of the air. Odgers developed his model by utilizing data obtained from premixed combustion experiments in a spherical combustor. The air was vitiated in those experiments with nitrogen. In later extensions to the model [references 8 and 14], in order to incorporate the effect of vitiation of water vapor, the role of water vapor in a vitiated air mixture was assumed to be identical to that of nitrogen, that is, a simple inert diluent in combustion, lowering the flame speed to the extent that it acts as a heat sink, and reduces the flame temperature. The effects of vitiation were expressed in terms of changes in laminar flame speed, and flame stability limits.

The laminar flame speed was related to the reaction rate. The flame stability limits were provided by means of a map of oxygen-fuel equivalence ratio as a function of a so-called combustor loading parameter (N/VP^n, where N represents the air mass flux, V being the volume of the combustor, P being the static pressure in the combustor, and n representing the overall reaction order), with the vitiation ratio, m, as a variable parameter. Based on experimental data (obtained with a well stirred reactor), Odgers concluded that the flame velocity could be expressed as a function of m, which is equal to the molar ratio of the sum of the quantities of water vapor and nitrogen to the molar quantity of oxygen in a given combustion environment. The loading parameter was related to a function of the following combustion-related quantities: gas temperature, equivalence ratio, the vitiation ratio, and various other parameters pertaining to the combustion mixture composition.

Muller-Dethlefs and Schjlader [reference 15] studied the effect of steam on the burning velocity of premixed propane and ethylene flames and concluded that steam did not act as an inert diluent, but it gave rise to a greater heat release which counteracted the cooling effect of the added steam. No mechanism was offered.

Kuehl [reference 16], and Korol and Mulpuru [reference 17] investigated the effect on flame speed due to the changes in chemical reactions produced by the presence of water vapor. It was, however, not convincingly established that the flame speed, for example, is affected by the water vapor solely due to either of two parameters considered, namely the radiative heat loss, or the changes in chemical reactions.

More recently, work carried out by Pellet, Jentzen, Wilson and Northam [reference 18] on the effect of water vapor on H_2-O_2 diffusion flames, showed that the presence of water vapor in the air results in a decrease in the flame extinction limit. This contrasted strongly with the analogous substitution of water for N_2 in a premixed flame, which was shown by Koroll and Mulpuru to lead to a significantly kinetically-controlled increase in laminar flame speed. A spread of more than 25 per cent was found by Pellet et al., to exist between the quenching effect of water in a (relatively cool) diffusion flame at flameout and the combustion enhancing effect of water in a (relatively hot) premixed flame.

Finally, Williams [reference 19] presented a review of the work pursued by himself and his co-workers in the area of extinction of laminar flames. The review concentrated mainly on the use

of inert materials, such as H_2O, CO_2 and N_2, as suppressants and a means of extinguishing laminar diffusion flames. In this connection he presented the view that the ultimate extinction of a flame was due to excessive heat loss either to the rich or the lean side of the flame, or both. This heat loss, it was also pointed out, could be enhanced by the presence of a suppressant, for example water, or by a large flame stretch. Thus, he concluded that evaluation of the heat loss to water and the subsequent decrease in flame temperature were important in understanding flame extinction. Two factors pointed out by Williams as worthy of consideration were the non-uniformity introduced by turbulent mixing in the high speed flow, and differential diffusion of the liquid water suppressant and the gaseous (fuel, oxidizer and product) species. Due to these effects, flamelets at one location might be subject to very high quantities of suppressant while flamelets at another location may not see any suppressant at all. The separate flow phenomena could have significant effects on the extinction properties, and also, possibly radiation, due to their nonlinear dependence on flame cooling.

1.2.1 Combustion with Fuel and Oxygen Addition

A possible method of increasing the heat release in the primary zone of the combustor, in an attempt to recover the temperature drop that may have occurred across the combustor as a result of operation under air-water mixture conditions, is to increase the fuel equivalence ratio, enrich the air with oxygen, or utilize a combination of both. Operation under these modified conditions can be expected to result in modifications in the combustion processes caused by (i) increased combustion activity and rise in local temperature, (ii) the resulting vaporization of the water, when the quantity of water and the droplet sizes in the primary zone are small, and the simultaneous increase in vitiation due to water vapor, and decrease in gas phase temperature, and (iii) simple heating of water, when the quantity of water and size of water entities are large, and decrease in gas phase temperature.

The three types of processes are obviously interactive, and in a given case, there is a net effect on combustion. The three processes depend not only upon the total amount of water entering the primary zone but also on the manner in which the water enters the primary zone. For example, water may enter the primary zone in part through the injector cups and in part also due to recirculation of air-water mixture that enters through the primary holes. It must be noted here that the rest of the water in the original mixture enters the combustor further downstream through the dilution or secondary holes. However, the different parts of the water entering the primary zone have different physical characteristics, and, therefore, undergo and also cause different types of changes in combustion.

The operating condition with an increased fuel equivalence ratio may be related to the condition of operation using air enriched with oxygen. There are both similarities as well as differences between the two cases. When the fuel equivalence ratio is increased, there arises, also, an increase, locally, in combustor temperature, a resulting increase in water heating and vaporization, an increase in vitiation due to water vapor, and a decrease in gas phase temperature. However, increase of fuel equivalence ratio gives rise to increased combustor temperature due to the direct effect of fuel addition, with a possible decrease in combustion efficiency, while enrichment of mixture with oxygen improves local combustion efficiency and hence gives rise to an increase in combustor temperature. The combination of both an increase of fuel equivalence ratio and enrichment of air with oxygen could thus be expected to result in a larger increase in heat release due to an improvement in combustion efficiency, and hence, the quantity of fuel burned, as a result of the increased oxygen content in the air and a simultaneous increase in fuel equivalence ratio.

The possibility of recovering the combustor exit temperature in an environment of operation with an air-water mixture by the addition of fuel and oxygen, either separately or together, has significant implications to flight operations of an engine. At the same time, the experiments provide opportunities for understanding the detailed processes and mechanisms involved in combustion in the presence of liquid water.

1.2.2 Effect of Air and Fuel Temperature

An increase in the initial air temperature will increase the flame temperature. However, it has been shown [reference 20], that only about one-half of an increase in initial temperature is translated into an increase in flame temperature. Although the increase in initial air temperature raises the flame temperature, the extent of increase is less than anticipated, owing to additional energy being stored in the form of dissociated species, mainly CO and H_2. For the case of increased air temperature combined with addition of liquid water, the increase in flame temperature leads to larger quantities of water undergoing vaporization, and hence, higher levels of vitiation. In the current facility, the heated air requirements are met through pre-combustion, which results in a vitiated air supply (that is, abnormally high H_2O and CO_2 concentrations and lower O_2 concentrations). In both cases, the amount of fuel that can be burned per unit mass of mixture is reduced by the lower oxygen concentration. A secondary effect, also reducing the flame temperature, is the higher specific heat resulting from higher levels of CO_2 and H_2O in the products.

The fuel utilized in the investigation is JET-A. For convenience, some of the properties of the fuel are summarized in Table 1 [reference 20]. Concerning fuel temperature, an increase in the fuel temperature has a large effect on the fuel evaporation rate [reference 20]. In general, the evaporation rate increases with fuel temperature, partly because of the higher volatility but also because of finer atomization due to the reduction in viscosity. The higher evaporation rate leads to larger quantities of vaporized fuel in the primary zone which, in turn, results in higher amounts of heat release. In the case of combustion with an air-water mixture, the increase in local heat release may lead to larger levels of heat transfer to the water and a corresponding drop in gas phase temperature due to the vaporization of water.

TABLE 1. REPRESENTATIVE SPECIFICATION DATA FOR JET-A

Specification		JET-A
Flash point, ^oK	(min)	311
Density at 288 ^oK, kg/liter	(min)	0.775
	(max)	0.8930
Freezing point, ^oK	(max)	226
Viscosity at 253 ^oK, m^2/s	(max)	8.0×10^{-6}
Specific energy (net), MJ/kg	(min)	42.8

1.3 COMBUSTOR OPERATING CONDITIONS OF INTEREST

Typical aircraft fight operations that may be affected by water ingestion include take-off, cruise, descent, and landing. The engine operating condition, in particular, the throttle setting or the power setting during the foregoing flight operations will be different for each case. During take-

off, there is a possibility of rainfall being combined with other atmospheric events, however, the engine power output is at a maximum. In the cruise condition, the aircraft can be expected to operate at altitudes higher than those where rain and hail storms normally occur, and the engine power demand is comparatively low. Few incidents are reported during cruise operation. During a descent operation, the aircraft may pass through altitudes at which rain and hail storms may occur, and the engine power demand is substantial while the power setting may correspond to flight-idle operation. In view of the particularly critical conditions expected during a descent operation with flight-idle conditions, there is considerable interest in the performance of the combustor at power settings equal to or less than that of the flight descent setting. In the current investigation, the main interest is flight-idle and idle settings of the engine.

1.4 ISSUES

The major problem in the prediffuser-combustor element during ingestion of water into an engine is that water enters the prediffuser, along with air, in generally arbitrary state, form, and quantity. Several issues may then be identified as affecting the performance of the combustor, primarily the temperature increase, the pressure loss and the flame stability:

1. The prediffuser performance becomes modified on account of the presence of water in vapor, film, and droplet form. The state of the air-water mixture becomes modified at the diffuser exit.

2. In the space between the exit of the prediffuser and the combustor dome, there are further changes in the state of the mixtures as well as in its velocity distribution. These changes, affect the flow split and the supply of the mixture to the injector cup section.

3. The modifications in the flow through the cup and in the flow split affect the state of the mixture in the primary zone. Also, the flowfield in the primary zone becomes modified.

4. The flow that forms the outer and the inner secondary stream enters in part the primary zone and the dilution zone, and the rest of the flow is utilized to cool the turbine. The water in the secondary flow is affected by gravitational action.

5. The primary zone operates with a particular value of fuel equivalence ratio, that may be referred to as the local equivalence ratio, which is considerably larger than the nominal equivalence ratio at which the combustor may be specified to be operated.

6. The fuel distribution in the different parts of the primary zone may be affected by the distribution of water entering the zone, which itself may be affected by the air flow velocity. It should be noted that the residence time of the mixture is inversely proportional to air flow velocity.

On ignition, a series of processes occur that are strongly interactive and almost simultaneous, although each of them has characteristic time and length (or velocity) scales. These processes may be identified as follows: (i) flame formation and heat release; (ii) mixing, and flame propagation; (iii) gas phase temperature increase; (iv) heat transfer to liquid phase; (v) phase change of part of the liquid; (vi) mixing, and generation of final value of mixture temperature and pressure; (vii) escape of unburnt hydrocarbons; and (viii) escape of variously heated water in the exhaust and in the secondary stream supplied to the turbine.

The progress of the above multi-faceted and complex processes is the major issue in the combustor element.

One method of understanding the above issue is to proceed as follows:

a. The combustor may be divided into the primary zone and the dilution zone.

b. The combustor may be said to operate with a combustion efficiency, with reference to the heat released, η_c. It is clear that η_c is affected by the initial vapor content in the air-water mixture, the vapor that becomes generated by the evaporation of water, the reduction in the gas phase temperature due to heat release to the liquid phase, and the mechanical effects of the presence of water entities. The combined effect of these processes results in changes in the extent of completion of the combustion reactions and the quantity of unburnt hydrocarbons in the exhaust gases.

c. The heat transfer to the liquid phase and vaporization may be said to occur with efficiencies η_h and η_v, respectively. They are functions of the difference between the temperatures of the gaseous and liquid phases, the total amount of water present in the primary zone as a fraction of the gas phase, and the form in which water is present. A small quantity of water in fine droplet form, for example, affects the processes of heat transfer and the combustion very differently from a large quantity of water in fine droplet form or in large drop and ligament form.

The interactions among (a), (b), and (c) may be explained with reference to figure 6. The majority of the combustion processes occur in the primary zone of the combustor; thus, the quantity of water present in the primary zone has a large influence on the heat release in the primary zone. The overall combustor performance can be described in terms of (i) the local combustion efficiency, (ii) the resultant heat release, and (iii) the quantity of unburned fuel exiting the combustor. When a combustor operates with an air-water mixture, comprised of both liquid and vapor, a large fraction of the air-water mixture can be expected to enter the primary zone. Since the liquid water entities move slower than the air stream, an accumulation of water arises in the primary zone. The increase in the residence time of the water droplets in the primary zone, and subsequent flash evaporation, give rise to pockets of steam, and a reduction in the gas phase temperature, in the neighborhood of the vaporized water droplet. This results in a reduction in the heat release (or increase in inefficiency) and reaction rate, and an increase in the quantity of unburned hydrocarbons. The drop in flame temperature results in less heat being available for the heating and evaporation of water. The air-water mixture in the dilution zone tends to reduce the gas temperature further as a result of heat transfer to the water in the dilution zone. Thus, there exists a balance between the amount of heat release and the heat transfer to water during combustion with an air-water mixture. A dominance in the cooling effect of the water can be expected to lead to quenching of the flame.

1.5 OBJECTIVES

The overall objective of the investigation are (i) the determination of the nature and magnitude of the effects on the performance and output of a gas turbine prediffuser-combustor, operated with an air-water mixture, and (ii) the establishment of the extent to which such effects may be modified with an increase in fuel equivalence ratio, and an enrichment of air with oxygen.

Specifically, there are two objectives in the investigation: (i) establish the amount of water entering the primary zone as a function of the entry conditions to the prediffuser; and (ii) examine the gain in gas phase temperature, the loss in pressure across the combustor, and flame stability including the occurrence of flameout as a function of mixture and operational conditions. An attempt is made to elucidate the balance among chemical heat release, heating and vaporization of water, and physical-chemical effects due to liquid water entities and vitiation with water vapor.

1.6 OUTLINE

In Section 2, a description of the two-phase flow tunnel and the test article is given. In Section 3, an outline of the investigation is provided along with details of the test plans and procedure. The results and observations of the cold flow and burning tests are given in Sections 4 and 5, respectively. An analysis and discussion of the results is provided in Section 6. Finally, conclusions are given in Section 7.

2. TEST FACILITY AND TEST ARTICLE

The test facility consists of the following: (1) A three-dimensional, two-phase flow tunnel; (2) an air supply system; (3) processing and injection systems for water in film and spray form; (4) fuel supply and processing systems; (5) an oxygen supply system; (6) facility instrumentation and control systems; and (7) data acquisition and processing systems. The overall test facility is given in schematic in figure 7, and items 1 through 7 are described in Sections 2.1 through 2.7 sequentially. The test article consists of a 60 degree sector of an annular gas turbine prediffuser-combustor unit, with three fuel injectors. A photograph of the test article is provided in figure 8. A description of the test article and the fuel injection system, the ignition system and the instrumentation follows in Section 2.8.

2.1 TWO-PHASE FLOW TUNNEL

The two-phase flow tunnel consists of the following components: (1) an air supply duct; (2) an inlet plenum chamber in which the incoming flow from the system settles; (3) a converging duct which connects the plenum chamber to the spray injection section; (4) a section in which water is injected in spray form and also honeycomb segments are included to increase flow uniformity; (5) a transition section which alters the annulus geometry to match that of the prediffuser; (6) a section in which a film of water can be injected along the top wall of the tunnel; and (7) the test section with provision for installation of various test models. A schematic representation of the two-phase flow tunnel is provided in figure 9 and 10, and photographs of the tunnel (with the test article installed) configured for cold flow tests and burning tests are provided in figures 11 and 12, respectively.

Incoming flow from the air supply enters the inlet plenum chamber and flows through a series of baffle plates. The plates serve to settle and diffuse the air flow from the air supply ducting before it is supplied to the test section. The spray injection section has been designed to incorporate a series of commercial spray nozzles. A series of honeycomb segments, which envelope the nozzles, has been installed in this section in order to reduce possible flow disturbances that may be caused by the presence of nozzles and to ensure that the air is supplied to the test section with a fairly uniform velocity distribution. The converging transition section serves to homogenize the spray distribution before entry to the film injection section.

Currently, the unnel is configured for a 60 degree sector of an annular combustor. The sector, as shown in figure 13, is circular in cross-section. Therefore the two-phase flow tunnel elements (iii) and (iv) in figure 9 are also circular in cross-section. A test article may be incorporated in the tunnel in several ways relative to the gravitational direction as shown in figure 14. It is possible in this manner to test a given sector in one of four relative positions in an annular combustor: position (1) and (4) vertical, in and in opposition to the gravitational direction, and position (2) and (3) inclined at an angle to the vertical. Since the shape of the flow annulus at the exit of the film injection section (part (vi) in figure 9) is matched to the shape of the prediffuser section of the test article, when the test article is turned to positions 2, 3 or 4, the elements (iv) through (vi) have to be turned as well. Provision has been made for such a rotation of the required part of the tunnel by means of a roller support mechanism (not included in figure 9) at the flanges between the connecting duct and the spray injection section, and between the test article and the exhaust duct. The photograph of the tunnel given in figure 12 shows the tunnel in position 2 of figure 14. Provisions have been made for drainage of excess water from the tunnel at the inlet plenum camber and the test article.

2.2 AIR SUPPLY SYSTEM

The air supply system utilizes high pressure air, which is stored in reservoir tanks at approximately 15.0 MPa. The air supply to the two-phase flow tunnel (at pressures of the order 7 to 41 kPa) is divided into the primary, and the (combustor) heated air subsystems. Each of those is regulated to the required pressure by means of a hand loader. A schematic of the air supply system is given in figure 15, and the associated nomenclature is given in figure 16.

The air can be supplied to the tunnel at desired value of temperature (up to about 95^{o} C) by mixing cold air with products of combustion from a methanol-fueled combustor. This method leads to some vitiation and depletion of oxygen in the air supplied to the test combustor. However, it has been found that combustion is feasible with cold air, although ignition requires heating of air and also the combustor itself. Meanwhile, it must be noted that operation under cold air flow conditions affects (a) the combustion efficiency, and (b) the amount of water, when injected, that can undergo heating and vaporization at different locations in the combustor.

A typical power gas turbine combustor-can is used to heat the incoming air to a temperature of the order of 55 C to 260 C. Figure 15 depicts the location of the air heater. The combustor burns methanol fuel (under lean conditions) which is supplied to the combustor from a large storage tank by utilizing a regulated nitrogen supply to pressurize the fuel system as required. The flow rate of methanol is controlled and measured by means of a needle valve-type rotameter of about 34 Lph capacity.

A standard airblast atomizer, mounted in the combustor wall near the dome of the can, is utilized to atomize the methanol fuel. The airblast is provided to the nozzle by regulating shop air, available at about 690 kPa, to a required pressure so that a high degree of atomization of the fuel is obtained for stable combustion.

The air-methanol mixture in the primary zone of the combustor is then ignited by providing a spark at a frequency of approximately 400 Hz. This is accomplished by means of an aviation spark plug connected to an electronic timing mechanism.

2.3 WATER PROCESSING AND INJECTION SYSTEMS

The water supply system has been designed to deliver filtered city water at the desired pressure and flow rate to both the film injection device and the spray nozzles. Figures 17 and 19 provide a schematic of the water supply and processing systems, respectively. City water (at about 690 kPa) is fed through a series of three Culligan water filters (5 micron mesh) to remove mineral deposits and rust. The filtered water flows into a reservoir tank from which it is pumped (26 Lpm, 620 kPa) to the twin control cabinets which supply the film injection device and the spray nozzles. A bypass loop with a relief valve (set at 690 kPa), which allows a desired amount of water to be bled off at the discharge of the pump, is used as a means of controlling the pressure of the water delivered to the control cabinets.

Each water regulation unit consists of two flowmeters of large (26 Lpm) and small (11 Lpm) capacities, corresponding 'coarse' and 'fine' adjustment valves for each flow meter, and a water pressure gauge. Filtered water at the desired pressure and flow rate can thus be delivered to either or both the film and spray injection systems.

2.3.1 Water Processing

Two heat exchangers have been mounted, in series, in the water supply system immediately upstream of the film and spray injection manifolds. The heat exchangers, which can be filled with a mixture of solid carbon dioxide, ice and liquid water, have been utilized to chill the water supply to a temperature of about 5^{o} to 7^{o} C. A strip heater has been mounted on each of the film and spray water supply lines downstream of the heat exchangers in order to adjust the temperature of the chilled water to a desired temperature. Thus the water is first chilled to a nearly constant low temperature and then heated to the desired value of temperature.

2.3.2 Water Injection Scheme

Water in the form of droplets of various volumetric mean diameters ranging from 20 μm can be injected into the test rig through a set of nozzles obtained from Spraying Systems, Inc. The characteristics of each of the selected nozzles, obtained form the manufacturer, are provided in Appendix A. An air-assist is required for obtaining small droplet diameters (<50 μm). In addition, large quantities of water, up to 18 per cent of the mass fraction of air flow, may need to be atomized. In such cases, in order to operate the selected nozzles, shop air at 690 kPa is regulated to obtain the air pressure specified by the manufacturer for the injector. Both types of nozzles chosen produce a flat type (elliptical) spray pattern and are thus mounted in the tunnel so that the spread of the spray is in the direction corresponding to the width of the tunnel. The nozzles also have relatively small spray angles so that the impact of spray on the side walls is small. Placement and orientation of the nozzles (relative to air flow direction) can be adjusted to provide a uniform distribution of spray at entry to the test article, and minimize the spread of the overall spray cone and impact of spray on tunnel walls. Figures 20 and 21 show the configuration of the spray nozzles mounted in the spray injector manifold. The nozzles are enveloped in honeycomb flow straighteners so that the spray enters nearly uniform flow. There are no further flow disturbing features up to the prediffuser-combustor test section.

Water can also be injected into the test rig in the form of a thin film. Provision has been made for film injection approximately 5 cm upstream of the test article. Film injection can be arranged either on the top or the bottom wall in order to establish the effects of gravity. Water, at the desired flow rate and pressure (slightly higher than the tunnel pressure), is delivered to a plexiglass reservoir where it is fed through a series of baffle plates. The water then flows into the tunnel through a 'knife edge' slit and enters the tunnel. A component of momentum in the air flow direction is imparted to the water as it flows out over the 'knife edge' and into the tunnel so that the film initially becomes attached to the top wall. The settling chamber ensures that the water is evenly supplied along the length of the slit to obtain steady and uniform injection. The knife edge has been machined so as to match the curved top surface of the tunnel exactly, and, at the same time, is suitably rounded in order to prevent the discharge water from appearing in the form of a jet. The thickness of the film admitted into the test section is controlled by adjusting the width of the slit. A photograph of the film injection device is provided in figure 22.

2.4 FUEL SUPPLY AND PROCESSING SYSTEMS

The fuel utilized for all the burning tests was JET-A. The physical properties of the fuel are previously summarized for convenience in Table 1. Details of the individual constituents and the reaction mechanisms which can be utilized in modelling the oxidation of JET-A may be found in reference 21.

2.4.1 Fuel Supply

JET-A fuel is supplied to the test article from a large storage tank by utilizing a regulated nitrogen tank to pressurize the fuel system to 550 kPa. The fuel is supplied to the three fuel injectors in the three injector cups of the combustor by means of individually regulated fuel lines. The fuel lines are connected to a single manifold that is supplied with pressurized fuel at a rate corresponding to the desired value of equivalence ratio. In the current series of tests, the fuel flow rate ranges from 7.0 to 15 ml/s and is controlled by means of a remotely actuated, 20-turn, needle valve. A paddle wheel type micro flow meter is utilized to determine the fuel flow rate to an accuracy of ± 0.5 per cent. A 5 μm particle size fuel filter has been installed in the fuel system upstream of the fuel control needle valve and the flow meter. A schematic of the fuel supply system is given in figure 23 and the associated nomenclature is given in figure 24. Details of the flow meter calibration and the individual fuel nozzle performance tests are given in Appendix B. A three-way, directional control, solenoid valve has been installed on each of the three fuel injector supply lines in close proximity to the test article. This allows the fuel supply to the three injectors to be bypassed into a dump tank during the warm up cycle, prior to an ignition attempt, for example, during the tests utilizing heated fuel.

2.4.2 Fuel Processing

A hot water heat exchanger has been installed in the fuel supply system, upstream of the fuel filter, to provide for burning tests carried out under conditions of heated fuel. The fuel can be heated up to a temperature of approximately 70° C. A thermostatically controlled immersion heater is utilized to heat the water in the fuel processor to a temperature less than 100°C. The fuel temperature is measured at the entrance to the fuel manifold by means of a type-k thermocouple. The water temperature in the processor is regulated by means of a PID temperature controller so as to give the desired fuel temperature at the fuel manifold. The heat exchanger, as well as the fuel lines downstream of the fuel processor, has been insulated in order to keep the drop in the fuel temperature between the processor and the manifold at a minimum.

2.5 OXYGEN SUPPLY SYSTEM

The oxygen supply system consists of a gaseous oxygen storage cylinder, pressurized to 13.8 MPa; a dual stage oxygen regulator; a two-way solenoid shutoff valve; and a choked flow nozzle. The flow rate of gaseous oxygen supplied to the air supply entering the two-phase flow tunnel, ranging between 7.3 and 9.1 g/s, is regulated by setting the oxygen pressure upstream of the choked flow nozzle to a predetermined value which corresponds to the desired oxygen mass flow rate through the choked flow nozzle. The upstream pressure is set by adjusting the dual stage regulator prior to a test. The solenoid actuated shutoff valve is then energized from the control room, adjacent to the test cell, whenever gaseous oxygen is required during a test. The oxygen is injected into the air flow immediately upstream of the inlet plenum chamber of the tunnel to ensure adequate mixing. A schematic of the oxygen supply system is given in figure 25.

2.6 FACILITY INSTRUMENTATION AND CONTROL SYSTEM

Instrumentation has been installed in the test facility in order to provide a means by which the state of the following can be determined: the air flowing through the air system (and the test rig), the water injected into the test rig, the aviation kerosene (JET-A) fuel supplied to the test combustor, and the oxygen additive supplied to the test rig. At the same time, provisions have

been made which enable each system to be controlled by utilizing the facility instrumentation in conjunction with various control devices to provide the desired conditions in the test article.

2.6.1 Air Supply

Measurements are required at several locations in the air system, as well as the test rig itself, in order to obtain the desired pressure, temperature and mass flow rate of air in the test article. Pressure gauges have been installed in the high pressure air supply, in the primary air system upstream of the mixing valves, and in the combustor air stream upstream of the combustor in order to establish the facility inlet conditions. Two calibrated orifice plates, obtained from Daniel Industries, one located in the primary air line upstream of the mixing valves, and one located in the heated air line upstream of the combustor, provide a means of determining the mass flow rate of air through each system. The differential pressure across each orifice place is measured with a pressure gauge located on the console. Type K thermocouples installed on the upstream side of each orifice are used to measure the temperatures of the inlet air streams which can be displayed on an Omega digicator.

Six type-k thermocouples have been installed on the outside wall of the prediffuser in order to provide a means of monitoring the temperature of the diffuser, particularly during a combustion test. The output from the thermocouples is displayed on the control panel. Thermocouples have also been installed in the air system at the heater exit and in the mixed-air line downstream of the mixing valves. The former is used as a safety feature in relation to proof-of-ignition and is continuously displayed and monitored. The output form the second thermocouple is displayed on the digicator and used in conjunction with the mixing valves as a means of determining and controlling the temperature of the mixed-air delivered to the test rig.

An ASME standard flow nozzle has been designed and incorporated into the air supply system upstream of the test rig in order to determine the actual mass flow rate of air entering the test rig. A Validyne pressure measurement system measures the upstream and the differential pressures and they are stored in the computer. The air temperature measured immediately upstream of the nozzle, is also stored in the computer and used for the calculation of the mass flow rate of air into the test rig.

2.6.2 Water Supply

The state of the water which is delivered to the film injector and the spray nozzles is determined by utilizing three pressure gauges, several flow meters (rotameters) of various capacities, and a type K thermocouple. Thermocouples just upstream of the film injector and the spray nozzles measure the water temperatures, that are stored in the computer for later reference.

Control of the water supply system is achieved in two stages. First, the pressure and the flow rate of water delivered to the twin injection control cabinets can be controlled by adjusting calibration valves, located in the calibration cabinet upstream of the four rotameters, and by opening bleed valves (which control the quantity of water to be bled off the mainstream back to the reservoir tank) until the desired pressure and flow rate is obtained. The second stage is based on adjusting coarse and fine needle valves in each injection control cabinet until the desired pressure and flow rate of water are delivered to the film injector and the spray nozzles.

2.6.3 Jet-A Fuel Supply

Aviation kerosene can be supplied to the test combustor at room temperature as well as at desired elevated temperatures (below 100°C). The fuel is pressurized to the required pressure by means of a regulated nitrogen supply, controlled with a hand loader on the control panel. The pressure of the fuel in the tank is measured and displayed on the control panel. In order to measure the temperature of the fuel, type-k thermocouples have been installed in the injector manifold and immediately upstream of the micro flowmeter. The temperature data are stored in the computer for later reference. The flow rat of the fuel is measured by means of a paddle wheel type flow meter. The output form the flow meter (pulses per sec) is converted to a corresponding calibrated flow rate and is displayed on a digital readout on the control panel. The signal is also stored on the computer.

The temperature of the water in the heat exchanger in the fuel processor is controlled by means of a thermostatically controlled immersion heater. The heat exchanger is operated only when heated fuel is required.

2.6.4 Oxygen Supply

The oxygen can be supplied to the test section at atmospheric temperature. The pressure of the oxygen supply is controlled by means of a pressure regulator. A solenoid-operated shutoff valve is utilized to control the flow of oxygen into the test rig. A choked flow nozzle has been installed in the oxygen supply line as a means of determining the mass flow rate of oxygen.

2.7 DATA ACQUISITION AND PROCESSING SYSTEMS

The data acquisition system consists of the following: a Zenith personal computer; a Computer Boards, Inc. CIO-DAS08-PGA analog-to-digital conversion board; a Validyne Data System; a Sony CCD (Charged-Coupled Device) V-99 video camera; and a Data Translation DT2851 high speed frame grabber. The Zenith computer utilizes a DOS operating system. Both the operating system and the experimental data have been stored on 3.5 inch floppy disks as well as on the hard drive of the computer. The analog-to-digital conversion board has been utilized to convert analog signals to the computer into binary form. The board has 8 channels with 12 bits of unipolar resolution. During a typical test, data are acquired at a rate of approximately 1 Hz. The Validyne Data System, which consists of a MC1-10 Multi-Channel Modular Transducer Main Frame and 7 CD-19 High Gain Carrier Demodulator Modules, has been used with 7 DP-15 transducers to acquire measurements pressure throughout the test facility. In particular, the pressure drop across the flow nozzle as well as the static pressure at the inlet, the test combustor inlet and exit stagnation pressures, the static pressure at the inlet plenum chamber, and the stagnation pressure in the spray injection section. The system also provide a means for acquiring weight measurements, with an Omega LCL-020 (9 kg, max) load cell, of water collected in the test article.

Photographic data have been obtained by means of a Sony CCD V-99 video camera. The data have been recorded on 8 mm video tapes. During processing of the data, the video tapes have been played back by means of a video recorder (in the camera), and the photographic data have been sent to a Data Translation DT2851 high speed frame grabber which captures any selected image. A software routine called IRIS has been used to control the DT2851 frame grabber, to process images, and to store the processed images on high density diskettes.

2.8 TEST ARTICLE

2.8.1 Prediffuser Combustor Sector

The test article is a 60-degree sector of an annular combustor from a typical high bypass ratio turbofan engine. It consists of a prediffuser and a combustor. The prediffuser is a dump-type prediffuser and has no compressor exit guide vanes. A photograph of the prediffuser is given in figure 26. The combustor has three swirl cups and three fuel injectors equipped with pressure swirl atomizers.

The combustor sector model is designed and built such that the cooling stream of air (in the outer and inner jacket of the combustor) enters the main chamber through the primary and secondary jets in total, in other words, there is no separate outlet for the combustor cooling air (figure 14). The combustor can be operated with corrected air mass flow rates corresponding to chosen engine operating conditions, such as flight-idle or ground-idle. The corrected air mass flow rates corresponding to the flight-idle and ground-idle operating conditions for the test article have been chosen as 0.45 kg/s and 0.36 kg/s, respectively. Under these operating conditions, the ratio of the flow split between the main and two cooling streams is approximately 29:33:38, and the pressure loss across the diffuser-combustor element ($\Delta P/P$) is between 5.0 and 6.0 per cent.

The combustor is mounted in a stainless steel test section equipped with viewing ports on either side and on top of the combustor liner for flow visualization studies. During the cold flow tests, plexiglass windows have been mounted in the viewing ports and during the combustion tests, the plexiglass windows have been replaced with quartz windows. The test section is equipped with water collection attachments, mounted at the lowest points in the inner liner, so that the flow rate of water through the main and secondary streams can be monitored. The position of the water collection attachments is illustrated in figure 13. The test article can be tested in one of four relative positions in an annular combustor: position 1 and 4 vertical, in and in opposition to the gravitational direction, and position 2 and 3 inclined at an angle to the vertical, as shown in figure 14.

2.8.2 Fuel Injection System

Fuel, pressurized to approximately 500 kPa, is supplied to the fuel injectors by means of three individually regulated fuel lines. Three Delavan pressure swirl atomizes (Model No. 27710-1), with spray angles of approximately 75 degrees have been utilized in the injectors. The combustor is equipped with three counter-rotational swirl-type injector cups. Swirl is induced in the flow through the cups by means of two sets of openings; one which is circular and induces rotation in a clockwise direction and the other which is a vane-type configuration and induces rotation in a counter-clockwise direction.

2.8.3 Ignition System

The sector is equipped with a standard aircraft gas turbine ignitor plug (Unison Industries 1374 M12 P01) and ignitor lead (Unison Industries 9339 M26 P09). A Unison Industries 9238 M66 P07 exciter has been utilized as the spark source. It is powered by a 115 V, 400 Hz aviation power supply and outputs a 15-20 kV pulse voltage at approximately 1.0 Hz.

2.8.4 Instrumentation

The tempe.ature rise and the pressure loss across the combustor have been measured using shielded type-k thermocouples, and pilot probes mounted in the prediffuser and in the combustor exit flange, respectively. The occurrence of an ignition or a flameout is established by the combined use of thermocouples for establishing the noticeable, large increase or reduction in exhaust gas temperature and also, a CD cell that is exposed to the primary zone of the combustor through the transparent quartz windows for detecting both a bright and a quenched flame.

3. OUTLINE OF EXPERIMENTAL STUDIES: PLANS AND PROCEDURES.

Details of the plans and procedures for the experimental investigation are provided in this chapter. The experimental investigation consists of two parts, first tests carried out under cold flow conditions, and second combustion tests. The cold flow tests have been devoted to determining the distribution of water in the combustor when an air-water mixture is supplied to it. Combustion tests have been carried out to establish the effects on performance, recoverability of combustor exit temperature, and flameout under various flow conditions, including the addition of oxygen to the air. A description of the test plans is given in section 3.1, and details of the test procedures for the cold flow and burning tests are provided in section 3.2.

3.1 TEST PLANS

3.1.1 Cold Flow Tests

In order to estimate the influence of water droplets on combustion, it is necessary to have a knowledge of the total quantity of water entering the primary zone of the test article, as well as the number density and droplet size distribution, as a function of the properties of the air-water mixture at the inlet of the prediffuser. In the cold flow tests, the principal parameters of interest are the amount of water flowing through the primary zone on account of entry through the dome and the primary jets, and the size distribution of water droplets and ligaments in the primary zone. The tests have been conducted with water in the form of a film, a spray, or a combination of film and spray. The tests have been carried out with both large and small droplet sizes and with the test article oriented in positions 1 and 2 of figure 14. The range of test parameters for the cold flow tests is given in table 2.

TABLE 2. RANGE OF COLD FLOW TEST PARAMETERS

1	Air mass flow rate ($\dot{m}_a$)	0.45-0 and 82 kg/s
2	Mass fraction of injected film (X_{wf})	0-2 per cent
3	Mass fraction of injected spray (X_{wd})	0-8 per cent
4	Nominal spray drop size (D_w)	25 and 750 μm
5	Combustor orientation (fig 13)	1 and 2

3.1.2 Burning Tests

The burning test plan consists of five parts: (A) Combustion at fixed equivalence ratio under conditions of operation with air only (denoted as A-A) and operation with air-water mixture (denoted as A-W), for establishing performance changes, including flameout; (B) combustion with various values of equivalence ratio in order to determine the possible recovery of exhaust temperature under conditions of water ingestion; (C) combustion utilizing air enriched with oxygen during operation with air only (denoted as C-A) and operation with air-water mixture (denoted as C-W), in order to determine the effects of addition of oxygen on combustion performance; (D) combustion with increased fuel equivalence ratio, and air enriched with oxygen in order to determine the possible recovery of exhaust temperature under conditions of water ingestion; and (E) ad hoc diagnostic tests with heating of air, etc. The range of test parameters utilized in the burning tests (A), (B), (C), (D), and (E) is given in table 3. The tests have been

carried out with both large and small droplet sizes and with the test article oriented in position 1 and 2 of figure 14.

TABLE 3. RANGE OF BURNING TEST PARAMETERS

1	Air mass flow rate ($\dot{m}_a$)	0.36 and 0.45 kg/s
2	Mass fraction of injected film (X_{wf})	0 and 2 per cent
3	Mass fraction of injected spray (X_{wd})	0 and 20 per cent
4	Nominal spray drop size (D_w)	25 and 750 μm
5	Fuel equivalence ratio (ϕ)	0.24 and 0.4
6	Injected water temperature (T_w)	7^o and 20^o C
7	Fuel temperature (T_f)	20^o and 40^o C
8	Mass fraction of oxygen additive (X_{O2})	0 and 2 per cent
9	Inlet air temperature (T_{air})	10^o and 40^o C
10	Combustor orientation (figure 13)	1 and 2

3.1.3 Test Conditions

All of the tests have been conducted, as stated in section 1.3, under conditions corresponding to flight idle and ground idle operation. The combustor operating parameters under those conditions are listed in table 4.

TABLE 4. COMBUSTOR OPERATING PARAMETERS

Operating Condition	Corrected air Mass Flow Rate (kg.s)	ϕ
Flight-Idle	0.454	0.3
Ground-Idle	0.363	0.24

In the foregoing, concerning the group (B) tests, it was mentioned earlier that a maximum limit was imposed on the fuel equivalence ratio utilized. The actual limit chosen for the fuel equivalence ratio ϕ was 0.4, based on limiting the temperature that the combustor wall and other parts of the test set-up could attain.

It may be pointed out that each of the groups of combustor tests has been conducted with the mass flow rate of air through the test article maintained constant, rather than the static pressure drop between the entry and the combustor internal section. In the current test article, it is not possible to vary the mass flux and the pressure drop independently of each other because of the fact that the secondary air flow does not have controllable exits. In other words, the secondary air stream exits are closed and all of the air discharges through the exit of the main stream.

3.2 TEST PROCEDURE

3.2.1 Cold Flow Tests

In the cold flow tests, the amount of water flowing through the primary zone has been established as the difference between the total amount of water admitted per unit time with air at entry to the combustor and the amount of water that could be collected per unit time by means of drains located in the inner wall of the combustor test section. During a test, the collected water is continuously drained into a jar which is suspended beneath the test rig on a load cell by means of a cable. The pressure in the jar is equalized with the local pressure in the test article near the drainage tubes by means of a breather tube. This allows the water to drain freely under gravitational action. The load cell provides a continuous signal to the computer by means of the Validyne carrier demodulator module (as described in Chapter 2). A computer routine has been utilized to sample the signal at a frequency of approximately 0.2 Hz and convert to a calibrated weight measurement of the jar and water as a function of time. The discretized time history can then be converted into a time history of the mass fraction of water as a per cent value of that which is injected into the airstream at entry to the prediffuser. The tests have been conducted with water addition in three ways as follows: film only; spray only; and the film content held a mass fraction 2.0 per cent and the spray content varied over a mass fraction range of zero to 8.0 per cent.

The overall scheme for determining the water distribution in the combustor is as follows: the combustor secondary holes are blocked off; the combustor is supplied with cold air-water mixture at chosen values of air pressure, temperature, and mass flux, and water form and mass fraction; and the amount of water entering the primary zone, through the dome section and the primary jets, is established as the difference between the amount of water at entry to the combustor and the amount of water that flows in the outer stream. In conducting the tests, the following have been noted:

(i) Each set of values of mass flow of air, pressure, and temperature represent a value of corrected mass flow, that corresponds to an engine operating condition such as ground-idle and flight-idle. The available ranges of air supply and test parameters in the investigation generally yield flowfields in the engine idle to flight idle operation ranges.

(ii) The combustor sector model has been designed and build such that the cooling stream of air (in the outer jacket of the combustor) enters the main chamber through the primary and secondary jets in total, in other words, there is no separate outlet for the combustor cooling air (figure 14).

(iii) A combustor operation is specified by the mass flow rate of air, the changes in stagnation pressure ($\Delta P/P_2$, say), and the air-fuel ratio. With known values of P_2 and T_2, the flow velocity in the combustor then becomes fixed. Also, a given combustor yields a unique flow split for given entry conditions and $\Delta P/P_2$. In view of the design of the test combustor sector, as described in (ii) above, there is no provision for obtaining in a test a desired combination of independent values of $\Delta P/P_2$ and mass flux. If a value of $\Delta P/P_2$ is chosen, then the mass flux becomes fixed, since the combustor is open to the atmosphere.

(iv) As stated earlier in the description of the overall scheme for determining the amount of water entering the primary zone, the secondary holes needed to be blocked off during the

tests. In the actual combustor operation with combustion, both the primary and the secondary jets must remain open to ensure necessary cooling. Thus the values of $\Delta P/P_2$ that are of interest are the values obtained with both the primary and the secondary jets open, with various air-water mixtures, and under burning conditions.

(v) One may consider obtaining the value of water entering the primary zone by establishing the difference in the amounts of water under conditions of the combustor operating with all holes open and with the dome and primary holes closed. Unfortunately such a test will produce an erroneous value for the amount of water entering through the secondary holes. Finally, one may consider direct collection of water entering through the dome and the primary jets by means of individual ducting. Such a scheme, however, introduces serious interferences in the flowpath.

In view of the foregoing considerations, the cold flow tests have been carried out as follows:

1. The amounts of water entering the primary zone have been established for several chosen air flow rates and with various air-water mixtures, under the conditions with the secondary holes closed, and with all the holes open. It has been realized that these two sets of tests would yield two different bodies of data, that together could yield an overall understanding of the distribution of water in various cases.

2. Video photography of the flowfield has been conducted under various conditions of air-water mixture.

Flow visualization has been carried out using video-photography with forward scattering of incandescent light being utilized as the illumination technique. A Sony CCD V-99 video camera with shutter speeds ranging from 0.1 to 4.0 msec and with 400 lines of horizontal resolution has been utilized. The camera has a wide angle lens with an adjustable focal length ranging from 11 mm to 2.5 meters. A macro lens has been used for focusing at distances between 11 and 88 mm. In each case, the flowfield in the primary zone of the combustor has been photographed (with the shutter speed set between 0.1 and 4.0 msec) in a plane that is parallel to the direction of flow. In order to determine the range of droplet sizes in the primary zone, photographic data that have been obtained during the flow visualization and stored on a video tape have been sent to a Data Translation DT2851 frame grabber during playback. Selected frames have been captured and stored on high density floppy disks for later processing. The diameter of the droplets in the images has been measured manually with the aid of a measuring grid provided by the software package (IRIS). The scaling factor that is required in estimating the actual size of the droplets has been obtained by comparing a known dimension on the test article to the corresponding dimension in the image. This scaling factor has been used to determine the actual size of the droplets. During video-imaging, the camera is focused on the X-Y plane coincident with the center cup. However, since the whole flowfield is illuminated, droplets in planes on either side of the focused plane also appear in the image. To overcome this problem, a representative set of 4 images for each test has been processed and the largest and smallest diameters obtained have been taken to be representative of the limits of the droplet sizes in the focal plane of the camera.

3.2.2 Burning Tests

The procedures for the five groups of burning tests A, B, C, D, and E may be summarized as follows:

Group A: (i) With chosen values of $\dot{m}_a$, T_{air}, T_f, and ϕ, the combustor is lighted up. Performance data are acquired on reaching steady state conditions. (ii) Water is admitted at the desired mass fraction and in the desired form. Performance data are acquired once the combustor exit temperature reaches a steady state value. (iii) The mass fraction of admitted water is increased to a chosen higher value, and performance data are acquired once a steady state condition is reached. (iv) Step (iii) is repeated until a flameout occurs. This group of tests has been conducted with the combustor in settings 1 and 2 (figure 14). It should be pointed out that for the tests carried out in setting 2, the mass fraction of water in spray form has been increased to a maximum value of 4 per cent and a flameout has not been observed.

Group B: The test procedure is as follows: (i) and (ii) The same as (i) and (ii) in Group A. (iii) The fuel flow rate is then gradually increased until either the combustor exit temperature increases to the steady state values observed in step (i), or the equivalence ratio, ϕ, reaches the chosen limiting value of 0.4. The combustor performance data are then acquired. Then ϕ is reduced to the steady state value in step (i), and the water is turned off. (iv) Once the combustor exit temperature reaches the same steady state value that had been observed in step (i), water is admitted at a mass fraction value larger than in the previous step, and performance data are acquired once a steady state condition is reached. (v) Steps (iii) to (iv) are repeated until a flameout occurs.

Group C: These tests consist of the performance tests with oxygen addition. The oxygen has been injected into the air supply immediately upstream of the inlet plenum chamber of the tunnel. The maximum mass fraction of injected water has been limited to 12 per cent. The procedure for the performance tests is as follows: (i) With chosen values of $\dot{m}_a$, T_{air}, T_f, and ϕ, the combustor is lighted up. Performance data are acquired on reaching steady state conditions. (ii) 2.0 per cent oxygen is then added to the air supply and performance data are recorded once a stable combustor exit temperature is obtained. (iii) The oxygen is then turned off, and once the combustor exit temperature has stabilized, water is admitted at the desired mass fraction and in the desired form. Performance data are acquired once the combustor exit temperature has reached a steady state value. (iv) 2.0 per cent oxygen is then added to the air supply and performance data are acquired. (v) Step (iii) is repeated and the mas fraction of admitted water is increased to a chosen higher value, and performance data are acquired once a steady state condition is reached. (vi) The test is concluded once performance data have been obtained for water mass fractions up to 12 per cent.

Group D: The procedure of the recovery test with oxygen addition is as follows: (i), (ii) and (iii) The same as (i), (ii) and (iii) in Group C performance test. (iv) 2.0 per cent oxygen is added to the air supply and then the fuel flow rate is gradually increased until either the combustor exit temperature has increased to the steady state values observed in step (i), or the equivalence ratio, ϕ, has reached the chosen limiting value of 0.4. The combustor performance data are acquired. Then ϕ is reduced to the steady state value in step (i), and the oxygen and the water are turned off. (v) Once the combustor exit temperature has reached the same steady state value that had been observed in step (i), water is admitted at a mass fraction value larger than in the previous step, and performance data are acquired once a steady state condition has been reached. (v) Steps (iv) to (v) are repeated until the mass fraction of injected spray has reached 8.0 per cent.

Group E: The test procedure in this case is identical to that adopted in case A, except that the inlet air is heated and, in some cases, the fuel is also preheated before being injected into the combustor.

In each test, the temperature rise and the pressure loss across the combustor have been measured using standard probes. The occurrence of flameout has been established by the combined use of thermocouples for establishing the noticeable, large reduction in exhaust gas temperature and also, a CD cell that is exposed to the primary zone of the combustor through the transparent quartz windows for detecting a quenched flame.

It should be noted that in curved combustor sector, the flow of water into the primary zone through the injector cups and the primary jets is affected by gravity. The center cup and the primary zone in front of it can be expected to receive a different amount of water, and also, in all likelihood, a different amount of air compared with the side cups and the primary zone sections in front of them. The water distribution becomes further affected when the combustor is operated in the different angular settings (shown in figure 14). In the combustion experiments performed with the combustor in setting 2 (figure 14), the cup displaced the most in the gravitational direction can be expected to be the most readily and the greatest affected for a given set of operational and water injection conditions, while the cup that is displaced the least with respect to the gravitational direction may be operating simultaneously with little water and hence with little change in combustion characteristics. In this regard, one can also expect substantial differences in the performance of different cases for the cases of injection of film, spray, and combinations of film and spray.

The current tests, conducted with a specific prediffuser-combustor, yield results that are in several respects test article geometry-dependent. Thus, the results must be considered from the point of view of the trends realized with respect to various processes and aspects of phenomenology.

4. COLD FLOW TESTS: RESULTS AND OBSERVATIONS.

The results and observations obtained during the cold flow series of tests are provided in this chapter. Experimental results of the redistribution of water in the test article, along with observations obtained during the tests are provided in Section 4.2. Observations that have been made during flow visualization are presented in Section 4.3 and results of the droplet size distribution in the primary zone are given in Section 4.3.

4.1 COLD FLOW TEST SERIES

The objectives of the cold flow series of tests are as follows: (1) to conduct 'water collection' tests in an attempt to establish the quantity of water entering the primary zone of the test article; (2) by means of flow visualization carried out during selected tests in (1), to determine the physical characteristics of the water and the manner in which the water enters the primary zone; and (3) by analyzing the video data obtained in (2), to obtain an estimate of the droplet size distribution in the primary zone for various air-water mixtures at the entry of the prediffuser and obtain an approximate value of the fraction of water in the primary zone that would undergo 'flash evaporation'. The tests have been conducted with water addition in three ways as follows: film only; spray only; and the film content held at a mass fraction of 2.0 per cent and the spray content varied over a mass fraction range of zero to 8.0 per cent. The table of cold flow test cases is given in Table 5.

TABLE 5. COLD FLOW TEST CASE CONDITIONS

Case	Test Conditions					
	Setting (figure 13)	$\dot{m}_a$ kg/s	X_{wf}	X_{wf}	D_w μm	Sec. Holes
I	1	0.45; 0.82	0; 2	0; 2; 4; 6; 8	25	closed
II	1	0.45; 0.82	0; 2	0; 2; 4; 6; 8	750	closed
III	2	0.45; 0.82	0; 2	0; 2; 4; 6; 8	25	closed
IV	2	0.45; 0.82	0; 2	0; 2; 4; 6; 8	750	closed
V	1	0.45; 0.82	0; 2	0; 2; 4; 6; 8	25	open
VI	1	0.45; 0.82	0; 2	0; 2; 4; 6; 8	750	open
VII	2	0.45; 0.82	0; 2	0; 2; 4; 6; 8	25	open
VIII	2	0.45; 0.82	0; 2	0; 2; 4; 6; 8	750	open

4.2 WATER ENTRY INTO THE PRIMARY ZONE

This series of tests consists of water collection tests carried out on the test article with (i) the secondary holes closed in order to determine the fraction of injected water entering the primary zone, and (ii) with all of the dilution holes open in order to determine the fraction of injected water that enters the combustor. By processing the data obtained in (i) an estimation of the mass fraction of water in the primary zone as a function of the air-water mixture inlet conditions is also established.

4.2.1 Fraction of Injected Water Entering the Primary Zone

The amount of water flowing through the primary zone per unit time has been established as the difference between the total amount of water admitted per unit time with air at entry to the combustor and the amount of water that could be collected per unit time by means of drains located in the inner wall of the combustor casing. For each test data point, the duration of the test is three minutes, with data being acquired at five-second intervals. The value of the test data point shown in the figures is the arithmetic mean of the values obtained during the three minute test. The error bars correspond to the standard deviation of the readings obtained at each test point. The results of this part of the cold flow tests are given in figures 27 to 50. In those figures the test data points have been joined by continuous lines only for visual purposes.

Setting 1: With reference to figures 27 to 30 the following observations can be made: The amounts of water entering the primary zone have been found to be larger than the amounts in the two-dimensional tests reported in figures 2 to 5 of reference 7. For the case of film injection only, . appears that nearly all of the water enters the primary zone, partly through the dome, and substantially through the primary jets. This observation was confirmed during the flow visualization series of tests. In the case of the two-dimensional tests, this trend was also observed, however, the fraction of film entering the primary zone was slightly less than in the three-dimensional case. For a constant mass fraction of film injection, with no spray injection, an increase in the air speed results in larger quantities of water entering the primary zone. This observation is consistent with those reported in references 6 and 7. The amount of water entering the primary zone increases with an increase in air mass flux when water is present simultaneously in film and spray form. This observation appears to be independent of the mean volumetric drop size of the spray. When the air mass flow rate is small and the spray content is large, then the spray appears to shield the dome form impact of the film on the dome, and thus reduces entry of water into the primary zone. This observation also seems to be independent of the mean volumetric drop size of the spray. When there is only a spray of water (either large or small droplets), the amount of water entering the primary zone appears to be less than the amount entering when a film of water is present simultaneously; thus indicating that a film has a greater tendency to enter the primary zone. A similar observation was made in the tests carried out on the two-dimensional models, however, the increase in the total flow of water into the primary zone ranged form 5 to 15 per cent, which is slightly larger than that in the three-dimensional case.

Setting 2: With reference to figures 31 to 34, it can be noted that the observations noted above apply in general both when the combustor is in setting 1 as well as in setting 2. The most significant observation obtained during the tests in setting 2 is the large effect that gravity has on the motion of the film in the prediffuser, causing the film to flow sideways in the direction of the swirl cup displaced in the gravitational direction. This was not observed in tests carried out on the two-dimensional models [references 6 and 7], since there was no curvature in cross section. The effect of gravity in the two-dimensional model tests was, however, observed to influence the point of separation of the film form the top wall of the diffuser.

4.2.2 Fraction of Water Entering the Combustor

This series of tests has been carried out with the secondary holes of the combustor open. The results obtained for this series of tests are given in figures 35 to 42. For the cases with the secondary holes open, the fraction of injected water that enters the combustor is slightly larger, as expected, than that in those cases where the secondary holes are closed. Also, an increase in

the air flow rate results in a (negligibly) small change in the fraction of water entering the combustor, which was not the case in the results of the tests described in section 4.2.

4.2.3 Mass Fraction of Water in the Primary Zone

The results for these tests are given in figures 43 to 50. With reference to these figures, it can be observed that there is a linear relationship between the mass fraction of water entering the pre-diffuser and the mass fraction of water entering the primary zone. With an increase in the air mass flow rate, there is a corresponding proportional increase in the mass fraction of water entering the primary zone. For the case of large droplet injection at mass fractions higher than 4 per cent, a larger quantity of water seems to enter the primary zone when the combustor sector is tilted than when the sector is located symmetrically with respect to gravity, indicating that gravity distributes the droplets unfavorably with respect to the cups when the sector is tilted (i.e. setting 2) and favorably when the sector is located symmetrically with respect to the gravitational direction.

4.3 FLOW VISUALIZATION

Observations made during tests conducted with film injection, indicate that most of the water enters the primary zone through the primary jets, with a small amount of water exiting from the injector cups. At low air flow rates (that is, 0.45 kg/s or less), large globules of liquid can be seen exiting the swirl cups. At higher air flow rates, the globules tend to break up and the flowfield shows water in spray form. In the flow visualization tests carried out on the two-dimensional models [references 6 and 7], similar features were observed. It is established in the current tests that both the entry and dispersion of water (in film form) are strong functions of air flow velocity. At low air flow rates, a small amount of water exits through the cups, however, at higher air flow rates, the amount of water entering the primary zone through the cups increases to such an extent that visibility became extremely limited.

In the case of spray injection, it has been observed that a significant quantity of water enters the primary zone through the swirl cups, predominantly in the form of a swirling spray cloud, although at large flow rates some of the water seems to emerge in the form of slow-moving ligaments. For mass fractions of injected spray larger than 6.0 per cent, the amount of water that flows through the cups becomes so large the visual observation and video photography become extremely difficult. At higher air flow rates, there is an increase in the droplet velocities and a decrease in the droplet size. Similar observations were made during tests carried out on Models II and III in references 6 and 7, which had similar geometries and swirl cups as the three-dimensional test article. However, in the case of Model I, the water flowing through the injector cups was in the form of a jet for all conditions tested. This occurred as a result of the combined effect of flow through the cups being without swirl, and the large quantity of flow through them as set by the flow split.

Observations of the flowfield during tests conducted with simultaneous injection of film and spray have generally been difficult in the primary zone. At high values of air flow rate (more than 0.45 kg/s), no details can be identified due to the large quantity of water in the primary zone, and the apparently small drop sizes in the misty flow. At air flow rates of 0.45 kg/s or less, the flowfield is similar to that which arises in the case of spray injection (only) under identical conditions, except that the quantity of water entering the primary zone through the primary jets is substantially larger as a result of the additional water (in film form) which enters the primary zone predominantly through the primary jets. In the two-dimensional tests, similar observations were made, however, the increased flow of water through the primary jets was not observed.

In all cases of water injection, the water entering through the primary jets is predominantly in large droplet or ligament form. Following impact of jets in the center of the combustor, the water enters the air recirculation zones towards the dome. During injection of large quantities of water, there may arise severe flooding in the vicinity of the dome. In tests carried out on the two-dimensional models, no distinction could be made between the different modes of water injection.

Considering the influence of droplet size on the redistribution of water in the combustor, the small droplets are observed to enter the primary zone through the injector cups and the primary jets roughly in proportion to the flow split. However, in the case of large droplets, although the total quantity of water entering the primary zone does not seem to differ from the value in the case of small droplets, there is a greater quantity of water entering through the primary jets. In particular, a large amount of water appears to enter through the 'lower' primary holes, perhaps assisted by increased flow into the inner stream due to gravity.

During water ingestion into an engine, the prediffuser can be expected to operate, under certain circumstances, with droplet-laden air flow in the core of the flow and a flow of water over the walls. At low air flow rates, it has been found in flow visualization that gravity can have a noticeable effect on the film motion, causing the film to flow sideways towards the ends of the combustor sector, i.e., from the 12 o'clock position towards the 2 and 10 o'clock positions over the sector. It has been observed visually in tests conducted with the combustor sector in setting 2 of figure 14, that the film tends to flow towards the cup displaced in the gravitational direction, causing the other two cups to be partially starved of the film. A pictorial representation of the motion is presented in figure 51. It can be surmised that even in setting 1 of figure 14, the outer two cups probably receive a slightly greater amount of film than the center (12 o'clock position) cup.

4.3.1 Droplet Size Distribution

The distribution of the droplets has been determined based on the following assumptions: (1) the observed cross-sectional dimension of the measured droplets is taken to be equal to the diameters of the equivalent spherical droplets. (2) The droplet distribution in all of the X-Y planes throughout the primary zone is identical to that obtained in the observed plane. (3) The distribution of sizes between the largest and smallest droplets is of a Modified Rosin-Rammler form [reference 22]. An example of a typical video photograph of the droplet distribution in the primary zone is given in figure 52.

The number density distribution in the water flux has been determined by utilizing assumptions 2, 3, mentioned above, and the measurement of the total flux entering the primary zone, obtained by the methods described in section 4.2.

The following two assumptions have been made in order to fit a Modified Rosin-Rammler distribution to the droplets in the primary zone: 99 per cent of the water droplets is assumed to have a diameter less than the diameter of the largest droplet, and 5 per cent of the water droplets is assumed to have diameters less than the diameter of the smallest measured droplet.

In references 6 and 7, a maximum water droplet diameter of 1,000 microns was established to be the limiting diameter for which flash evaporation would occur in the primary zone. Utilizing the same criterion for droplet evaporation, it has been bound for the droplet distribution in the

combustor for the three test cases considered (Table 6), that 22, 21 and 20 per cent of the water entering the primary zone could be expected to evaporate, in cases 1, 2 and 3, respectively.

TABLE 6. CASES FOR ESTIMATION OF EFFECTS OF VITIATION ON COMBUSTION

Case Number	$\dot{m}_a$ (lb/sec)	X_{wf} (per cent)	X_{ws} (per cent)	D_w (microns)
1	1	2	2	750
2	1	1	4	750
3	1	2	8	750

The possible effect of vitiation due to flash evaporation of the water is discussed later in section 6.2.

5. BURNING TESTS: RESULTS AND OBSERVATIONS.

The results and observations obtained during the burning tests are presented in this chapter. The burning test series is outlined in section 5.1. A description of the performance parameters that have been utilized in the analysis of the results is provided in section 5.2. Section 5.3 contains the results of the tests carried out in order to establish the pressure loss across the combustor sector. The results of the combustor performance, and the occurrence of flameout under air-water mixture conditions are provided in Section 5.4. In section 5.5, the results of the tests carried out with increased fuel equivalence ratios, in an attempt to increase the combustor exit temperature to a value obtained under conditions of operation with air only, are presented. The results of the tests carried out in an attempt to establish the effect of water temperature and combustor orientation on the combustor exit temperature are provided in sections 5.6 and 5.7, respectively. The results and observations for the burning tests with oxygen addition are presented in section 5.9. Finally, section 5.10 is devoted to the results obtained with a combined increase in fuel equivalence ratio and air enriched with oxygen.

5.1 BURNING TEST SERIES.

The overall objective of the burning test series is to establish the effects on performance, recoverability of combustor exit temperature, and flameout under various flow conditions, including the simultaneous addition of fuel and oxygen to the air. The burning tests consist of five parts, namely, (A), (B), (C), (D), and (E), as described in section 3.1. The range of test parameters utilized for the various burning tests and the corresponding case numbers are given in tables 7 and 8.

Table 7 contains the test cases for parts (A), (B), and (E) of the burning test series. The combustor performance (and flameout) test cases are listed as cases 1 and 2. Case 1 corresponds to conditions of small droplet injection, and case 2 corresponds to large droplet injection. The combustor exit temperature recovery tests are listed as cases 3 and 4, with case 3 corresponding to small droplet spray injection, and case 4 corresponding to large droplet spray injection. Case 5 pertains to the tests carried out with chilled water and small droplet spray injection. The tests carried out with the test article oriented in setting 2 of figure 14 are listed as case 6. The ad hoc tests carried out with fuel at room temperature and ambient air temperatures, and with heated fuel and heated air, are listed as cases 7 and 8, respectively.

The burning test cases with oxygen addition, parts (C) and (D), are presented in table 8. Small droplet spray injection has been utilized for all of the tests carried out with oxygen addition to the air. Case 1 consists of the combustor performance tests carried out with air-water mixture and enriched air conditions. Case 2 pertains to the tests carried out with a simultaneous increase in the fuel equivalence ratio and the oxygen content in the air.

Before the results of the burning tests are presented, a brief discussion regarding the principal performance parameters employed in analyzing the results is provided.

TABLE 7. BURNING TEST CASES

Case	Setting (figure 13)	$\dot{m}_a$ kg/s	ϕ	Xwf	Xwd	Tw C	Tf C	Dw µm	XO2	Tair C
1	1	0.36	0.24/0.3	0/2	0-f/o	20	40	25	0	15
	1	0.45	0.24/0.3	0/2	0-f/o	20	40	25	0	15
2	1	0.36	0.24/0.3	0/2	0-f/o	20	40	750	0	15
3	1	0.36	0.24-0.4	0/2	0-f/o	20	40	25	0	15
	1	0.45	0.24-0.4	0/2	0-f/o	20	40	25	0	15
4	1	0.36	0.24-0.4	0/2	0-f/o	20	40	750	0	15
5	1	0.36	0.24/0.3	0/2	0-f/o	7	40	25	0	15
	1	0.45	0.24/0.3	0/2	0-f/o	7	40	25	0	15
6	2	0.36	0.24	0/2	0-4	20	40	25	0	15
	2	0.45	0.3	0/2	0-4	20	40	25	0	15
7	1	0.45	0.24	0/2	0-f/o	20	20	25	0	40
	1	0.45	0.24	0/2	0-f/o	20	20	25	0	15
8	1	0.36	0.3	0/2	0-f/o	20	40	25	0	40
	1	0.45	0.24	0/2	0-f/o	20	40	25	0	40

TABLE 8. BURNING TEST CASES WITH OXYGEN ADDITION

Case	Setting (figure 13)	$\dot{m}_a$ kg/s	ϕ	Xwf	Xwd	Tw C	Tf C	Dw µm	XO2	Tair C
1	1	0.36	0.24	0	2	20	40	25	1	15
	1	0.36	0.24/0.3	0/2	0-12	20	40	25	2	15
	1	0.45	0.3	0/2	0-8	20	40	25	2	15
2	1	0.36	0.24-0.4	0/2	0-8	20	40	25	2	15
	1	0.45	0.3-0.4	0/2	0-6	20	40	25	2	15

5.2. PERFORMANCE PARAMETERS.

The principal performance parameters employed in analyzing the results are as follows:

Pressure Loss Factor: The total pressure loss across the diffuser-combustor unit is a significant performance parameter. A relative pressure loss factor may be defined as follows for constant values of P_O, and T_O for the air supplied.

$$F_{PL} = \frac{\left(\frac{\Delta P_0}{P_0}\right)_{a+w} - \left(\frac{\Delta P_0}{P_0}\right)_a}{\left(\frac{\Delta P_0}{P_0}\right)_a} \qquad (1)$$

where

$$\frac{\Delta P_0}{P_0} = \frac{\left(P_{0ref} - P_{04}\right)}{P_{0ref}}, \tag{2}$$

P_{oref} being a reference total pressure upstream of the prediffuser and P_{o4} being the combustor exit total pressure.

Concerning the pressure loss factor, it must be noted that the pressure loss depends upon both the temperature rise and the loss due to discreteness of the liquid phase. The overall temperature rise depends upon the increase in temperature due to heat release and the decrease in temperature caused by direct cooling and evaporation of water. The pressure loss due to the presence of droplets and ligaments is a complicated function of the form in which the water is present as well as its mass fraction. For example, in Figure 54, there are two instances (one for $\dot{m}_a = 0.454$kg/s and one for $\dot{m}_a = 0.363$ kg/s) where the pressure drop across the combustor reduces slightly when 2 per cent spray is admitted.

<u>Temperature Increase Factor:</u> The total temperature increase across the diffuser-combustor unit is another significant performance parameter. A relative temperature increase factor may be defined as follows:

$$F_{TI} = \frac{\left(\frac{\Delta T_0}{T_0}\right)_{a+w} - \left(\frac{\Delta T_0}{T_0}\right)_a}{\left(\frac{\Delta T_0}{T_0}\right)_a} \tag{3}$$

$$\frac{\Delta T_0}{T_0} = \frac{\left(T_{04} - T_{0ref}\right)}{T_{0ref}}, \tag{4}$$

T_{oref} being a reference total temperature upstream of the prediffuser and T_{o4} being the combustor exit total temperature.

It should be noted that during some cases in the series of oxygen addition tests, the combustor exit temperature was higher than the temperature obtained under operating conditions with air only.

The factors F_{PL} and F_{TI} provide a measure of the changes caused in the stagnation pressure loss and the temperature increase due to combustor operation with water. The changes affect directly the thrust produced by the engine. Appendix D provides a detailed description of the implications of changes in T_{o4} and P_{o4}.

<u>Recovered Combustor Exit Temperature:</u> Under each set of operating and injection conditions, the highest combustor exit temperature that could be realized with adjustment of equivalence ratio (to a maximum value of 0.4) is presented as a performance parameter.

5.3. ESTIMATION OF PRESSURE LOSS ACROSS THE COMBUSTOR SECTOR.

A relationship between the pressure loss across the combustor, under cold flow conditions, and the corrected air mass flow rate entering the combustor has been established for corrected mass flow rates ranging from 0.32 to 0.5 kg/s. The results are presented in figure 53.

5.4. PERFORMANCE AND FLAMEOUT.

The conditions pertaining to the performance and flameout tests are listed in table 7 as cases 1 and 2. The results are presented in figures 54-59. The combustor performance parameters, F_{PL} and F_{TI}, are plotted as a function of the mass fraction of water injected in spray form. In each figure, the mass fraction of injected water that resulted in a flameout can be read off the abcissa directly below the data point corresponding to the highest mass fraction of injected water.

Observations from figures 54 to 57 indicate that an increase in the corrected mass flow rate of air has a significant effect on the amount of injected water that results in a flameout. For example, in Case 1, the combustor, operating with an air mass flux of 0.45 kg/s, experiences a flameout with and X_{wd} of only 6 per cent whereas, with an air mass flux of 0.36 kg/s, and X_{wd} of 12 to 18 per cent is required to extinguish the flame.

By comparing the results of cases 1 and 2, the effect of the two droplet sizes utilized in the spray can be observed. For the case of large droplet injection, the flameout limits are slightly extended, that is, the X_{wd} required to produce a flameout is larger, compared to the case of small droplet injection. This could be explained in terms of increased droplet evaporation due to the smaller droplet sizes, and hence increased vitiation of the air, resulting in a decrease in the flame speed and stability limits.

5.5. RECOVERABILITY OF COMBUSTOR EXIT TEMPERATURE.

The conditions for the tests involving an attempt to recover the combustor exit temperature for various air-water mixtures are given in table 7 as cases 3 and 4. The results, given in figures 60 to 62, are presented in the form of a time history of the combustor exit temperature for the duration of each recovery test. The start and end of each recovery stage are indicated by appropriately labeled vertical lines in the figures.

From figures 60 to 62 it can be seen that attempts to recover the exit gas temperature loss, due to the injection of an air-water mixture, have had limited success. For the case of small droplet injection, case 3, a complete recovery of combustor exit temperature, for both high and low air flow rates, is only possible for the case with an X_{wf} of 2.0 per cent. It is not possible to obtain a complete recovery of the combustor exit temperature under any conditions with spray injection. These observations appear to be valid for both high and low air mass flow rates. However, in case 4, it is possible to obtain a complete recovery for mass fractions of injected spray of 4.0 per cent or less, again illustrating the differences between the injection of large and small droplets.

5.6. EFFECT OF INJECTED WATER TEMPERATURE.

The conditions for the tests carried out with a supply of chilled water in place of room temperature water are given in table 7 as case 5. The results are presented in figures 63 and 64. The combustor exit temperature is plotted as a function of the mass fraction of injected spray. For comparison purposes, the relationship between the combustor exit temperature and the mass

fraction of injected spray, obtained with water at ambient temperature and the combustor operating under identical conditions, is included in each figure.

From figures 63 and 64 it can be seen that a reduction in the temperature of the water does not have a large effect on the exit gas temperature changes, however, it should be noted that the water was only chilled to 7°C.

5.7. EFFECT OF COMBUSTOR ORIENTATION.

The conditions for the tests carried out in setting 2 of figure 14 are given in table 7 as Case 6. The results are presented in figures 65 and 66. The temperature loss factor is plotted as a function of the mass fraction of injected spray. For comparison purposes, the F_{TI} obtained under identical conditions with the combustor oriented in setting 1 is included on each figure. Flameout tests were not carried out in this position for two reasons. Firstly, under conditions of large mass fractions of water injection, the observation window adjacent to the gravitationally displaced cup would become obstructed (due to flooding) which would lead to erroneous outputs from the photo-diode flame detector, and secondly, it was in all likelihood expected that a flameout would occur in the gravitationally displaced cup first and would spread to the other two cups. Since the flame detector could not focus on a particular cup in general, this too would lead to flooding of the sector with unburned fuel from the injector in the cup that had flamed out. for this reason, a maximum mass fraction of injected water had been set at 6.0 per cent.

In figures 65 and 66, the effect of the combustor orientation on the reduction in T_{04} is quite evident. The major cause for this phenomena may be attributed to the increased flow of water through the gravitationally displace cup, as a result of the effect of gravity on the flow of water in the prediffuser.

5.8. EFFECT OF HEATED AIR AND HEATED FUEL.

The combustor operating conditions utilized for this series of tests are given in table 7 as cases 7 and 8. The results are presented in the form of observations obtained during the tests.

It is observed that the overall effect of either an increase in the air temperature or an increase in the fuel temperature was a reduction in the mass fraction of injected water required to result in a flameout. In the case of heated air, the effect is more severe than in the case of heated fuel. This could be attributed primarily to the increased vitiation of the air with the products of combustion from the methanol fueled air heater, however, the increased air temperature could also have resulted in slightly larger quantities of water being vaporized. The increase in fuel temperature may have caused slight changes in the viscosity of the fuel and hence changes in the atomization performance of the injectors, as well as reducing the time required for complete vaporization of the fuel. This could have lead to slightly more intense combustion in a short distance in front of the injectors, which gives rise to increased vaporization of water and hence, increased levels of vitiation.

5.9. COMBUSTION WITH OXYGEN ADDITION.

The conditions under which these tests were carried out are presented in table 8 as Case 1. The results are displayed in figures 67 to 70. The combustor performance parameters, F_{PL} and F_{TI}, are plotted as a function of the mass fraction of water injected in spray form. (It may be noted that F_{TI} values for the case of oxygen addition tests are shown as positive values, since there is

an increase in temperature rise during oxygen addition compared to the case of operation with air. For comparison purposes, the values of F_{PL} and F_{TI} obtained under identical conditions without enriched air are included in each figure.

With reference to figures 67 to 70, it can be seen that the effect of enriching the air with oxygen leads to larger increases in pressure loss across the combustor, and hence large values of F_{PL}. This may be attributed to increased combustion generated pressure losses as a result of the local increase in combustion intensity in the primary zone as a result of air enriched with oxygen. In terms of the temperature increase factor, the addition of oxygen tends to result in a slightly larger temperature rise across the combustor. This is particularly noticeable with mass fractions of injected water less than 4.0 per cent. This indicates that in cases with small amounts of water in the primary zone, vitiation appears to be the dominant cause resulting in lower heat release in the primary zone. In this case, the addition of oxygen is somewhat effective at reducing the vitiation of the air, however, under conditions of large mass fractions of injected water it appears that the effects of gas temperature cooling tend to dominate.

5.10. RECOVERABILITY OF COMBUSTOR EXIT TEMPERATURE WITH OXYGEN ADDITION.

The operational conditions for this series of tests are summarized in table 8 as Case 2. The results, given in figures 71 and 72, are presented in the form of a time history of the combustor exit temperature for the duration of each recovery test. Start and end of each recovery stage are indicated by appropriately labeled vertical lines in the figures.

In the case of injection of mass fractions of spray of 4.0 per cent or less, the increase in temperature as a result of a combined increase in fuel equivalence ratio and oxygen content in the air seems to be adequate to regain the combustor exit temperature obtained during operation with air. Similarly when there is 2 per cent injection of film, the combustor exit temperature can be recovered. However, when spray as well as film are injected, the gain in temperature is considerably less, and decreases further as the spray content is increased.

6. DISCUSSION.

A discussion of the results obtained in the cold flow and burning tests is provided in this chapter. The test results on the overall water distribution in the combustor are discussed in section 6.1. An analysis of the effects of vitiation on the laminar flame speed and the flame stability is given in section 6.2. A discussion on the results of the burning tests is presented in sections 6.3 through 6.5. The results of the burning tests with oxygen addition are discussed in section 6.6. In section 6.7, an attempt has been made to establish the effects of changes in the fuel equivalence ratio and oxygen concentration in air, individually and in combination as a function of air flow velocity, or, equivalently, engine operation setting. finally, in section 6.8, a tentative analysis of the effects of changing the fuel and oxygen equivalence ratios is presented based on energy balance considerations.

6.1. OVERALL WATER DISTRIBUTION.

Considering the type of air-water mixture conditions, the amount of water entering the primary zone is a complex function of the air-water mixture conditions at entry to the prediffuser, the air flow velocity, and the effects of gravity on the flowfield.

Concerning the air-water mixture it appears that, for the current prediffuser-combustor configuration, most of the water in film form enters the primary zone (figures 27 to 34). Through flow visualization it has been established that the water enters the primary zone predominantly through the primary jets, with some of the water entering through the swirl cups. An increase in the air mass flow rate result sin a larger fraction of the film entering the primary zone. In addition, gravity has a substantial effect on the motion of the film, and on the quantity of water entering the primary zone. In figures 32 and 34 it can be seen that in all cases, 90 per cent (or more) of the injected film enters the primary zone when the combustor is oriented in setting 2, compared to 80 per cent (or more) when the combustor is in setting 1. The amount of water entering the primary zone of the combustor is thus influenced by the air flow velocity, the quantity of film, the orientation of film flow with respect to gravity, and the location of break-away of the film form the diffuser wall. The location of the film break-away mainly influences the flow split of film entering the primary zone through the swirl cups and primary jets.

Considering the case of spray injection, it appears that the fraction of water entering the primary zone is influenced by the air mass flow rate, and the orientation of the combustor with respect to gravity. The amount of water entering the primary zone seems to be independent of the mean droplet size of the spray, except when the combustor is oriented in setting 2, and operating under conditions of low air flow rates with mass fractions of injected spray greater than 4.0 per cent. Referring to figures 27, 29, 31, and 33, it can be seen that in all cases an increase in air velocity results in a larger mass fraction of water entering the primary zone. At high air mass flow rates the fraction of water entering the primary zone appears to be almost independent of the mass fraction of injected spray, whereas, at low air flow rates the fraction of water entering the primary zone increases slightly with an increase in the mass fraction of injected spray. This phenomenon is more pronounced when the combustor is oriented in setting 2 (figures 31 and 33).

With reference to figures 28, 30, 32, and 34, is observed that the combined injection of film and spray into the test article results in a slight increase in the fraction of water entering the primary zone compared to cases with spray injection only. This can be explained in terms of the larger fraction of film entering the primary zone compared to the spray, thus indicating that the film has a greater tendency to enter the primary zone in the current prediffuser-combustor configuration.

Again, the fraction of water entering the primary zone appears to be independent of the mean drop size of the spray.

Finally, the investigation on the combustor oriented differently with respect to gravity, namely in setting 2, has shown that the flowfield is likely to display a complex asymmetry in regard to the growth of the film and the distribution of water in the primary zone of an annular combustor.

6.2. ANALYSIS OF EFFECTS OF VITIATION ON FLAME SPEED AND STABILITY.

The method of establishing the amount of water that may undergo evaporation in the primary zone, based on the use of video images, has been given in section 4.3. In order then to establish the effect of vitiation on flame dynamics, namely flame speed and flame stability, the methodology developed by Kretschmer and Odgers [reference 8] has been applied to selected cases in the cold flow tests carried out on the combustor. In order to utilize the methodology, some simplifying assumptions have been made:

1. The influence of temperature and mixture composition on the air loading factor are constant over the range of values of vitiation ratio supplied to the combustor.

2. A given value of equivalence ratio, that is, stoichiometric, is considered for all calculations.

These assumptions enable the air loading factor (and hence, the flame speed and stability limits) to be expressed as a function of vitiation ratio only. Thus, by measuring the total quantity of water entering the primary zone (for a given set of entry conditions) and the fraction of water that is likely to evaporate, the vitiation ratio can be determined and its resultant effect on the combustion parameters.

As an illustration of the application of the methodology, a propane combustor operating at an air inlet temperature of 600 K under various air-water mixture conditions has been considered. For each set of operating conditions, the value of the vitiation ratio has been determined utilizing the values of the fraction of water that is likely to evaporate in the primary zone that have been determined in section 4.3. Finally, once the mass of water vapor is known, the vitiation ratio is calculated and the estimated effects of the vitiation can be determined. The inlet flow conditions for each of the cases are presented in table 6. The calculated vitiation ratios, the resultant maximum air loading parameters and the ratios of the reduced flame speed, Su, to the maximum flame speed (for the ideal, unvitiated case), Su (O_2 = 21), are given in table 9.

TABLE 9. RESULTS FOR ESTIMATION OF EFFECTS OF VITIATION ON COMBUSTION

Case Number	m	N/VP^n (gmol/s)/L atm^n	Su/Su(O_2=21)
1	4.4	85.7	9.94
2	4.5	84.2	0.93
3	4.52	84.1	0.93

Note: $[N/VP^n]$=96 when m=3.76 in the unvitiated case and that (O_2=21): 21 per cent by volume in the combustor

Figures 73 and 74 provide typical examples of such effects in the case of the three-dimensional combustor sector model. The stability map indicates the ranges of values of fuel equivalence ratio, ϕ, over which the flame can be stable for the relevant values of combustor loading, NVP^n.

In each of the three cases under consideration, the combustor has been operated under conditions of low air flow rates, that is, 0.45 kg/s. Operating conditions with higher air mass flow rates have not been conducted since the pictures obtained at higher air flow rates have not been of adequate quality to distinguish between different droplets. It can be observed that for the three low flow rate case there seems to be very little effect of the estimated water vapor in the primary zone on the maximum value of air loading parameter. Similarly, the effects of vitiation on flame speed are similar to those on air loading (figure 73). The flame speed has been normalized with respect to the value of flame speed obtained by Kretschmer and Odgers for 21 per cent oxygen content, by volume, in the mixture. concerning the flame stability map, figure 74, it can be observed that there arises very little effect of vitiation on the map, and in fact the results correspond very nearly to conditions for no vitiation, m=3.76. In a similar analysis carried out by Minster [references 6 and 7] on two-dimensional prediffuser-combustor models, the foregoing results were also obtained for conditions of low air mass flow rate, however, at higher air mass flow rates, there arose a substantial reduction in the flame speed and flame stability limits.

During the flow visualization tests carried out under cold flow condition, described in Section 4.3, it has been observed that at low air flow rates, the size of the water droplets entering the primary zone are larger than those for the higher air flow rates. Also, more water seems to enter the primary zone at high air flow rates than at low air flow rates. Consequently, it can be expected that the fraction of water entering the primary zone that is likely to evaporate will be less in the case of low air flow rates, hence, the resulting effect of vitiation will be less than in the case of high air flow rates. These phenomena have been observed during the burning tests. At low air flow rates, large mass fractions of water, of the order of 15 per cent and higher, were required to induce a flameout, whereas at higher air flow rates, mass fractions of water of approximately 6.0 to 8.0 per cent resulted in a flameout.

6.3. EFFECTS OF WATER ON COMBUSTOR PERFORMANCE

Combustion processes and the combustor performance are affected by the ingestion of water into the engine in several ways. The following three phenomena, acting both individually and interactively, occur as a direct result of a given mass fraction of water entering the diffuser-combustor element: A reduction of the gas temperature in the combustor as a result of rapid evaporation of some of the water in the primary zone; a corresponding drop in pressure as a result of the reduction in the gas temperature due to the evaporation of water; and vitiation of air by water vapor.

The air supplied to the combustor sector during the tests is in all likelihood humid, although it may not be fully saturated. For a given amount of water supplied in film and droplet forms, the amount of water entering the combustor in total through the injector cups and the primary jets is known as a function of mass flow rate of air; however, in the current series of tests, the amount of water entering through the injector cups is not known in itself as a function of the amount of water entering the prediffuser in film and spray forms.

On injection of about 2.0 per cent by mass of water into the air stream entering the prediffuser, it has been found in the burning tests that the burning zone recedes towards the injector cups, as

observed in video photograph through the transparent windows, and the combustor exit temperature drops by about 100° to 150°C depending upon the combustor operating conditions. In the current test article with a dump-like prediffuser, the drop in temperature is also a function of the form, namely film or spray, in which the water is injected (figures 60, 61, and 62).

On increasing the amount of water injected, a series of changes is observed from visualization in the flame zone in succession: (i) First,. the flame zone tends to become shorter; (ii) next, the flame zone displays an oscillatoriness or unsteadiness in its extent; and (iii) finally, the flame may either go out or may display, at low air mass flow rates, a tendency to remain attached in the vicinity of the swirl cup. In the latter case, the flame may eventually be quenched only with the admission of very large amounts of water, of the order of 15.0 per cent and higher, that may introduce severe blockage through flooding in the region of the primary jets. It should, however, be noted that in setting 2 (figure 13), observation (ii) may apply particularly to the outer two cups, and observation (iii) to the center cup.

At low air mass flux and, correspondingly at low air velocity, it has been observed that a flame may not be quenched even with large amounts of water, although the exit gas temperature drops as the amount of water is increased (figures 55, 57, and 59). It may be reasonable to conclude that the amount of water entering through the injector cups may not increase in direct proportion to the amount of water injected initially into the air stream, an increase in the amount of water injected into the air stream affects primarily the amount of water entering the combustor through the primary jets, the flameout depends upon the vitiation occurring in the vicinity of the injector cups, and the additional water entering through the primary jets affects mainly the gas temperature by cooling and also, possible distortion of the flame and its propagation, and the occurrence of flameout may be due to a combination of causes related to vitiation, flame-water mechanical interaction, and chemical effects.

With reference to figures 58, 59, and 62, the effect of spray droplet size on the occurrence on flameout can be observed. For the case of large droplet injection, the flameout limits are slightly extended, that is, the X_{wd} required to produce a flameout is larger, compared to the case of small droplet injection. This could be explained in terms of increased droplet evaporation due to the smaller droplet sizes, and hence increased vitiation of the air, resulting in a decrease in the flame speed and stability limits.

The results obtained with a supply of chilled water in place of room temperature water have reinforced the conclusion that the processes in the immediate vicinity of the swirl cup are the ones that control the occurrence flameout (figures 63 and 64). In fact a reduction in the temperature of water does not seem to affect even the exit gas temperature changes to any substantial extent. However, it must be noted that the water has been chilled only to 7°C.

Considering the reduction in the gas exit temperature, it is found that, as the amount of injected water (at diffuser entry) is increased, the reduction for an additional amount of water is not as large as the initial reduction with 2.0 per cent. The nature of reduction in exit gas temperature remains nearly unchanged with changes in equivalence ratio at fixed mass flux of air, and also, to a substantial extent with changes in air mass flux at fixed equivalence ratio figures 60 to 62).

Lastly, the combustor setting (figure 14) has a major effect on performance changes. In setting 2, compared to setting 1, there are differences among the three injector cups in regard to (a) the distribution of water injected in film form, (b) the effect of addition of water in film form on (a), and (c) the effects on combustion (figures 65 and 66). The cup that is not displaced in the

gravitational direction in setting 2 yields nearly the same performance as the center cup in setting 1. The cup displaced most in the gravitational direction in setting 2 becomes affected by film injection and consequent water flow through the primary jets more than the outer cups in setting 1.

6.4. EFFECTS OF HEATING OF FUEL AND AIR.

Referring to the results in section 5.8, it has been observed that both the effect of heated air and of heated fuel resulted in a reduction in the amount of injected water required to result in a flameout For the case of heated air, the air was vitiated with the combustion products from the methanol-fueled air heater. Thus the increased vitiation of the air, combined with a slight increase in the fraction of water that could be expected to vaporize in the primary zone as a result of the higher air temperature, appeared to be the most likely cause for the flameout. For the case of heated fuel, an increase in the fuel temperature results in a slight decrease in the viscosity of the fuel and the time required for complete vaporization of the fuel. The change in the viscosity could be expected to have an effect on the atomization performance of the pressure swirl atomizers used in the three fuel injectors in the test combustor. Hence, the combustion may occur more intensely in a short distance (or volume) in front of the injectors. This larger heat source (more heat/unit volume) gives rise to increased evaporation of water, and hence, flameout. thus ultimately, in both cases, the flameout may be due to increased vitiation of the air.

6.5. EFFECTS OF CHANGES IN FUEL EQUIVALENCE RATIO.

Regarding exit gas temperature recovery by means of changing fuel equivalence ratio (figures 60 to 62), it can be seen that the change in exit gas temperature obtained in the case of film injection is larger than the obtained in the case of an equivalent amount of spray injection. This may be due to the fact that the spray enters the flame zone through an injector cup and affects combustion directly. However, when the air mass flux is increased, the flow through the primary jets increases on account of diversion of a larger fraction of water into the outer stream. In other cases, for example with spray only or film and spray, there arises no appreciable gain in the temperature rise. It may be pointed out here that a change in equivalence ratio gives rise simultaneously to an increase in (a) temperature due to increased fuel burning, (b) heat and mass transfer of the water, and (c) vitiation by water vapor. The observed results are due to an interactive combination of the three changes.

6.6. EFFECTS OF ADDITION OF OXYGEN.

Under conditions of constant air mass flux and fuel equivalence ratio, it has been observed that when a small quantity of water (2.0 per cent) is injected in spray or film form, there arises a substantial reduction in exit gas temperature with air. On adding 2.0 per cent oxygen, it is found that there is a substantial gain in temperature (figures 67-70). Any oscillatoriness in the flame zone disappears completely. When the air-water mixture contains spray at a mass fraction well above 2.0 per cent, there does not arise a substantial, further reduction in temperature, without or with film. However, when air is enriched with 2.0 per cent oxygen, the gain in temperature seems to depend on whether water injection includes a film or not. Thus the gain in temperature with oxygen seems to be the lowest with 2 per cent each of film and spray, and with further increases in the spray content, the gain seems to increase slightly. The qualitative conclusion from this is that the processes of water evaporation, water heating, vitiation, and reduction in gas phase temperature interact in a complex fashion, both in the final value of combustor exit temperature obtained, and also in the stability of the flame. In particular, the changes in the local temperature and the proportion of unburned hydrocarbons are affected strongly by the changes in

the proportions and physical characteristics of water entering through the swirl cups and the primary jets.

When the combustor is operated under conditions of increased fuel equivalence ratio and oxygen addition (figures 71 and 72), the combined addition of fuel and oxygen can give rise to a larger quantity of local heat release in the primary zone, and hence an improved combustor temperature recovery when compared to the cases of either an increase in fuel equivalence ratio or an increase in the oxygen content in the air.

6.7. EFFECT OF AIR FLOW VELOCITY DURING CHANGES OF ϕ AND ϕ_O.

During operation with air-water mixtures, one of the parameters of interest is the air flow velocity, which is proportional in the current tests to the air mass flow rate. The influence of air flow velocity may be examined by considering, for example, the possibility of recovering the combustor exit temperature under conditions of an increase in the fuel equivalence ratio, an increase in the oxygen content in the air, and a simultaneous increase in the fuel equivalence ratio and the oxygen content in the air for the following cases, namely (1) injection of 2.0 per cent film, (2) injection of 2.0 per cent spray, and (3) the combined injection of 2.0 per cent film and 2.0 per cent spray.

Several features may be noted in regard to these different cases. In the case where water is injected in film form, the water enters the combustor largely through the primary jets. Hence it can be expected that only a part of it becomes recirculated into the part of the primary zone that is adjacent to the swirl cups. On the other hand, in the case where water is injected in spray form, the water enters the primary zone through the cups. Thus, there arises a basic difference in the distribution of water in the two cases. Next, an increase in fuel equivalence ratio affects the heat release by a change in the total amount of fuel burned. In general, when the fuel equivalence ratio is small, the heat release is increased by an increase in fuel equivalence ratio. An increase in oxygen content of air, on the other hand, gives rise to increased heat release by an improvement in combustion efficiency. In both cases, the fuel distribution and the heat transfer to the liquid phase may be controlling factors. Finally, an increase in air flow velocity reduces the residence time of the mixture in the primary zone, and, therefore, has an effect on the magnitude of changes in heat transfer and transport, and distribution of fuel droplets. With reference to figures 60 to 62, 68, and 70 to 72, the results of the 'recovery tests' for the aforementioned cases are summarized in table 10.

TABLE 10. COMBUSTOR EXIT TEMPERATURE RECOVERABILITY WITH 2.0 PER CENT INJECTED WATER IN FILM AND SPRAY FORM

$\dot{m}_a$ (kg/s)	Case 1: 2.0 Per Cent Film Injection		
	Increased ϕ	Increased ϕ_O	Increased ϕ and ϕ_O
0.36	possible	not possible	possible
0.45	not possible	not possible	possible
	Case 2: 2.0 Per Cent Spray Injection		
0.36	possible with large droplets; and not possible with small droplets	questionable	possible
0.45	questionable	not possible	possible
	Case 3: 2.0 Per Cent Film and Spray Injection		
0.36	not possible	not possible	not possible
0.45	not possible	not possible	not possible

In the case of the low air flow rate, with either 2.0 per cent film (case 1) or spray (case 2), the combustor exit temperature can be recovered with either an increase in the fuel equivalence ratio, or a combined increase in fuel equivalence ratio and oxygen content in the air. Concerning the former, in case 2, the temperature could only be recovered in the case of large droplet injection. However, at the high air flow rate, an increase in the fuel equivalence ratio, alone, did not result in a complete recovery of the exit temperature for either case 1 or case 2, but, a combined increase in the oxygen content of the air and the fuel and the fuel equivalence ratio is sufficient for a complete recovery of the exit gas temperature. An increase in the oxygen content of the air, by itself, did not result in a complete recovery, except in the case of low air flow rates, high equivalence ratio, and spray injection. In the case of a combined injection of 2.0 per cent film and spray (case 3), a recovery of the exit temperature was not possible under any conditions of increased fuel equivalence ratio, increased oxygen content in the air, or a combination of both an increase in fuel equivalence ratio and oxygen content in the air.

In the cold flow tests (described in section 4), it has been observed that at the high air mass flow rate, a larger mass fraction of the injected water enters the primary zone than in the case of the low air mass flow rate. Further, at the high air mass flow rate the water enters in the form of a fine droplet mist, whereas, at the low air flow rate, the droplet size seems to be larger. Considering these observations, it can be concluded that the air flow velocity has a significant effect on the quantity and distribution of water entering the primary zone.

Concerning the burning tests, the presence of water can be seen to have two separate but interactive effects on the net heat release: an effect through heat transfer from the gaseous to the liquid phase, and an effect due to the modifications in the design fuel distribution by the action of liquid entities. Thus, in the case when water enters in film form, since the water is predominantly in the form of large ligaments entering through the primary jets, only part of the water could be expected to affect the distribution of fuel in the near-cup region of the primary zone. On the other hand, water in spray form enters the primary zone largely through the swirl cups in the form of a mist, and hence, gives rise to a more uniform distribution of water. As a consequence, the water distribution leaving the swirl cups can be expected to affect the amount of heat transfer to the water, and also the fuel distribution in the primary zone. Since the water distributions affected by the air flow velocity, it can be surmised that the fuel distribution is affected by the air velocity. Referring to table 10, it can be seen that at the high air flow rate, the

possibility of a recovery of exit gas temperature is significantly less than in the case of low air flow rates, thus indicating that the air velocity is a significant parameter in the performance of a gas turbine combustor operating with an air-water mixture.

It is of interest to note here that Williams [reference 19] also pointed out, as stated in section 1.2, that in the case of liquid suppressants of fire, a major factor controlling extinction is the effect of the suppressant on the fuel distribution.

6.8. ANALYSIS OF EFFECTS OF CHANGES IN ϕ AND ϕ_o

A tentative, simplified analysis of the measurements and observations made during the tests with modifications in fuel equivalence ratio and oxygen content of air may be attempted as follows.

Six test cases (based on the same classification as given in section 3.1) are considered as follows:

Case 1: Operation with air only (denoted as A-A).

Case 2: Operation with air and water (denoted as A-W).

Case 3: The same as (A-W) with a modified value of fuel equivalence ratio (denoted as B).

Case 4: Operation with air enriched with 2.0 per cent oxygen (denoted as C-A).

Case 5: The same as (A-W) with addition of 2.0 per cent oxygen to air (denoted as C-W).

Case 6: The same as (C-W) with a modified value of fuel equivalence ratio (denoted as D).

An analysis has been carried out to establish the fuel fraction that can be expected to have undergone combustion based on enthalpy changes across the combustor in the six cases.

The experimental data for the chosen cases are given in table 11.

TABLE 11. BURNING TEST CASE CONDITIONS

Case	$\dot{m}_a$ (kg/s)	Fuel ϕ	Fuel (g/s)	X_w	X_{o2}	T_{o4} (K)	T_{o3} (K)
A-A	0.363	0.24	5.89	0	0	588	293
A-W	0.363	0.24	5.89	2	0	433	293
B	0.363	0.4	9.81	2	0	503	293
C-A	0.363	0.24	5.89	0	2	611	293
C-W	0.363	0.24	5.89	2	2	460	293
D	0.363	0.31	7.60	2	2	578	293

6.8.1. Parameters Involved.

The following parameters have been utilized in the analysis.

1. Geometry of the combustor: The ratio of the areas between the primary, dilution, and cooling holes in the combustor liner have been specified for the test article and hence the flow split between the primary and secondary streams is fixed.

2. Inputs: For each case values for the following experimental parameters have been obtained and utilized in the subsequent analysis, namely fuel flow rate, air mass flux, air inlet temperature, combustor exit gas temperature, and mass fraction of injected water.

3. Outputs: The results of the analysis provide estimates for the overall combustor efficiency, ideal and actual quantities of heat release due to the combustion of fuel, and fraction of fuel that can be expected to have undergone combustion.

6.8.2. Basic Experimental Constraints.

In the course of the combustion tests, values of the following experimental parameters have not been obtained: the quantity of unburned hydrocarbons in the combustor exit gas, and the state, quantity and temperature of water, if any, in the combustor exhaust gas. Consequently, a complete heat balance, accounting for unburned hydrocarbons, incomplete reactions, and heat transfer to water, cannot be undertaken within the scope of measurements of the current investigation.

6.8.3. Assumptions.

In order to carry out the analysis, the following assumptions have been made: fast chemical reactions, and vaporization of all of the water in the primary zone by flash evaporation.

6.8.4. Approach

The analysis has been carried out in the following manner.

1. First, an estimate of the mass fraction of water and the local fuel equivalence ratio in the primary zone, ϕ_f, has been obtained utilizing data obtained from the combustion tests and the cold flow water collection tests. With these values, an estimate of the effect of vitiation on the adiabatic flame temperature in the primary zone of the combustor, for various values of ϕ_f, has been obtained by utilizing the NASA CEC85 computer code [reference 23].

2. An estimate of the overall combustor efficiency, based on actual experimental data, has been calculated for the range of ϕ_f and $\dot{m}_a$ utilized in burning tests.

3. Finally, an enthalpy and a fuel balance across the combustor has been carried out, taking into account the energy required to vaporize the water, in order to obtain an estimate of the fraction of fuel that can be expected to have undergone combustion. An estimate of the flame temperature has also been obtained.

As an illustration of the approach, the complete calculation procedure for case 6 is presented in appendix C as an example.

6.8.5. Results of the Analysis.

The effect of the presence of water and of water vapor is to reduce the flame temperature. Assuming the reaction kinetics as in reference 23, one can estimate the reduction in the flame temperature as a result of the effect of vitiation on combustion (figure 75).

The overall efficiency of the combustor during operation with air (only) is provided in figure 76. This can be rationalized on the basis of incomplete reactions and unburned fuel, and heat transfer to the wall.

With reference to the results presented in table 12 the following observations can be made with respect to each of the five test cases under consideration.

Case A-A: In this case, the fuel equivalence ratio is 0.24, and the combustor is operated with air only. The overall combustor efficiency, η_c, is approximately 48 per cent. Utilizing this value of η_c the amount of fuel that is likely to undergo combustion, and the corresponding local fuel equivalence ratio in the primary zone have been established. With the value of ϕ_f as an input to the NASA CEC85 code, an estimate of the resulting flame temperature in the primary zone has been found to be approximately 1079 K.

TABLE 12. ENTHALPY AND FUEL BALANCE RESULTS

Case		A-A	A-W	B	C-A	C-W	D
Enthalpy	Ideal heat release, kJ	252	252	420	252	252	325
Balance	Actual heat release, kJ	122	56	86	130	66	118
	Heat for vaporization of water, kJ	-	18.8	18.8	-	18.8	18.8
	Overall efficiency η_c	48	30	25	52	34	42
Fuel	Equivalence ratio, ϕ	0.24	0.24	0.4	0.24	0.24	0.31
Balance	Fuel flow rate, g/s	5.89	5.89	9.81	5.89	5.89	7.60
	Est. fuel burned	2.83	1.75	2.45	3.04	1.99	3.20
	Est. flame temperature, $^{\circ}$K	1079	615	614	1172	615	1150

Case A-W: The effect of combustion with an air-water mixture is presented in this case. A mass fraction of 2.0 per cent of water is injected into the air supply upstream of the prediffuser. When compared to case A-A, the actual heat release for this operating condition is reduced by 66 kJ and the resulting overall combustor efficiency, η_c, drops to 30 per cent. The difference between η_c in case A-A and η_c in case A-W can be explained on the basis of the sum of effects of heat transfer to water, vitiation of the air by water vapor, incomplete chemical reactions, incomplete combustion of fuel, and reduction in heat transfer to the walls.

Case B: In this case, the fuel equivalence ratio is increased from 0.24 to 0.4 in an attempt to recover the temperature loss across the combustor as a result of a 2.0 per cent mass fraction of injected water. When compared to case A-W, the increase in fuel equivalence ratio results in a slight increase in the heat release, however, the overall combustion efficiency drops by 5.0 per cent in this case.

Case C-A: This case illustrates the effect of operating the combustor with air enriched with 2.0 per cent oxygen. When compared to case A-A, it can be observed that the additional oxygen results in an improvement in the combustor efficiency by approximately 4.0 per cent, which, in turn, leads to an increase in the local heat release of about 8 kJ.

Case C-W: This case illustrates the effect of combustion with an air-water mixture and air enriched with 2.0 per cent (by mass of air) of gaseous oxygen. This results in a slightly larger amount of heat release than in case A-W, and less than in case B, however, the overall combustor efficiency increases to 34 per cent.

Case D: Finally, this case illustrates the effects of a combined increase in ϕ and ϕ_O on the recoverability of the combustor exit gas temperature. In this case a complete recovery has been obtained with a ϕ of 0.31. This value of ϕ is lower than that in case B, however, the addition of 2.0 per cent oxygen results in an improved combustion efficiency in the primary zone and hence a substantially larger heat release. This confirmed the observation made during the burning tests with oxygen addition that a combined increase in the fuel and oxygen equivalence ratios was the most effective method of increasing the local heat release in the primary zone of the combustor.

6.8.6. Sources of Error in the Estimates.

Although the heat balance analysis has provided a qualitative explanation of the observed result, several sources of error may be noted in the estimates as follows:

(a) Heat required for complete vaporization of water: The state and temperature of the water exiting the combustor have not been measured, and thus, whether all of the water leaves the combustor in vapor from is unknown.

(b) Estimate of fuel burned: The quantity of unburned hydrocarbons in the exhaust gas has not been measured; hence, the estimate of unburned fuel, as well as that of local fuel equivalence ratio, based on the heat release estimates may be inaccurate.

(c) Estimated flame temperature: The value of local equivalence ratio, is utilized as an input in order to obtain an estimate for the flame temperature. In some cases, the estimated value of local equivalence ratio is predicted to be slightly below the lean flammability limit for the hydrocarbon fuel, $C_{10}H_{20}$, utilized in the code.

6.9. SUMMARY OF OBSERVATIONS.

1. The redistribution of water in the different streams and parts of the diffuser-combustor element is strongly dependent upon the form and mass fraction of water at entry to the diffuser and the air velocity in the different parts of the element.

2. The form of water in the primary zone is predominantly a function of air flow velocity, although the amount of water entering through the primary jets may depend to some extent on the form of admission of water in the diffuser.

3. The temperature increase factor and the pressure loss factor can be related to water entry conditions to the diffuser in a given diffuser-combustor element.

4. Following a reduction in the temperature rise across the combustor with a given air-water mixture, some recovery in the temperature rise is possible by a modest change in equivalence ratio only in the case of water in film form only. In other cases, for example with spray only or film and spray, there arises no appreciable gain in the temperature rise. It may be pointed out here that a change in equivalence ratio gives rise simultaneously to an increase in temperature due to increased fuel burning heat and mass transfer of the water, and vitiation by water vapor. The observed results are due to an interactive combination of the three changes.

5. The occurrence of flameout is a function of air mass flow, and temperature, form, and mass fraction of water. When compared to the case of small droplet spray injection, an increase in the mean droplet size leads to an increase in the amount of injected spray required to cause a flameout. This may be understood by noting that the water enters predominantly through the primary jets in large sized ligament form, and therefore undergoes little heating and vaporization.

6. It may be noted that a reduction in temperature of water (by chilling) tends to reduce vitiation by water vapor while giving rise to a small increase in the reduction of air temperature. The observed results must be understood in terms of the interactive combination of those two effects.

7. Attempts at recovery of exit gas temperature by an increase of fuel equivalence ratio have proved successful with low mass fractions of water in either film form or large droplet spray form, when the water could be expected to enter the primary zone mainly through the primary jets. On the other hand, an increase of fuel equivalence ratio was not effective when the water entered in small droplet spray form, largely through the swirl cups.

8. While the foregoing apply in general for both combustor settings 1 and 2, the effects of gravity on film motion and the resulting changes in combustion performance become substantially increased for the cup that is most displaced in the gravitational direction. It can be surmised that even in setting 1 the two side cups may suffer, but in setting 2, the effect is worse.

9. The effect of enrichment of air with oxygen is always a gain in combustor temperature unless the increase in heat loss to water and vitiation due to vapor generation tends to reduce the gain to a negligible value. For a fuel equivalence ratio of 0.24, in the case of spray injection only, the gain in temperature is the largest for values of X_{wd} less than 4.0 per cent; and for combined film and spray injection, the gain in temperature is largest for values of X_{wd} higher than 4.0 per cent. For a fuel equivalence ratio of 0.3, in the case of spray injection only, and combined film and spray injection, the gain in temperature is the largest for values of X_{wd} less than 4.0 per cent.

10. The combined effects of an increase in fuel equivalence ratio, and an enrichment of air with oxygen is to permit a regain of the temperature to the value obtained during operation with air (only), except in the case of injection of over 4.0 per cent water, when the effects of cooling and perhaps also, vitiation tend to dominate.

11. The air flow velocity affects the distribution of water in the primary zone, which in turn can be expected in influence the fuel distribution and the interphase heat transfer in the near-cup region of the primary zone, and hence the combustor exit temperature. Thus, for a

given state of air-water mixture, the mass flux through the combustor becomes a significant parameter.

12. Based on the foregoing, it can be projected that in practical, fuel annular combustor operating with an air-water mixture from a compressor the 6 o'clock cup is the one that becomes affected the most readily and the greatest, all of the remaining cups, except the 12 o'clock cup, undergo progressively less deterioration, and display unstable characteristics and eventual flameout with increases in water mass fraction and air mass flux, the 12 o'clock cup is probably the least affected and may remain lit up even with large mass fractions of water, especially at low air mass flux rates, and the exit gas temperature drops appreciably even with a small mass fraction of water, it is neither possible to recover the exit gas temperature nor establish a relight and a well distributed flame across the combustor cross section even with a change in fuel equivalence ratio only so long as the water content in the air remains high, and the pattern factor at the combustor exit may tend to become extremely poor and, possibly, time-dependent. Concerning regain of combustor exit temperature it appears that an increase in fuel flow rate, if accompanied by a modest amount of enrichment of air with oxygen (under 2 per cent), will result in nearly complete recovery, at least up to about 4 per cent of spray in the mixture. The oxygen requirement may be under 10 kg per engine for 5 minutes of operation under flight idle conditions in the current case.

13. The results of the tests performed with modified fuel equivalence ratio, and oxygen-enriched air may also suggest in the case of a practical engine a number of operational changes, for example in the form of engine speed regulation for different power demand settings, that can yield the same benefits as additions of fuel and oxygen.

It may be noted that the foregoing observations are based on tests conducted with near-atmospheric air, both in temperature and pressure. While the general trends in the results may not change if the combustor is operated with heated air, under pressurized conditions, there may arise substantial changes in the quantitative estimates of performance changes and flameout conditions.

7. CONCLUSIONS.

1. The amount of water entering the primary zone, as a fraction of the amount of water entering the diffuser, is high. But the size and space-wise distribution of the liquid entities in the combustor is a complex function of air water mixture conditions in the prediffuser, the air mass flux and distribution, and the design parameters of the swirl cups and primary jets.

2. The distribution of water in an annular combustor is strongly affected by the influence of gravity on the flowfield in the secondary streams in the outer section of the combustor.

3. The presence of water in the flow does not seem to affect the stagnation pressure loss across the sector.

4. The water distribution in the primary zone is affected by the form of water entering the prediffuser in addition to the air flow velocity. The water distribution affects both the heat transfer and the fuel distribution. Therefore, the air mass flux has a significant influence on the net heat release in the combustor.

5. During air-water mixture operation, heat loss to water, through heating and vaporization, resulting in a possible decrease in flame temperature in the primary zone, appears to be the dominant process leading to temperature loss across the combustor.

6. Considering the temperature reduction arising during operation with a small amount of water in the air-water mixture supplied, it is overcome only by means of an increase in heat release that is in excess of the heat required to heat and vaporize the water in the primary zone.

7. Among the possible means for increasing the heat release, namely an increase in fuel equivalence ratio, an enrichment of air with oxygen, and an increase in fuel and enrichment of air, it appears that the first two are effective in cases where there is little entry of water through the cups, while the third provides improvement over a large range of water mass fraction entering either through the cups or the primary jets.

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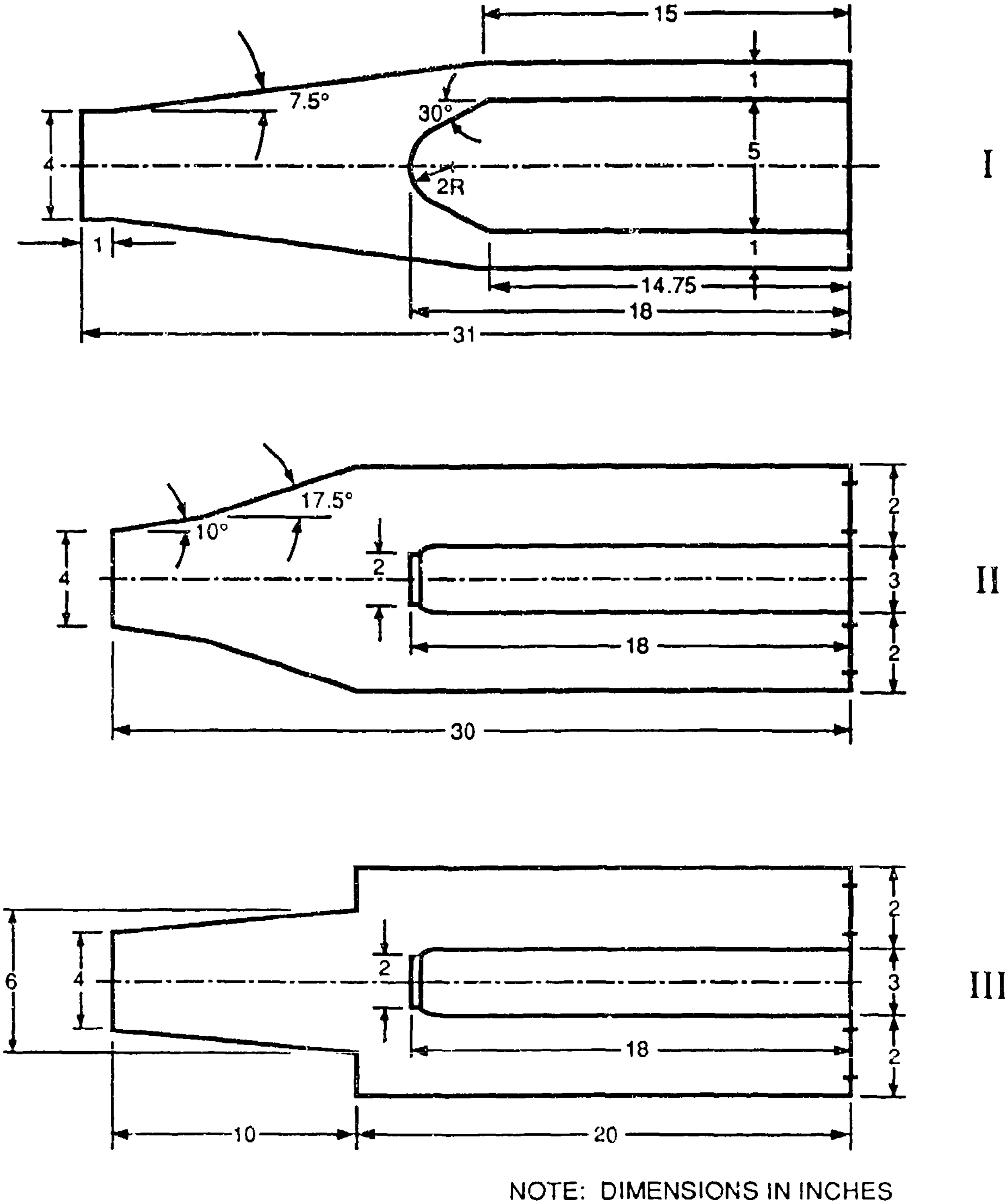


Fig. 1. Two-Dimensional Model Prediffuser-Combustor Configurations

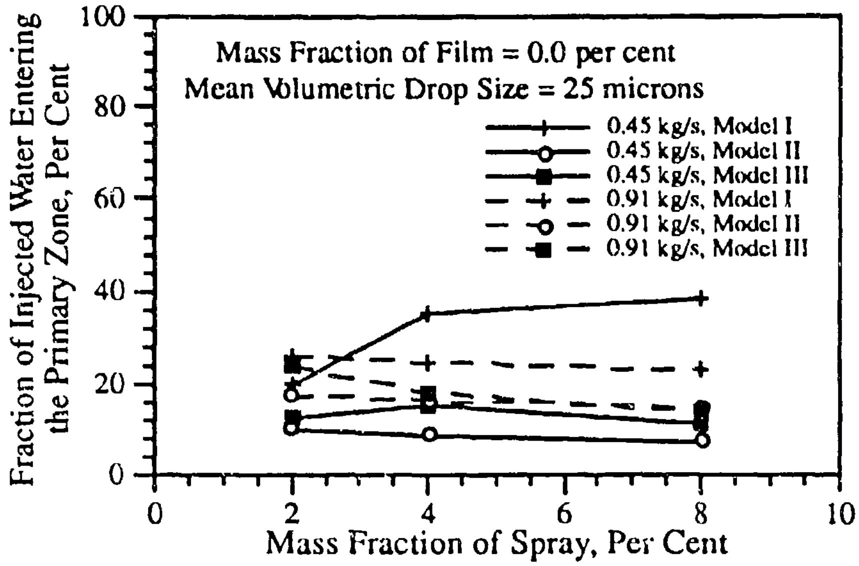


Figure 2. Effect of Mass Fraction of Injected Spray of Small Droplet Size on Water Entry into the Primary Zone Through the Dome and Primary Jets

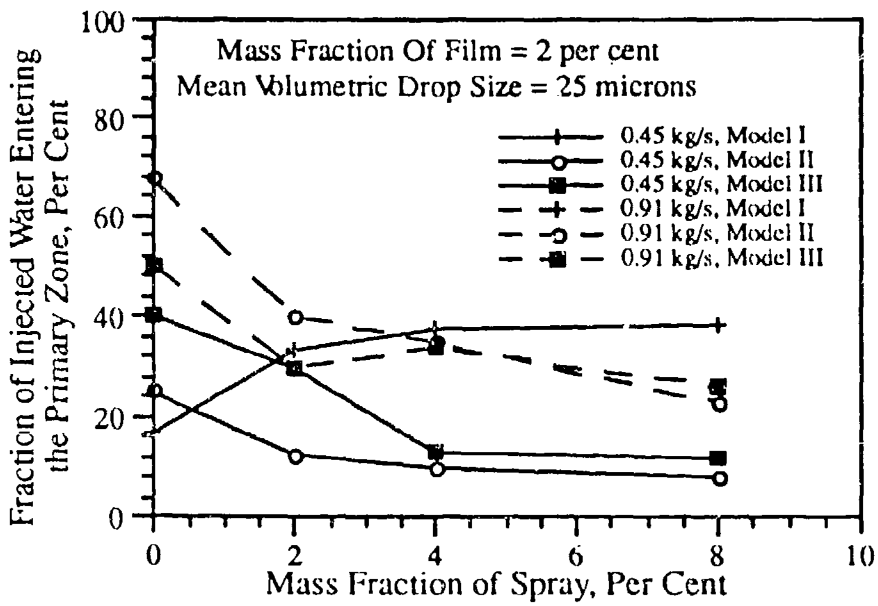


Figure 3. Effect of Mass Fraction of Injected Film and Spray of Small Droplet Size on Water Entry into the Primary Zone Through the Dome and Primary Jets

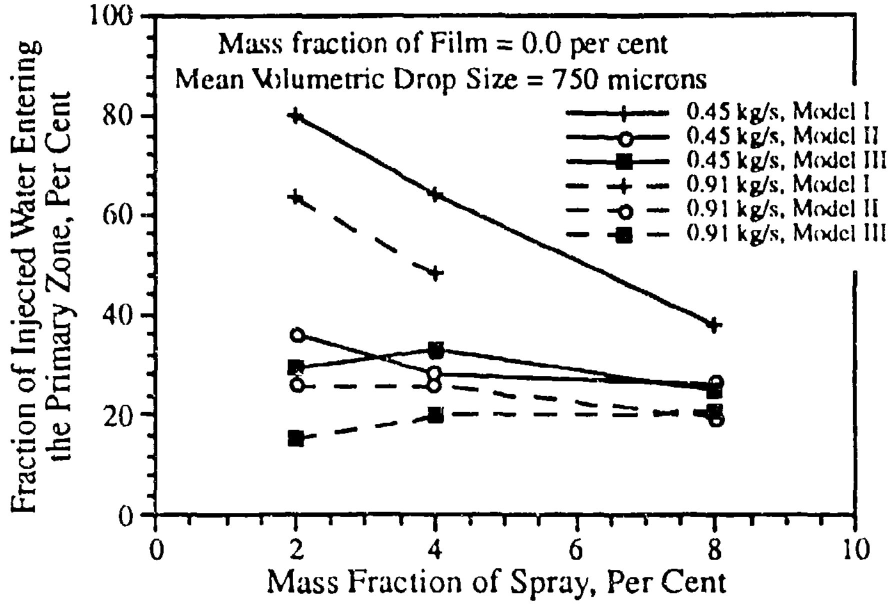


Figure 4. Effect of Mass Fracton of Injected Spray of Large Droplet Size on Water Entry into the Primary Zone Through the Dome and Primary Jets

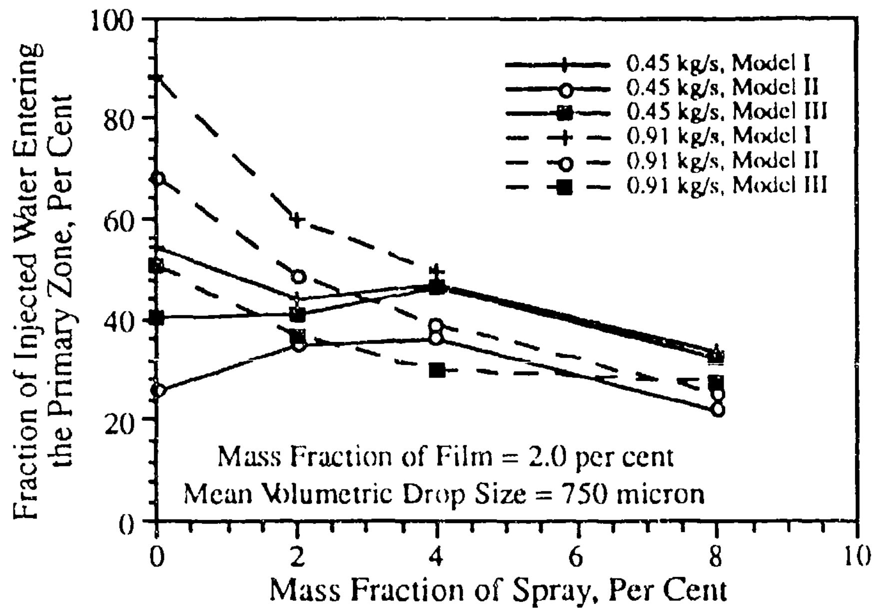
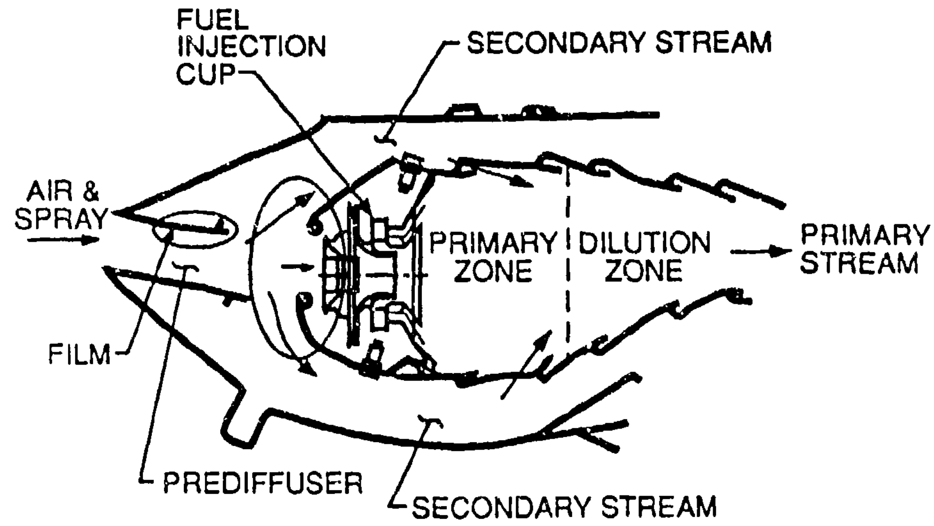


Figure 5. Effect of Mass Fracton of Injected Film and Spray of Large Droplet Size on Water Entry into the Primary Zone Through the Dome and Primary Jets



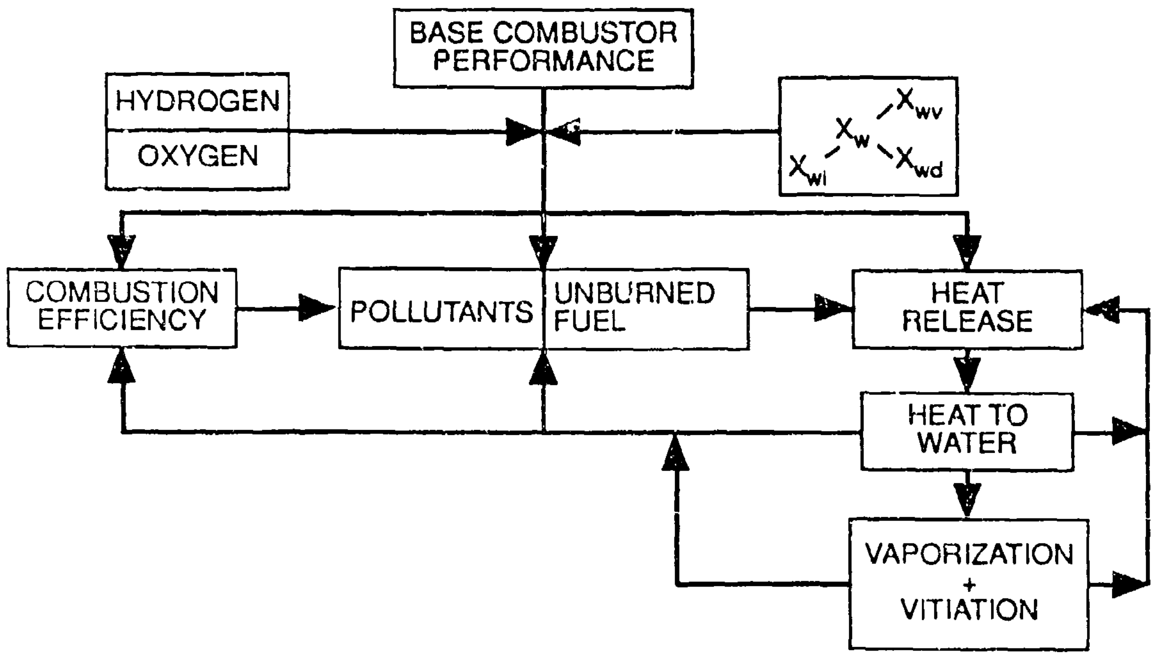


Figure 6. Interaction of Combustion Processes in the Combustor

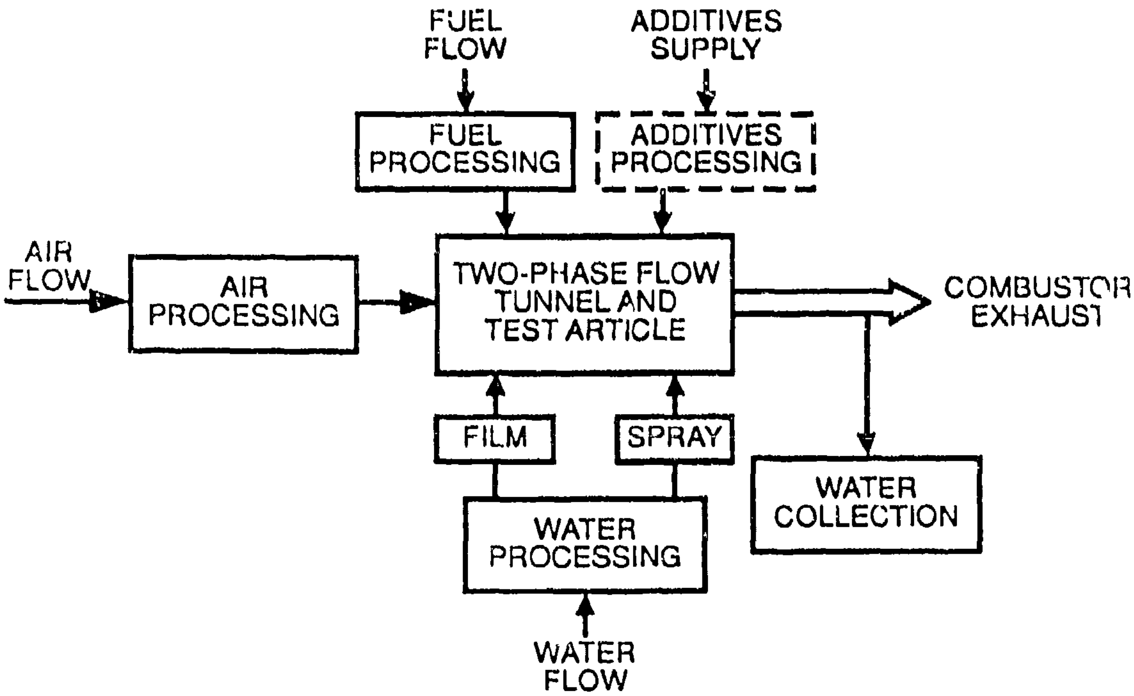
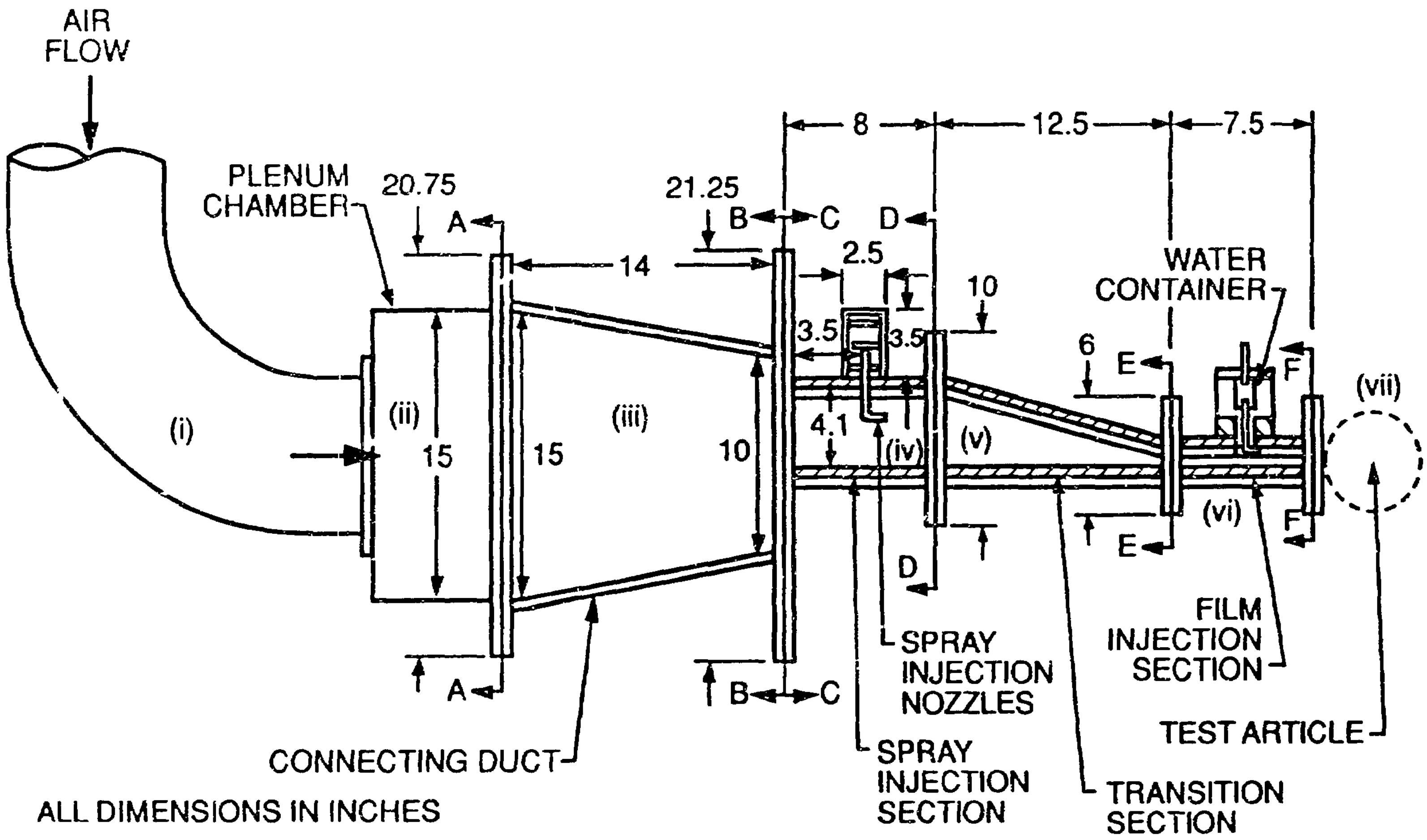


Figure 7. Schematic of the Overall Test Facility

Figure 8. Prediffuser-Combustor Test Article



(i) AIR FLOW DUCT
(ii) PLENUM CHAMBER
(iii) CONNECTING DUCT
(iv) SPRAY INJECTION SECTION
(v) TRANSITION SECTION
(vi) FILM INJECTION SECTION
(vii) TEST ARTICLE

Figure 9. Side View of the Two-Phase Flow Tunnel

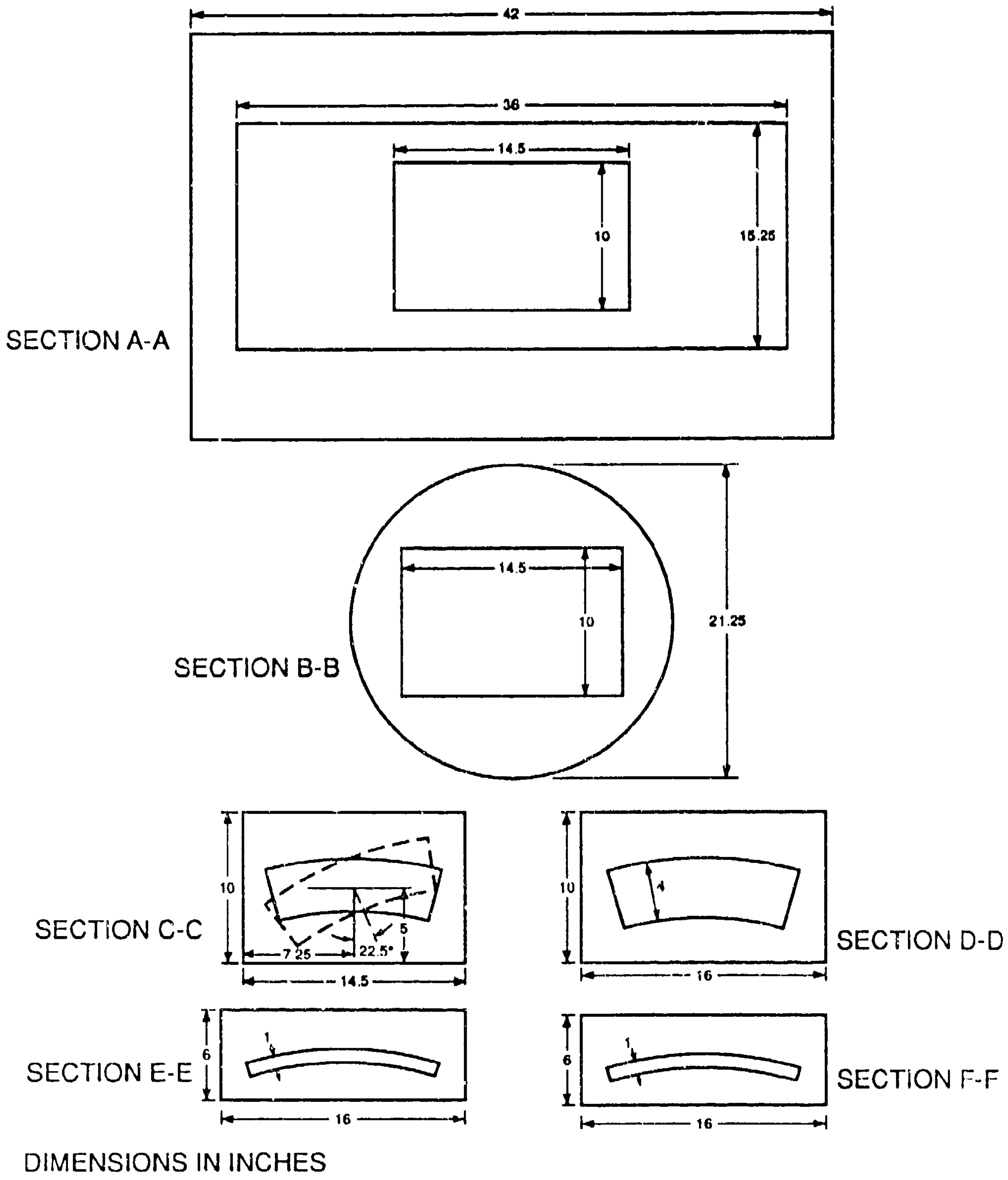


Figure 10. Cross Sections of the Two Phase Flow Tunnel

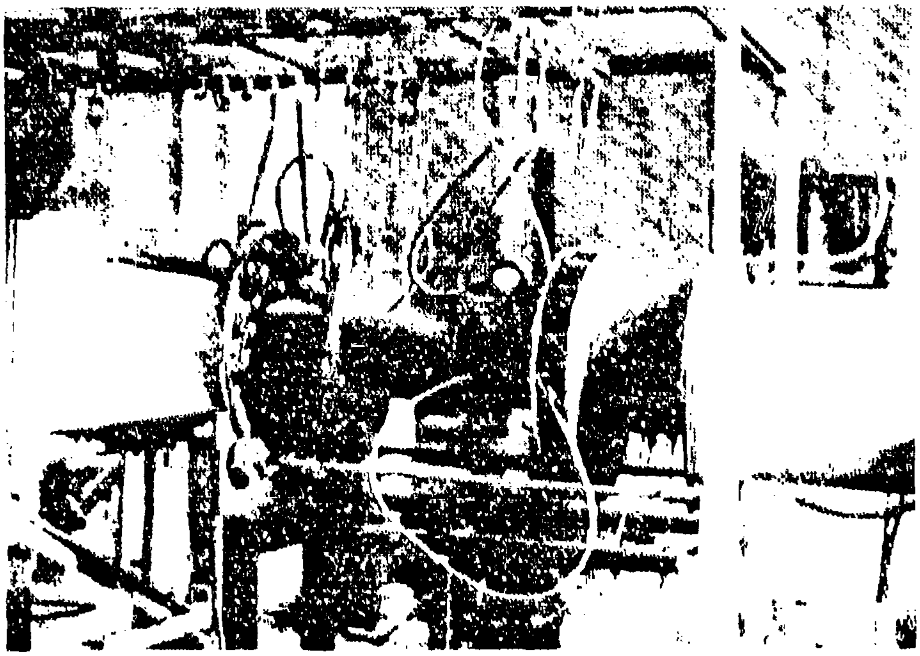


Figure 11. Two-Phase Flow Tunnel and Test Article Configured for Cold Flow Tests

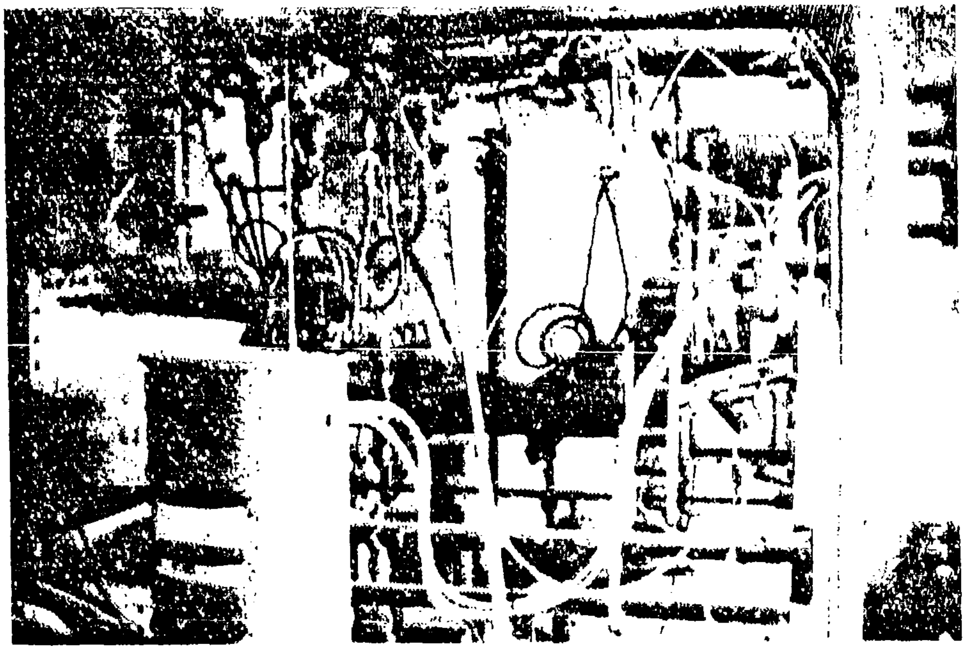


Figure 12. Two-Phase Flow Tunnel and Test Article Configured for Burning Tests

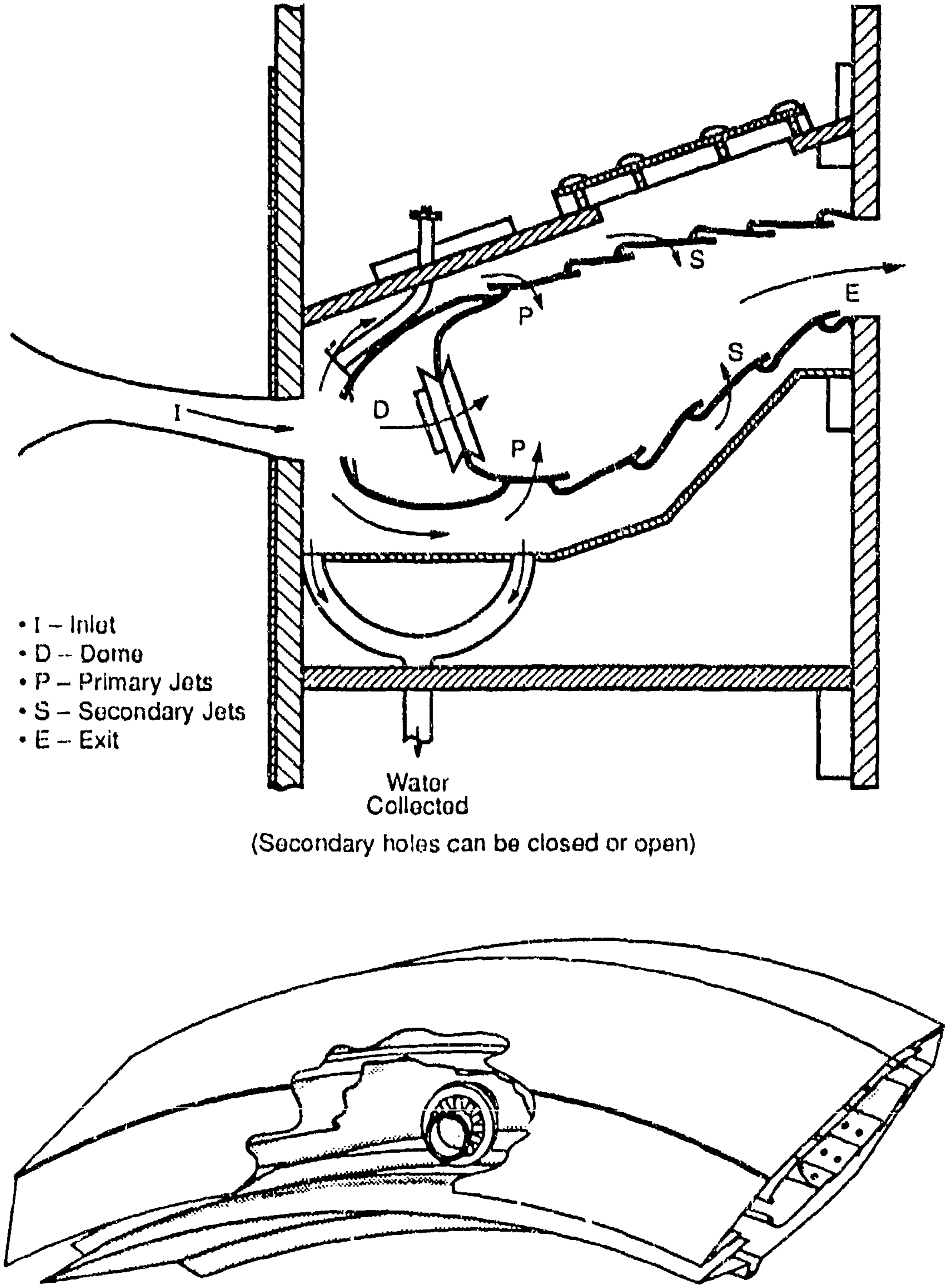


Figure 13. Combustor Test Article

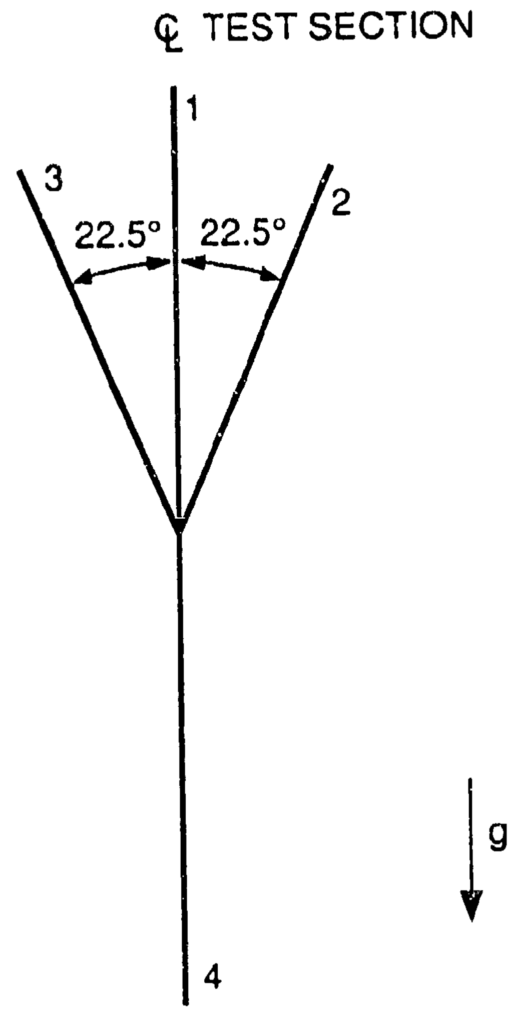
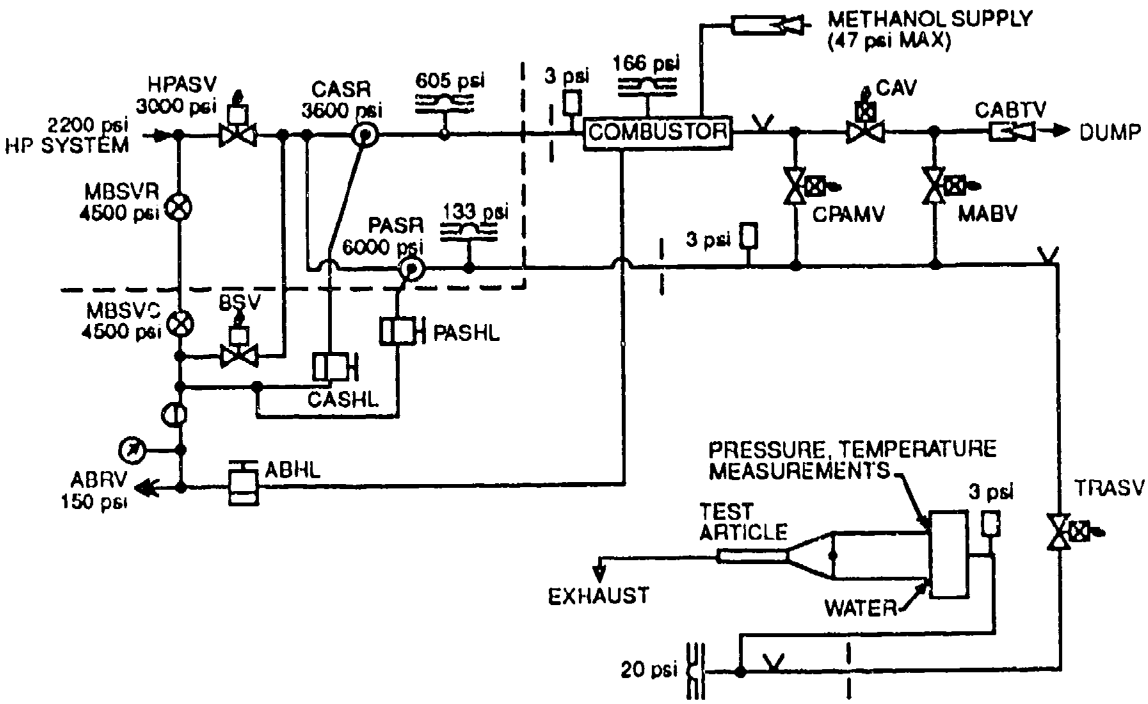


Figure 14. Test Article Orientation



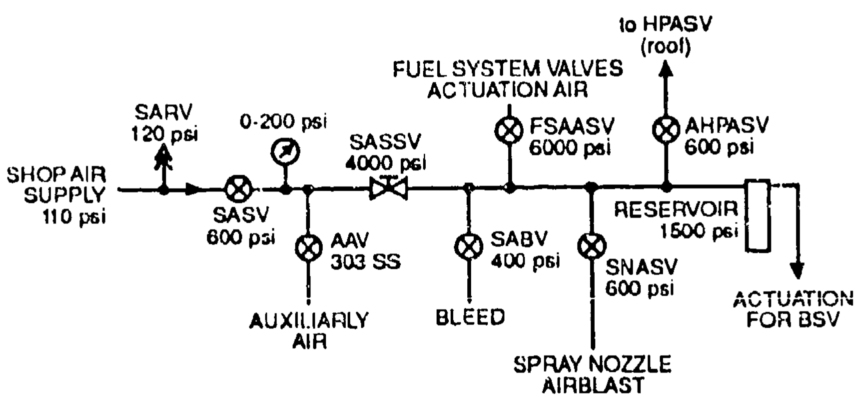


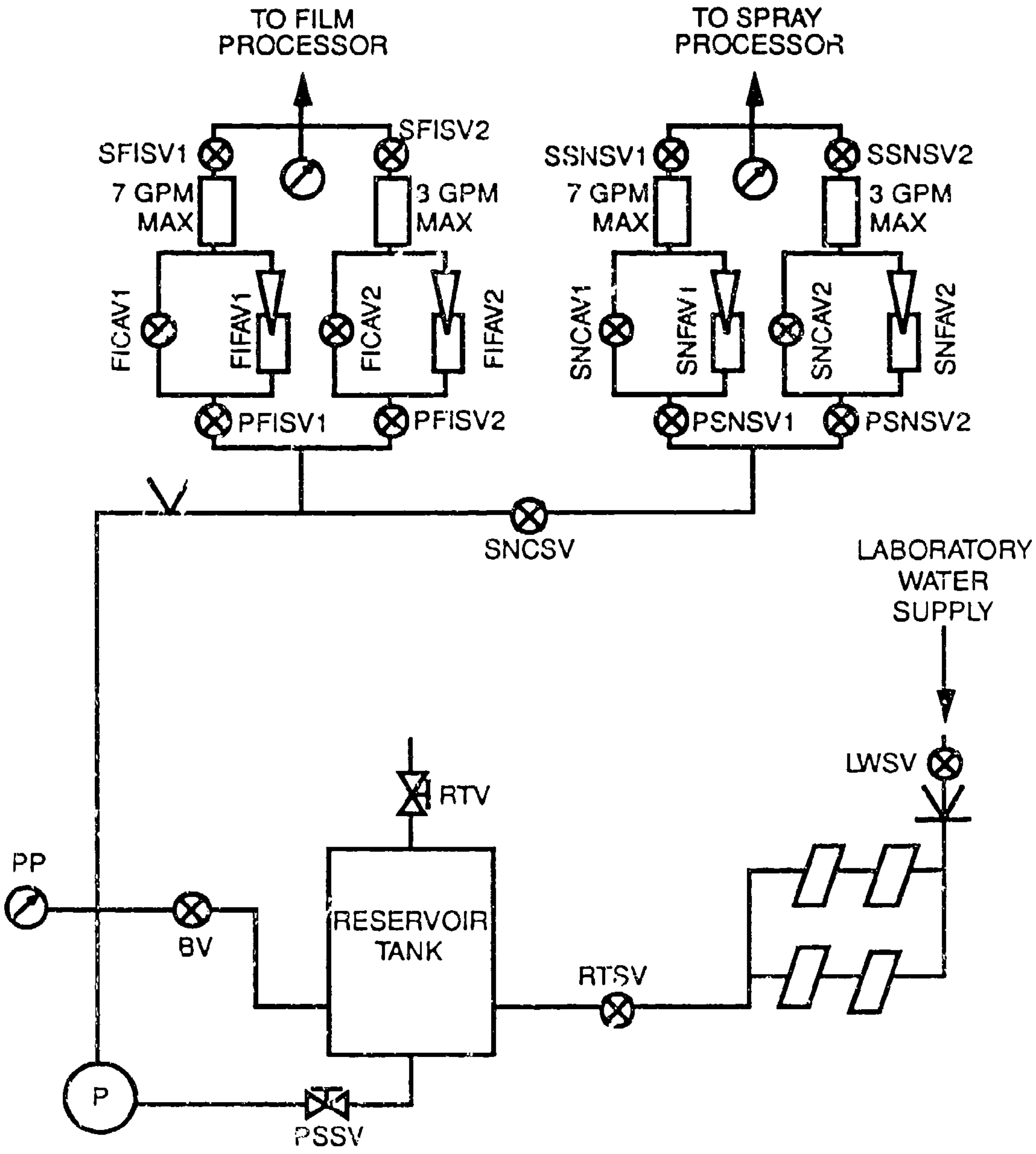
Figure 15. Laboratory Air Supply System (continued)

Nomenclature appears on following page.

- REGULATOR
- PRESSURE GAUGE
- MANUAL BALL VALVE
- MOTOR DRIVEN VALVE
- AIR-ACTUATED VALVE
- BURST DIAPHRAGM
- THERMOCOUPLE
- ORIFICE PLATE
- HAND LOADER
- AIR-ACTUATED REGULATOR
- MOTOR DRIVEN THROTTLE VALVE
- RELIEF VALVE
- PRESSURE SWITCH
- NEEDLE VALVE FLOWMETER

AAV	AUXILIARY AIR VALVE
ABHL	AIR BLAST HAND LOADER
CABTV	COMBUSTOR AIR BYPASS THROTTLE VALVE
CPAMV	COMBUSTION AIR MIXING VALVE
CASHL	COMBUSTOR AIR SUPPLY HAND LOADER
CASR	COMBUSTOR AIR SUPPLY REGULATOR
MABV	PRIMARY AIR MIXING VALVE
HPASV	HIGH PRESSURE AIR SHUTOFF VALVE
TRASV	TEST RIG AIR SHUTOFF VALVE
ABRV	AIR BLAST RELIEF VALVE
CAV	COMBUSTOR AIR VALVE
FSAASV	FUEL SYSTEM ACTUATION AIR SHUTOFF VALVE
BSV	BYPASS SHUTOFF VALVE
PASHL	PRIMARY AIR SUPPLY HAND LOADER
PASR	PRIMARY AIR SUPPLY REGULATOR
SASV	SHOP AIR SHUTOFF VALVE
SABV	SHOP AIR BLEED VALVE
MBSVR	MANUAL BYPASS SHUTOFF VALVE--ROOF
SASSV	SHOP AIR SECONDARY SHUTOFF VALVE
MBSVC	MANUAL BYPASS SHUTOFF VALVE--CELL
AHPASV	ACTUATES HPASV
SARV	SHOP AIR RELIEF VALVE
SNASV	SPRAY NOZZLE AIR BLAST SHUTOFF VALVE

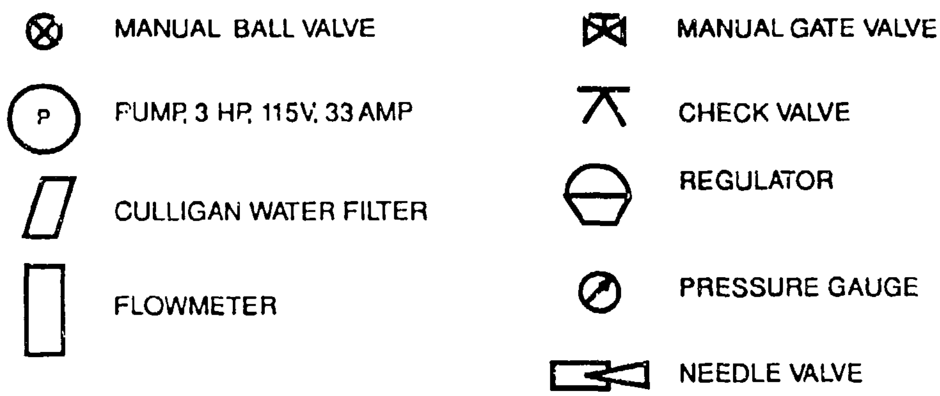
Figure 16. Laboratory Air Supply System (concluded)



NOTE: ALL PIPING IS PVC 1120 1" DIA. RATED AT 450 psi@ 78F

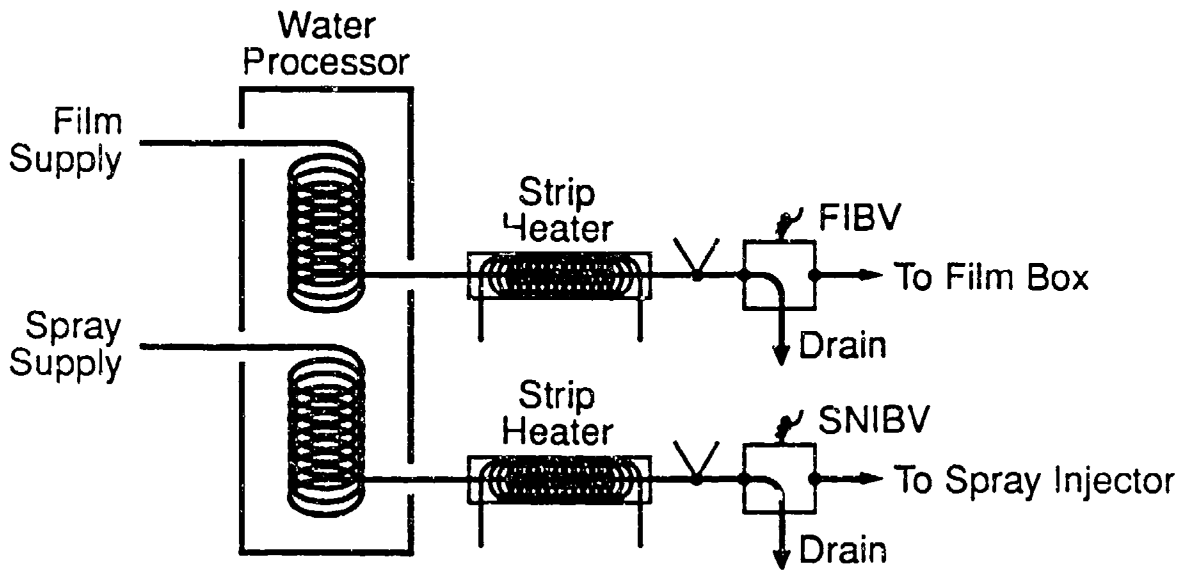
Figure 17. Water Supply System (continued)

Nomenclature appears on following page.



FICA	FILM INJECTOR COARXE ADJUSTMENT
FIFAV	FILM INJECTOR FINE ADJUSTMENT VALVE
LWSV	LABORATORY WATER SHUTOFF VALVE
BV	BLEED VALVE
PFISV	PRIMARY FILM INJECTOR SHUTOFF VALVE
PSNSV	PRIMARY SPRAY NOZZLE SHUTOFF VALVE
PSSV	PUMP SUPPLY SHUTOFF VALVE
RTDV	RESERVOIR TANK DRAIN VALVE
RTSV	RESERVOIR TANK SUPPLY VALVE
RTV	RESERVOIR TANK VENT
SFISV	SECONDARY FILM INJECTOR SHUTOFF VALVE
SNCAV	SPRAY NOZZLE COARSE ADJUSTMENT VALVE
SNCSV	SPRAY NOZZLE CABINET SHUTOFF VALVE
SNFAV	SPRAY NOZZLE FINE ADJUSTMENT VALVE
SSNSV	SECONDARY SPRAY NOZZLE SHUTOFF VALVE
PP	PUMP SUPPLY PRESSURE

Figure 18. Water Supply System (concluded)



FIBV – Film Inject-Bypass Valve
SNIBV – Spray Nozzle Inject-Bypass Valve

Figure 19. Water Processing System

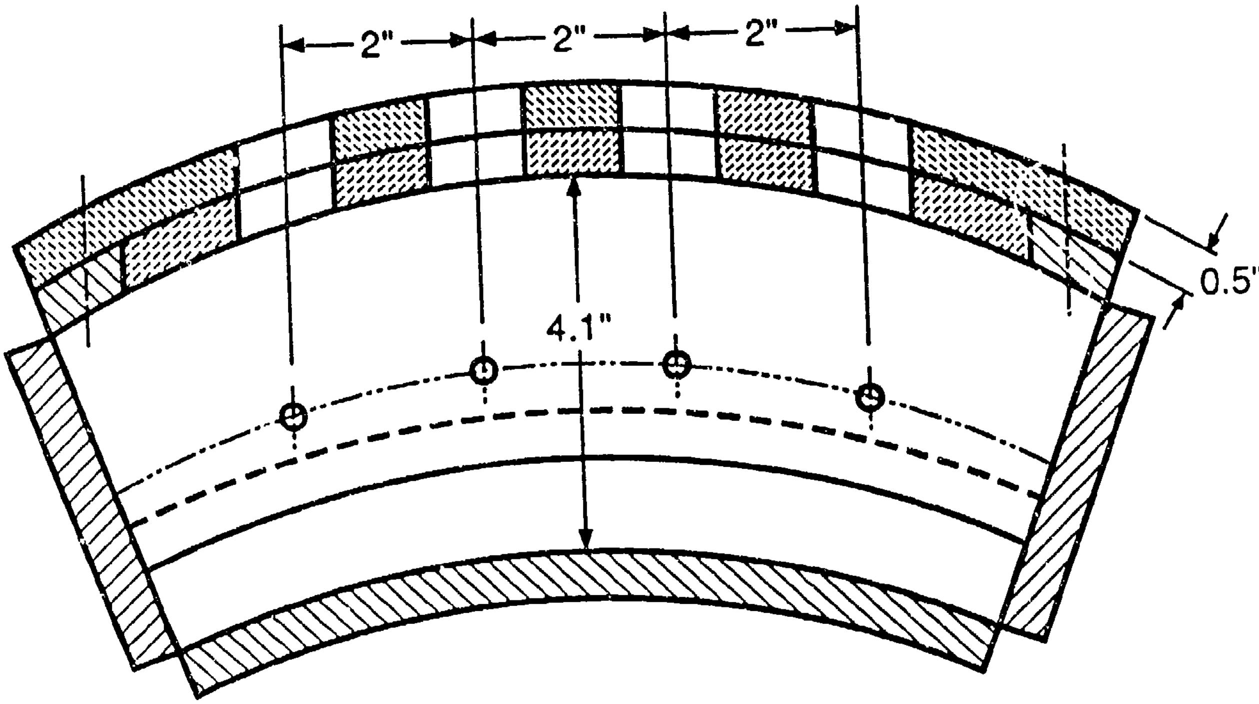


Figure 20. Water Spray Injection Manifold

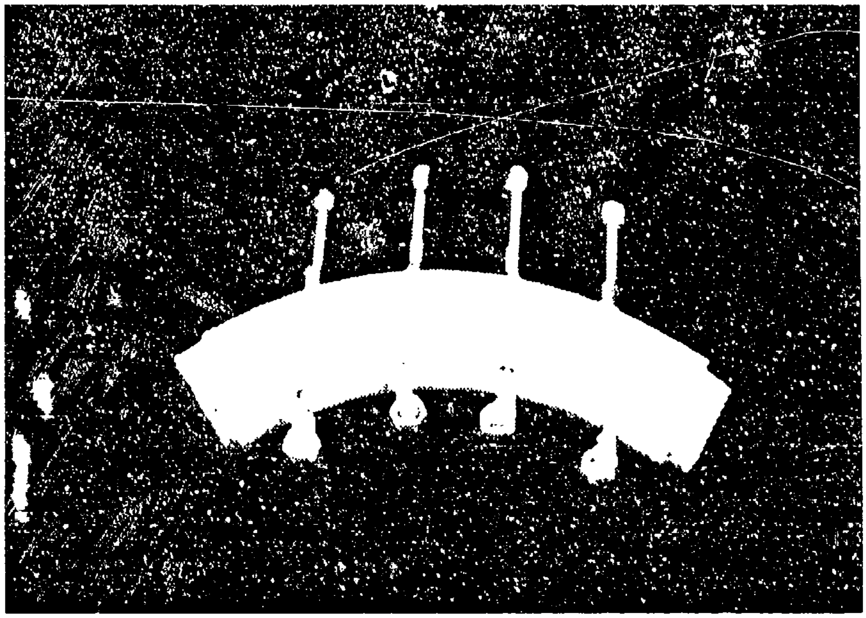


Figure 21. Large Droplet Spray Injection Manifold

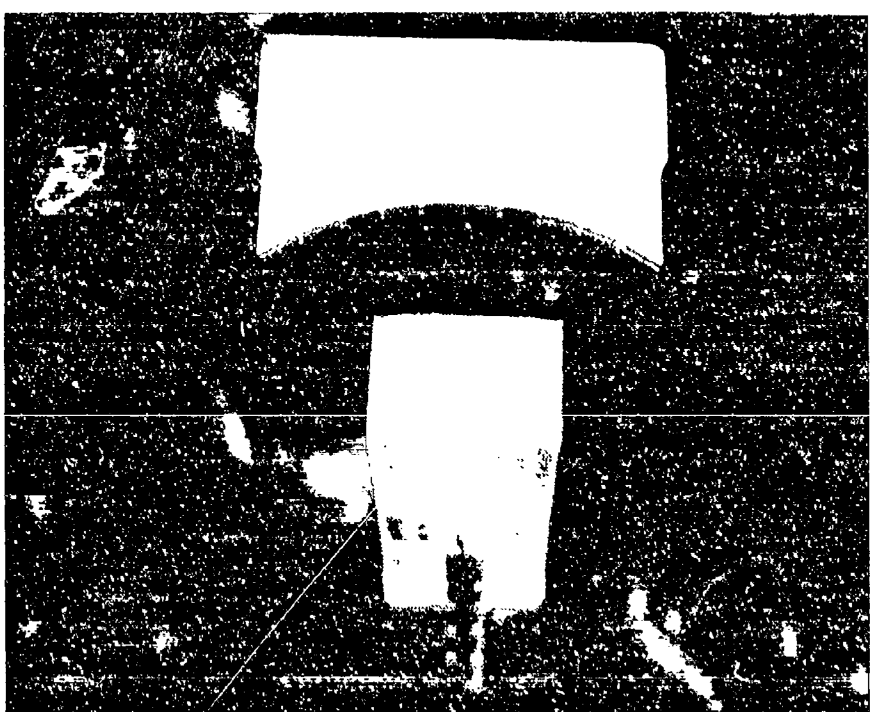


Figure 22. Water Film Injection Box

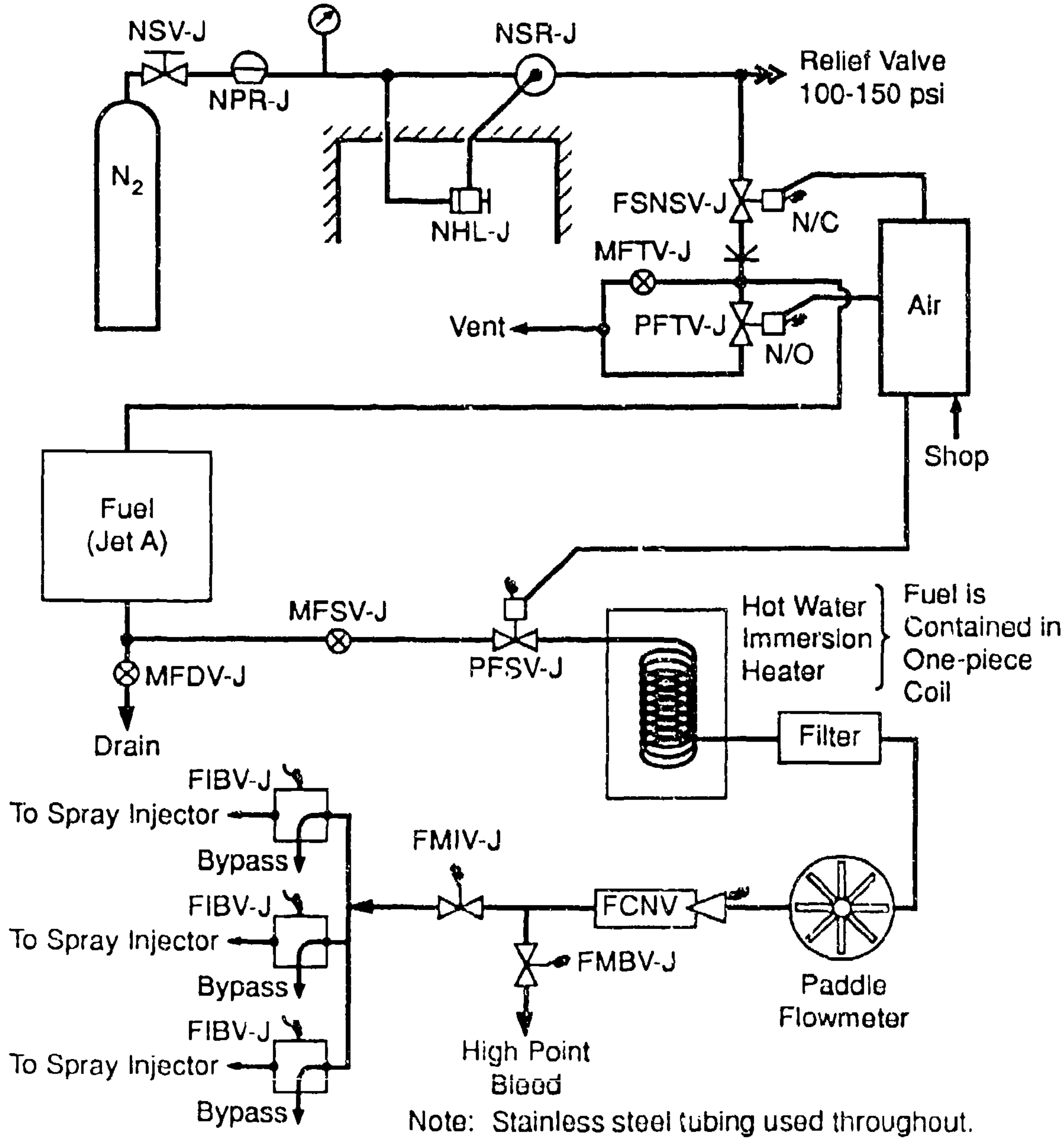


Figure 23. Fuel Supply and Processing Systems (continued)

Nomenclature appears on following page.

(-J)	⇒	Jet A fuel supply
NPR-J	–	Nitrogen pressure regulator
NSV-J	–	Nitrogen shut-off valve
NSR-J	–	Nitrogen supply regulator
NHL-J	–	Nitrogen hand loader
FSNSV-J	–	Fuel system nitrogen shutoff valve
MFTV-J	–	Manual fuel tank vent
PFTV-J	–	Pneumatic fuel tank vent
MFDV-J	–	Manual fuel drain valve
MFSV-J	–	Manual fuel shutoff valve
PFSV-J	–	Pneumatic fuel shutoff valve
FMBV-J	–	Fuel manifold bleed valve
FMIV-J	–	Fuel manifold injector valve
FCNV-J	–	Fuel control needle valve
FIBV-J	–	Fuel inject/bypass valves

Figure 24. Fuel Supply and Processing Systems (concluded)

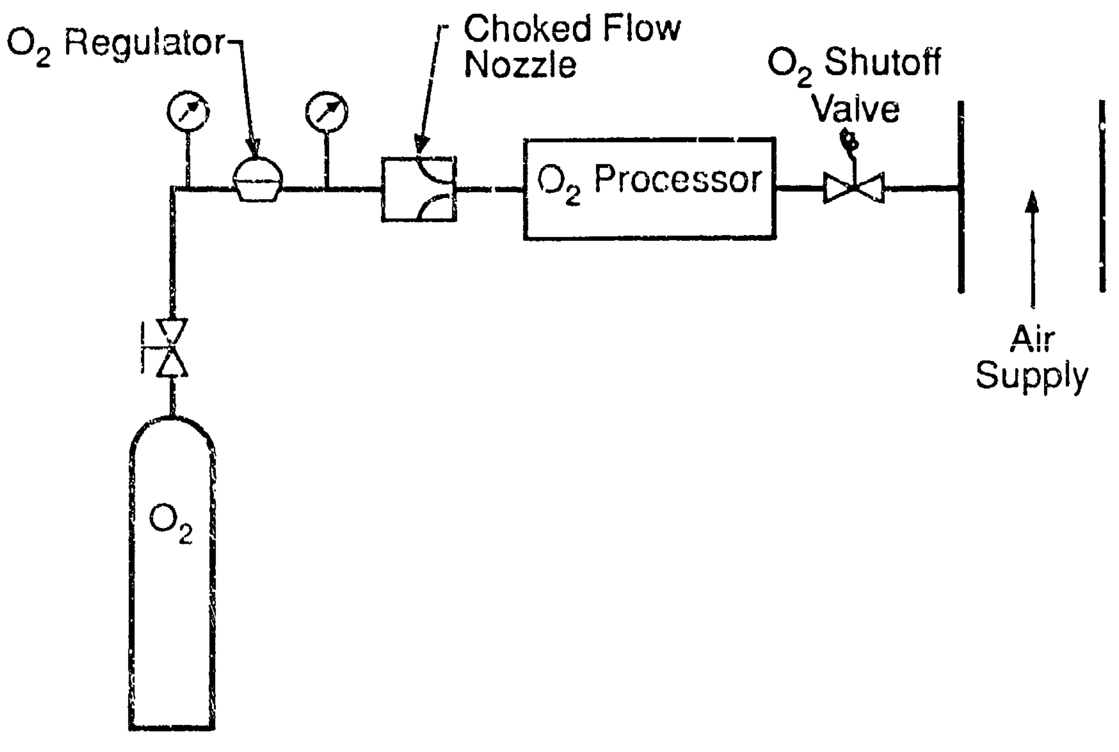


Figure 25. Oxygen Supply System

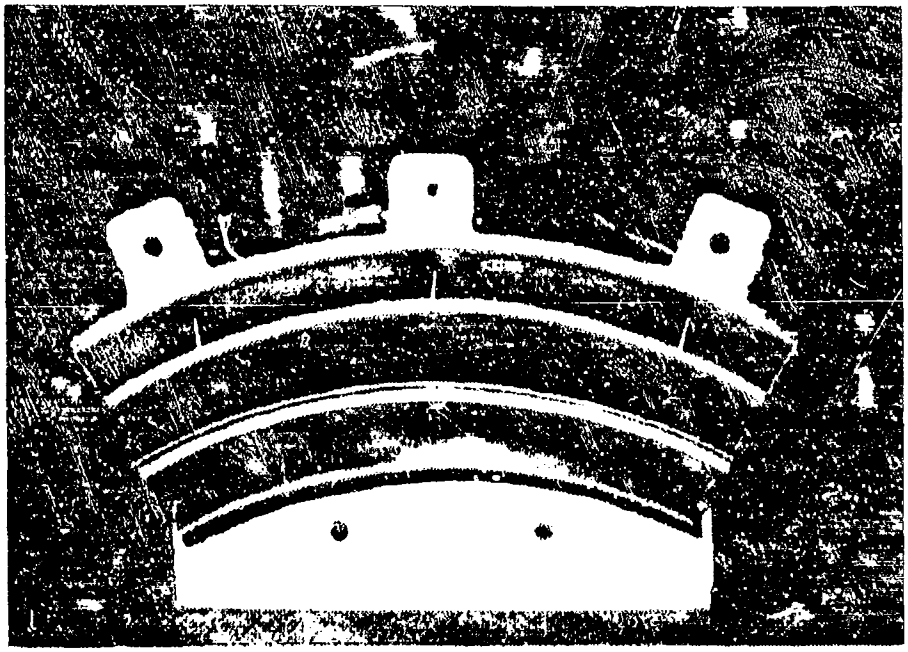


Figure 26. Prediffuser Section

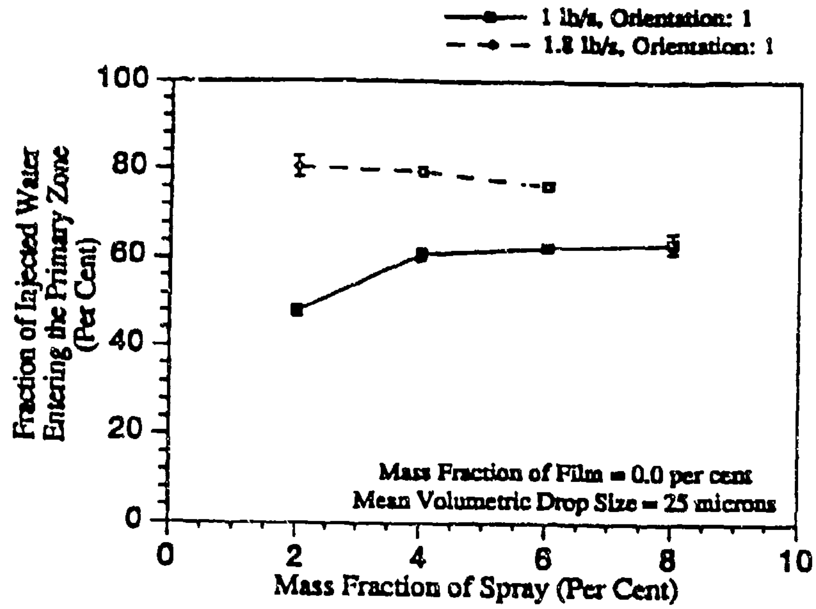


Figure 27. Case I: Effect of Mass Fraction of Injected Spray of Small Droplet Size on Water Entry into the Primary Zone Through the Dome and Primary Jets (Orientation 1)

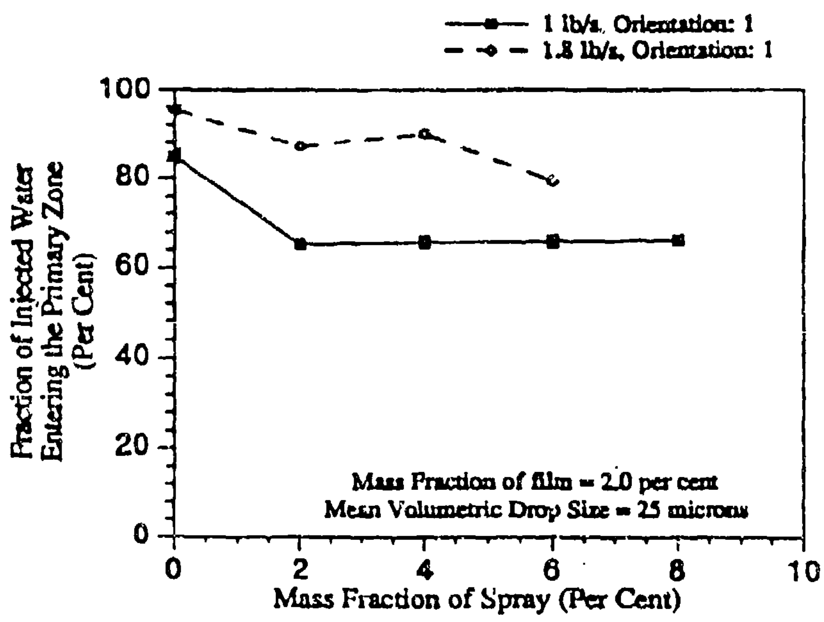


Figure 28. Case I: Effect of Mass Fraction of Injected Film and Spray of Small Droplet Size on Water Entry into the Primary Zone Through the Dome and Primary Jets (Orientation 1)

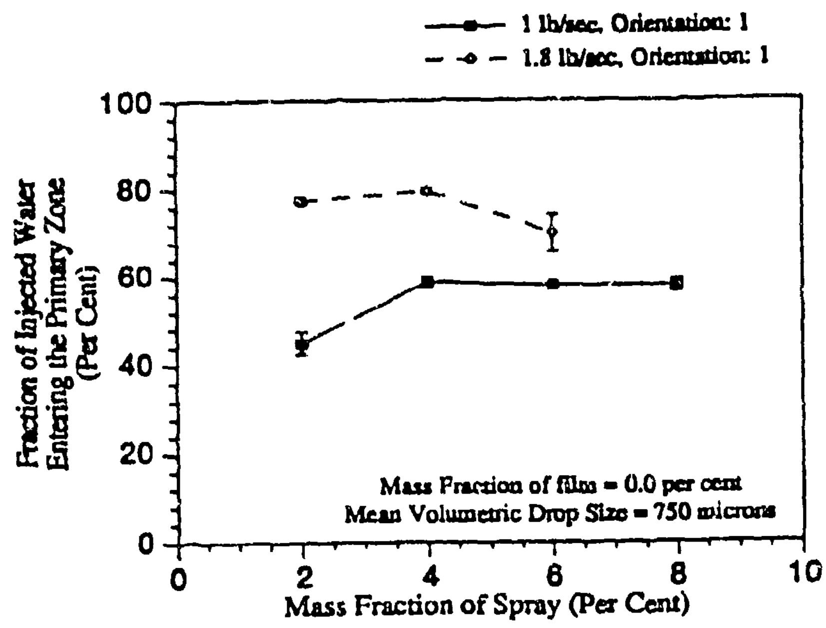


Figure 29. Case II: Effect of Mass Fraction of Injected Spray of Large Droplet Size on Water Entry into the Primary Zone Through the Dome and Primary Jets (Orientation 1)

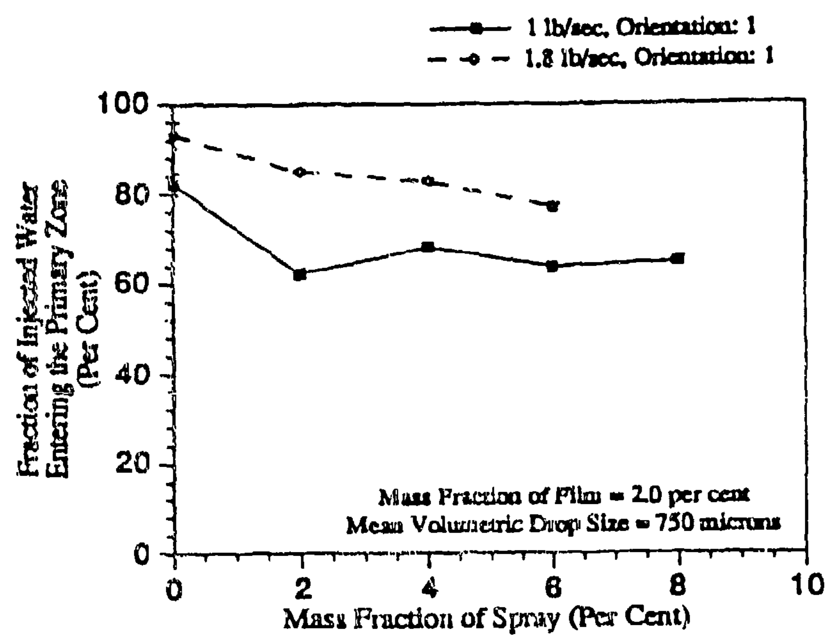


Figure 30. Case II: Effect of Mass Fraction of Injected Film and Spray of Large Droplet Size on Water Entry into the Primary Zone Through the Dome and Primary Jets (Orientation 1)

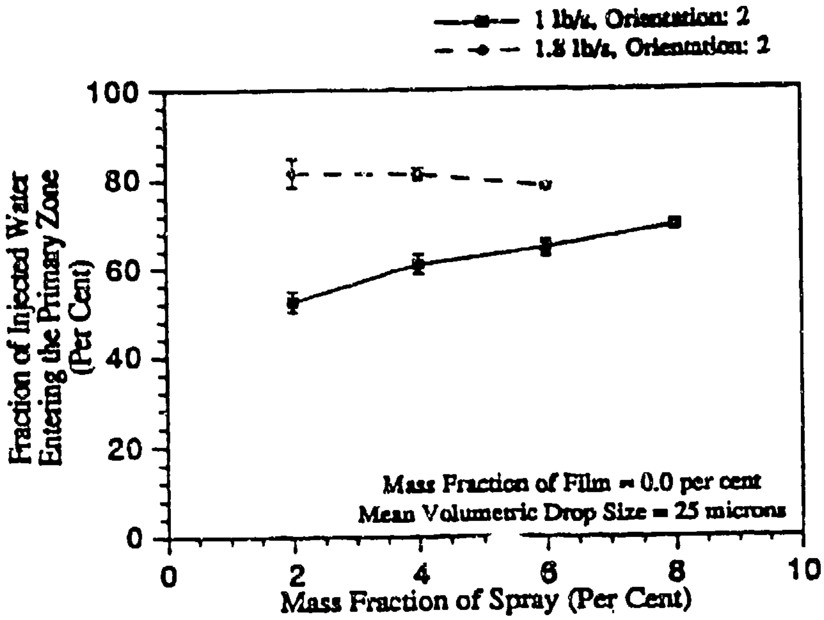


Figure 31. Case III: Effect of Mass Fraction of Injected Spray of Small Droplet Size on Water Entry into the Primary Zone Through the Dome and Primary Jets (Orientation 2)

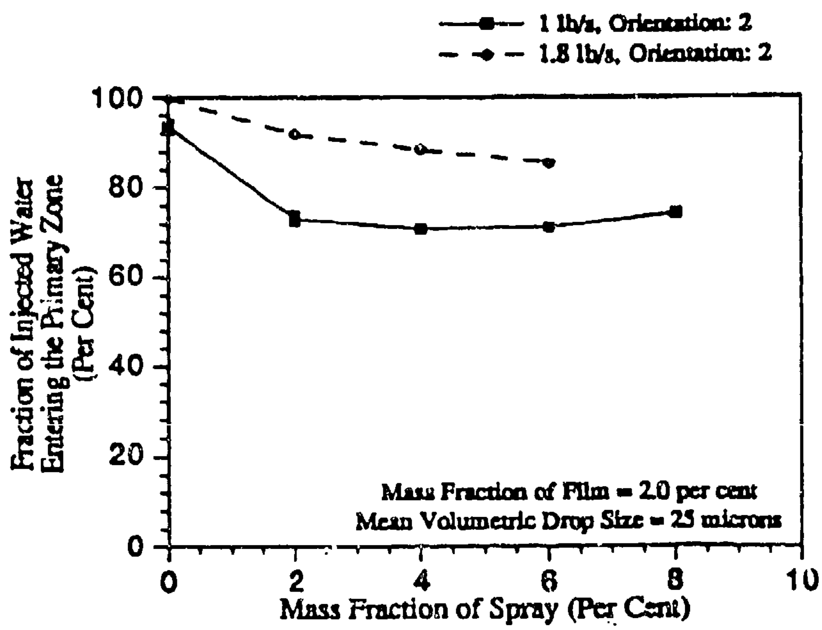


Figure 32. Case III: Effect of Mass Fraction of Injected Film and Spray of Small Droplet Size on Water Entry into the Primary Zone Through the Dome and Primary Jets (Orientation 2)

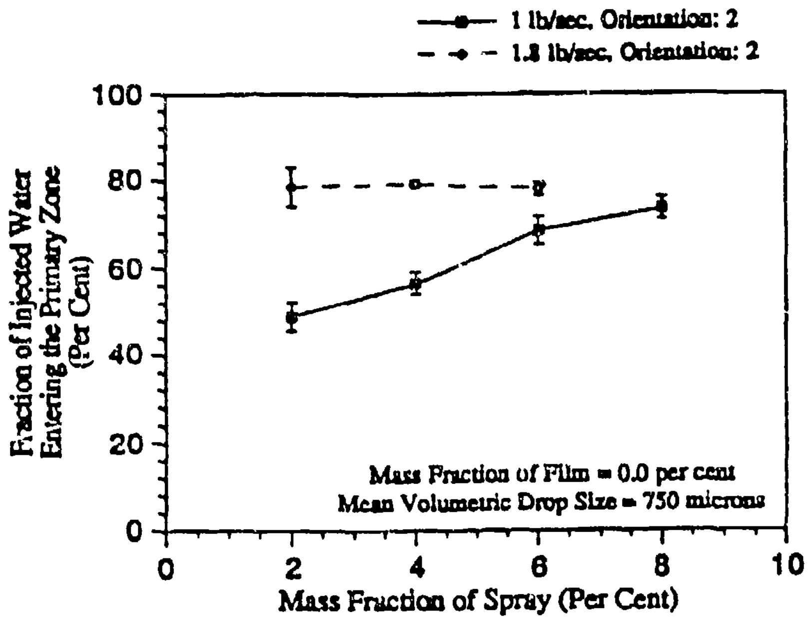


Figure 33. Case IV: Effect of Mass Fraction of Injected Spray of Large Droplet Size on Water Entry into the Primary Zone Through the Dome and Primary Jets (Orientation 2)

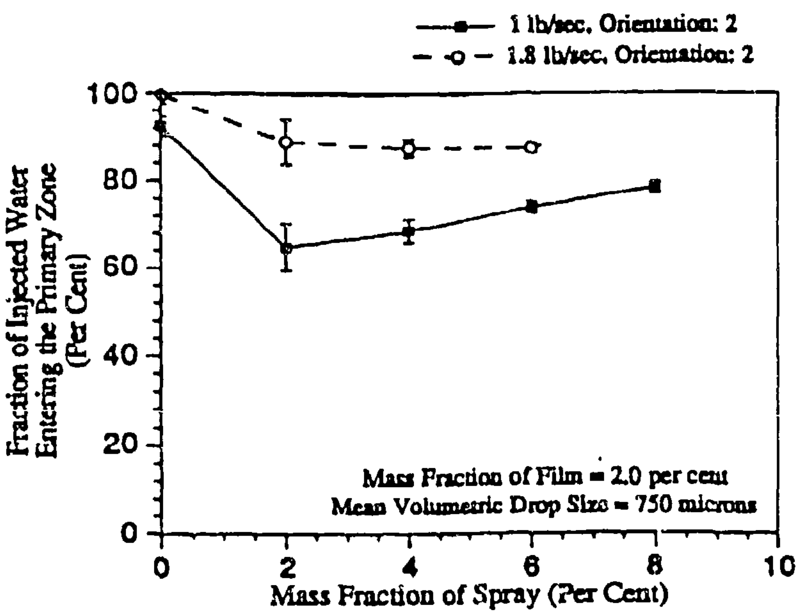


Figure 34. Case IV: Effect of Mass Fraction of Injected Film and Spray of Large Droplet Size on Water Entry into the Primary Zone Through the Dome and Primary Jets (Orientation 2)

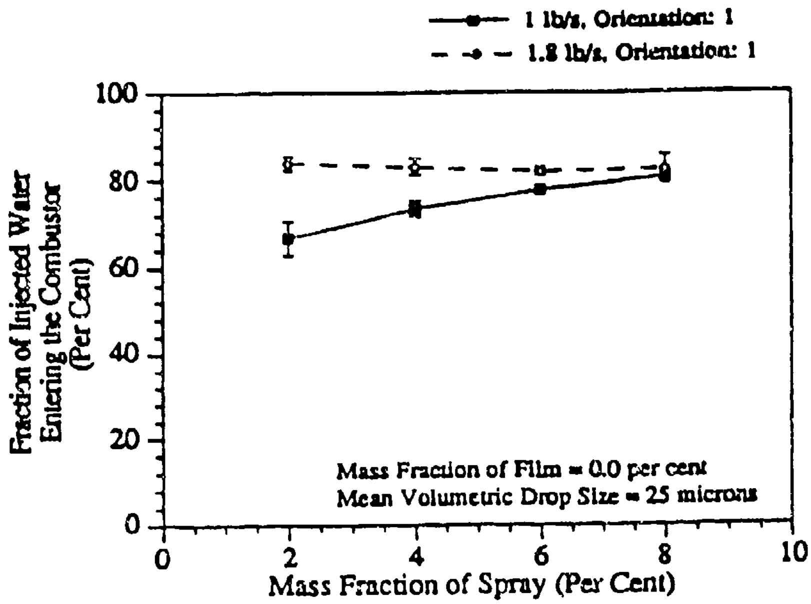


Figure 35. Case V: Effect of Mass Fraction of Injected Spray of Small Droplet Size on Water Entry into the Combustor (Orientation 1)

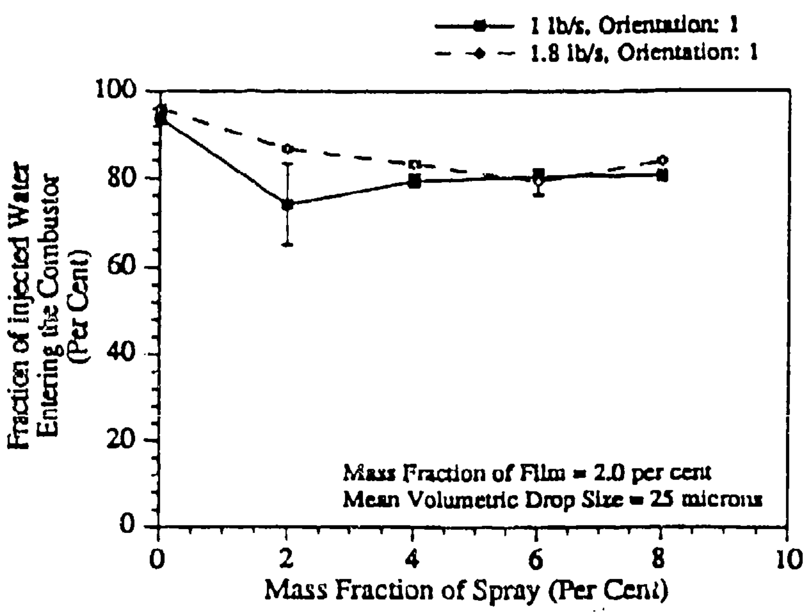


Figure 36. Case V: Effect of Mass Fraction of Injected Film and Spray of Small Droplet Size on Water Entry into the Combustor (Orientation 1)

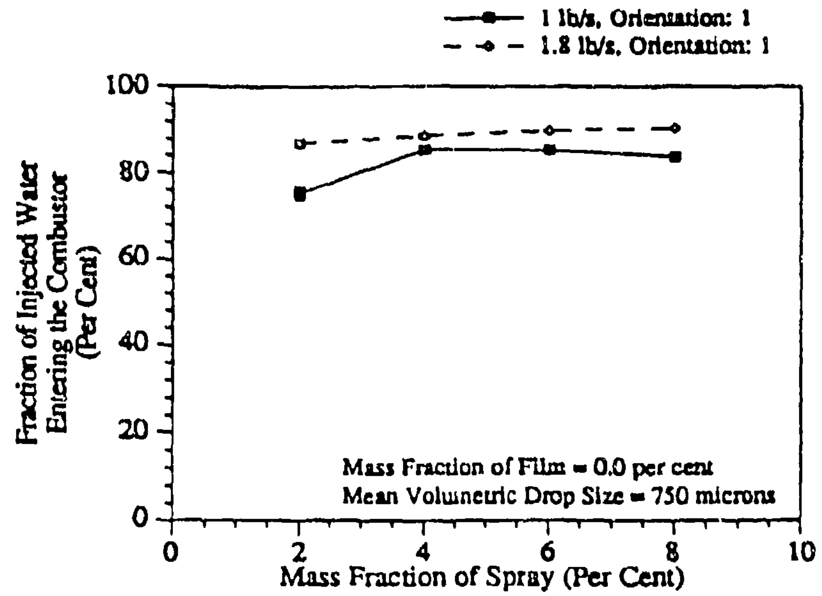


Figure 37. Case VI: Effect of Mass Fraction of Injected Spray of Large Droplet Size on Water Entry into the Combustor (Orientation 1)

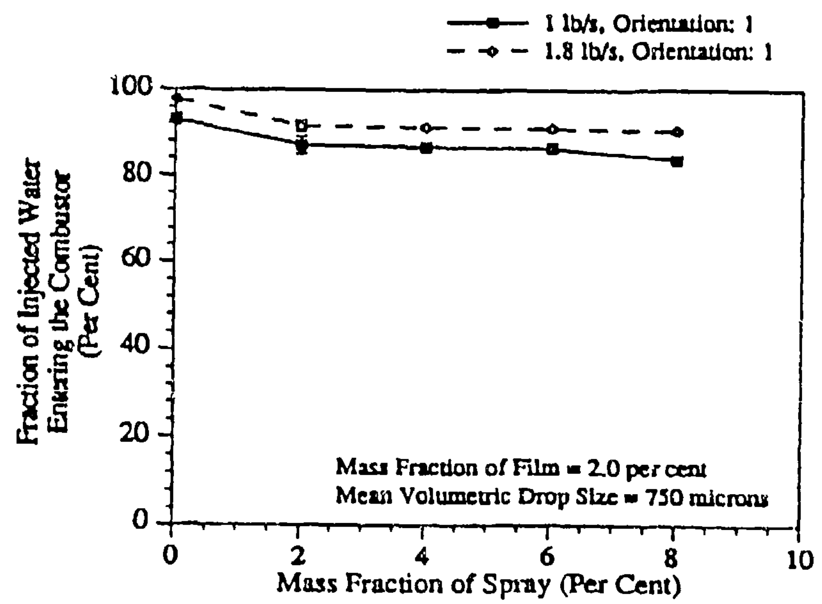


Figure 38. Case VI: Effect of Mass Fraction of Injected Film and Spray of Large Droplet Size on Water Entry into the Combustor (Orientation 1)

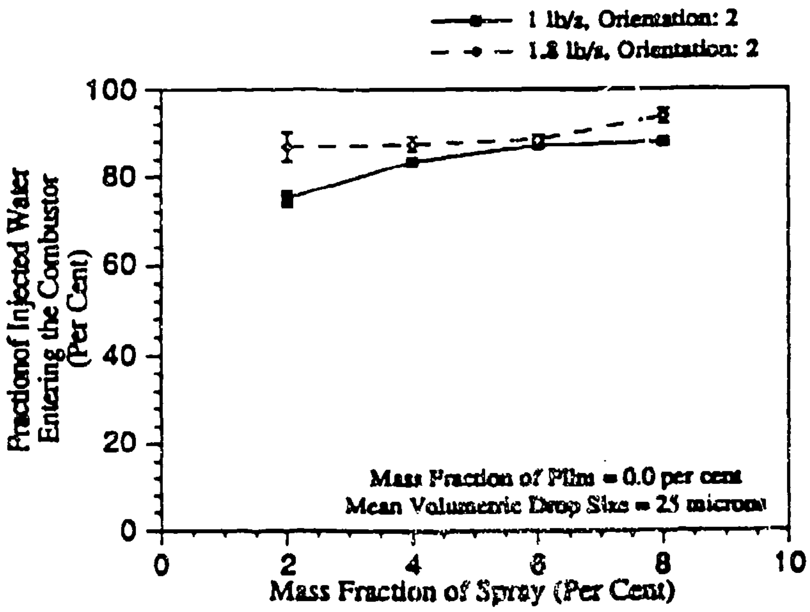


Figure 39. Case VII: Effect of Mass Fraction of Injected Spray of Small Droplet Size on Water Entry into the Combustor (Orientation 2)

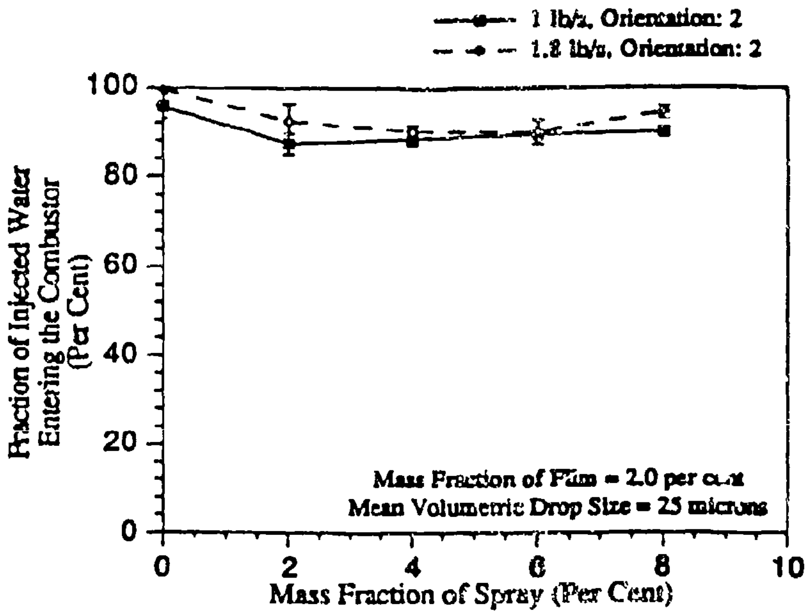


Figure 40. Case VII: Effect of Mass Fraction of Injected Film and Spray of Small Droplet Size on Water Entry into the Combustor (Orientation 2)

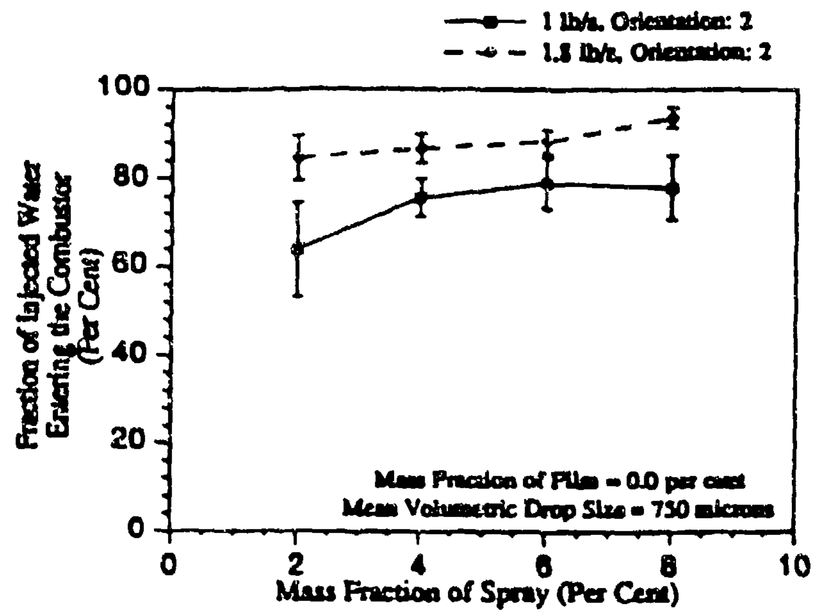


Figure 41. Case VIII. Effect of Mass Fraction of Injected Spray of Large Droplet Size on Water Entry into the Combustor (Orientation 2)

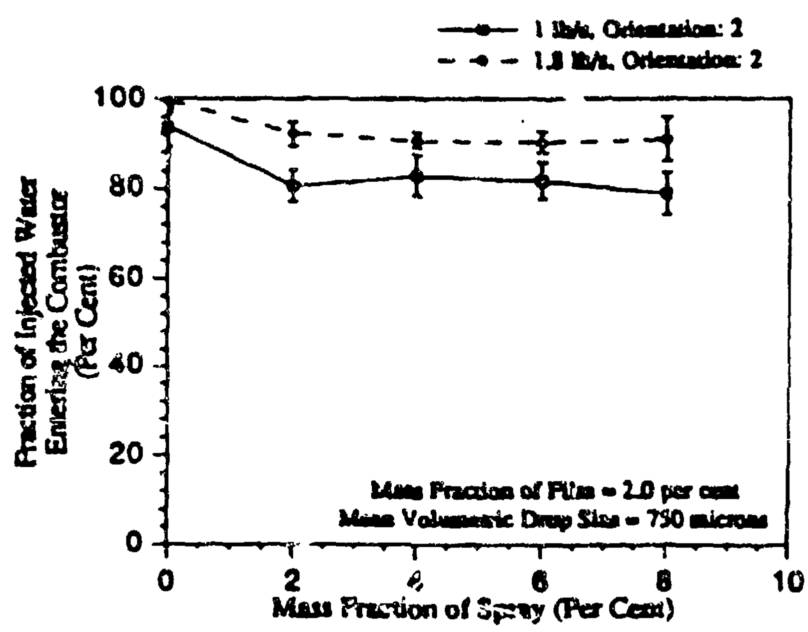


Figure 42. Case VIII. Effect of Mass Fraction of Injected Film and Spray of Large Droplet Size on Water Entry into the Combustor (Orientation 2)

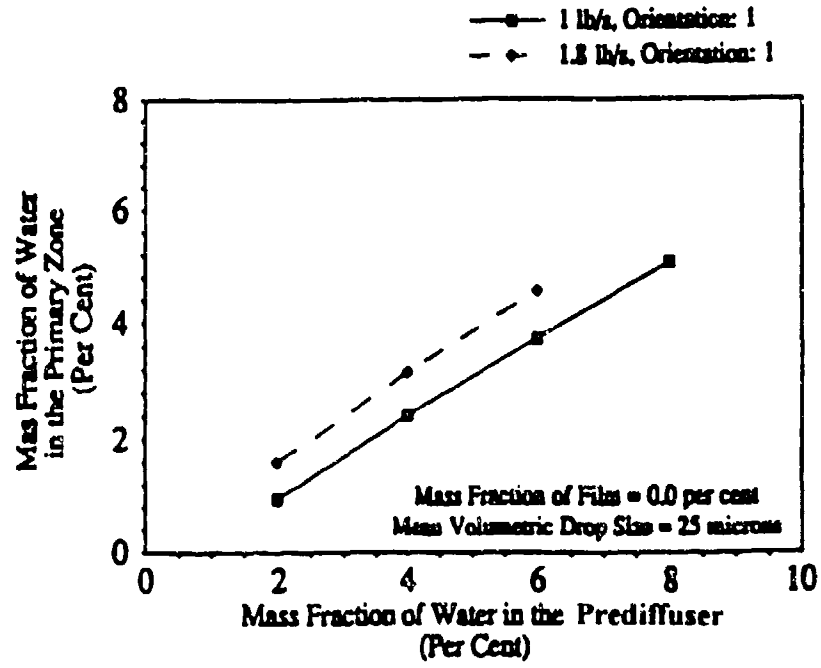


Figure 43. Case I: Effect of Mass Fraction of Injected Spray in the Prediffuser on Mass Fraction of Water in the Primary Zone

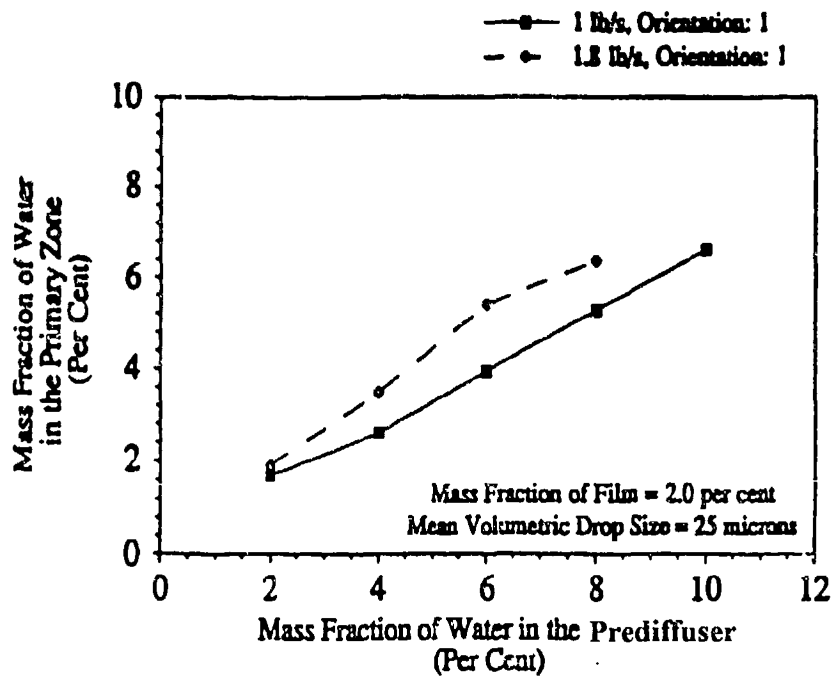


Figure 44. Case I: Effect of Mass Fraction of Injected Film and Spray in the Prediffuser on Mass Fraction of Water in the Primary Zone

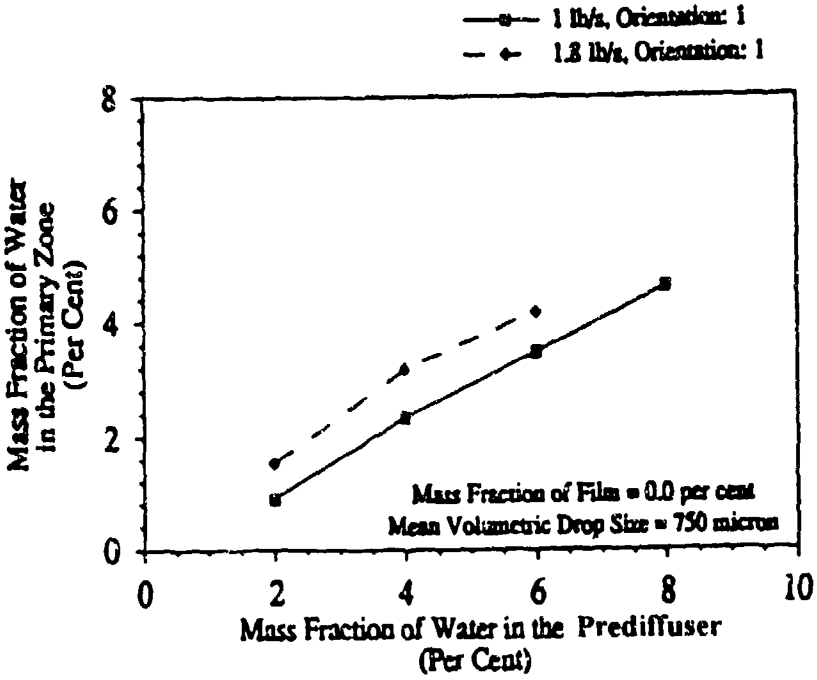


Figure 45. Case II: Effect of Mass Fraction of Injected Spray in the Prediffuser on Mass Fraction of Water in the Primary Zone

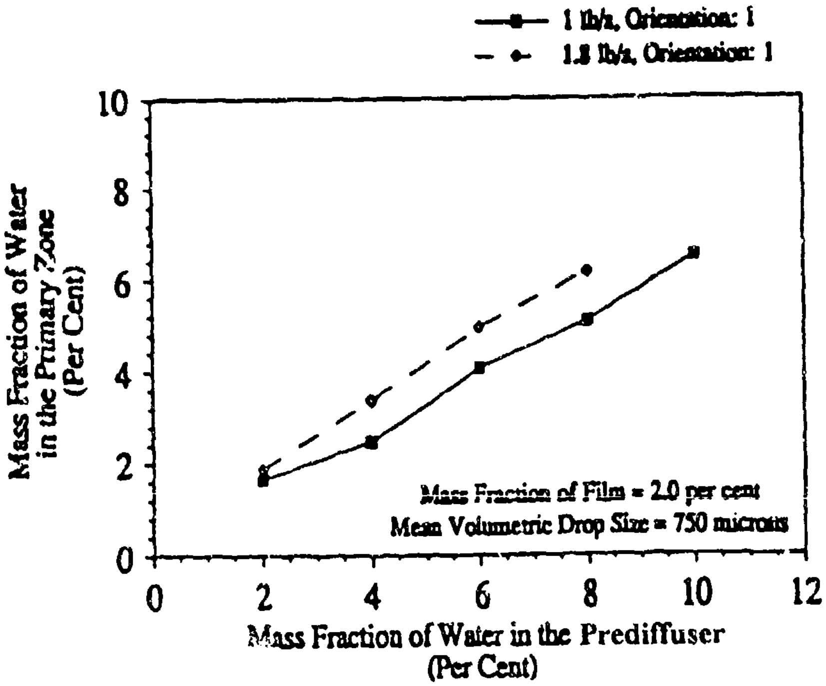


Figure 46. Case II: Effect of Mass Fraction of Injected Film and Spray in the Prediffuser on Mass Fraction of Water in the Primary Zone

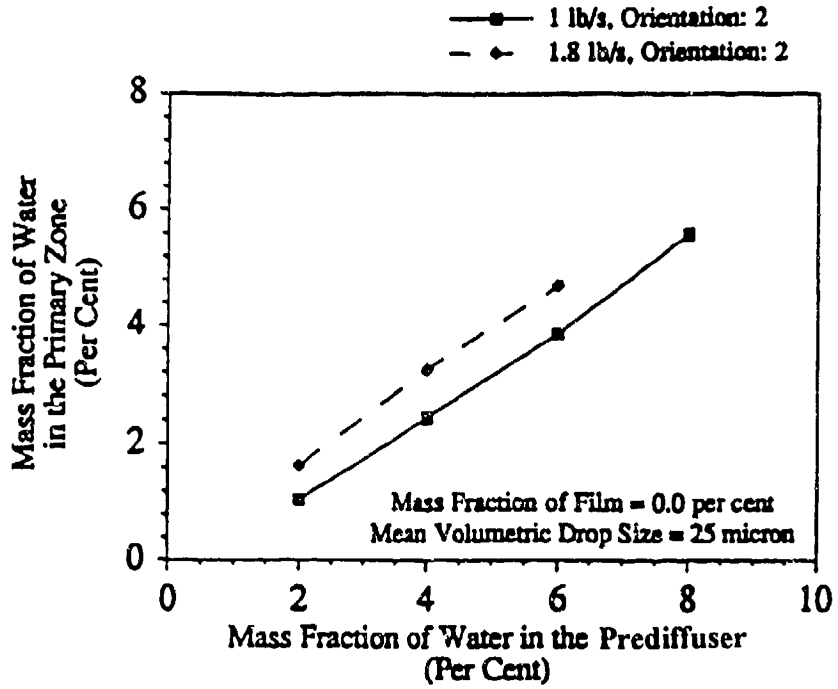


Figure 47. Case III: Effect of Mass Fraction of Injected Spray in the Prediffuser on Mass Fraction of Water in the Primary Zone

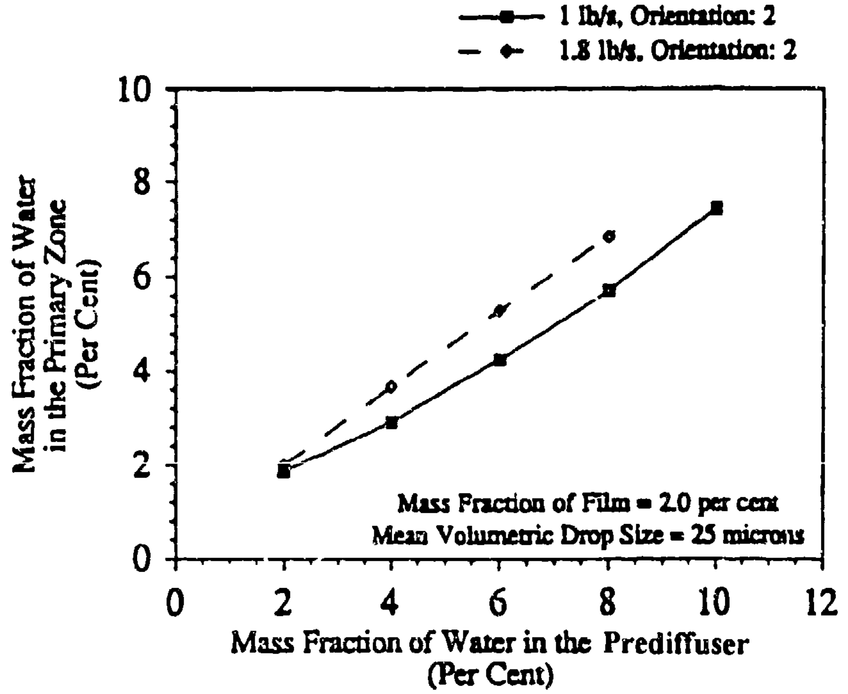


Figure 48. Case III: Effect of Mass Fraction of Injected Film and Spray in the Prediffuser on Mass Fraction of Water in the Primary Zone

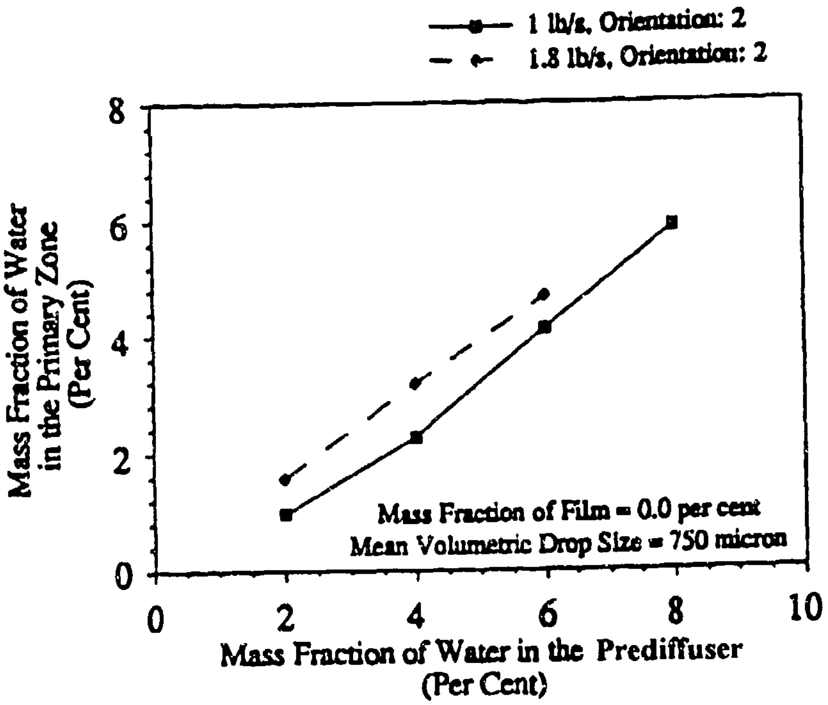


Figure 49. Case IV: Effect of Mass Fraction of Injected Spray in the Prediffuser on Mass Fraction of Water in the Primary Zone

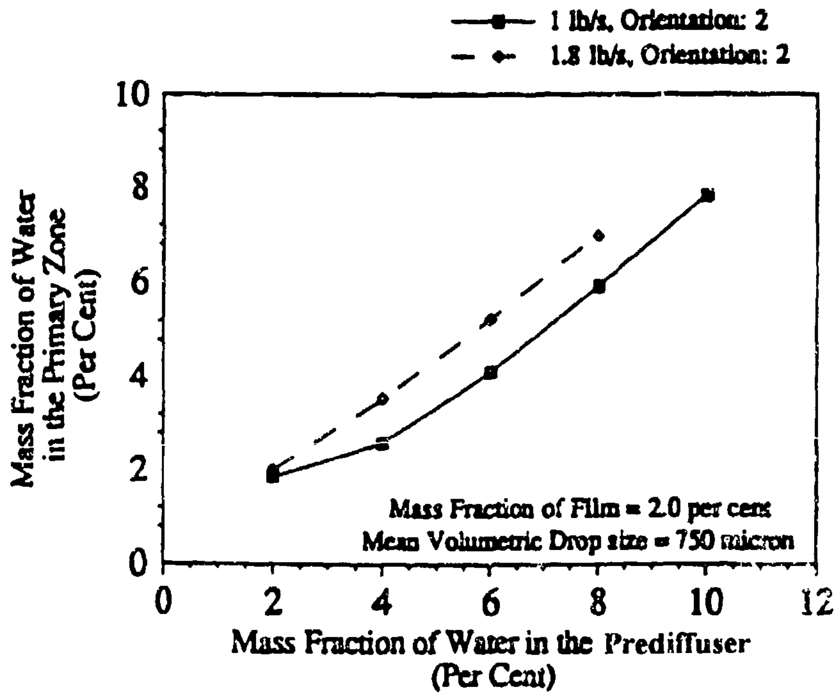


Figure 50. Case IV: Effect of Mass Fraction of Injected Film and Spray in the Prediffuser on Mass Fraction of Water in the Primary Zone

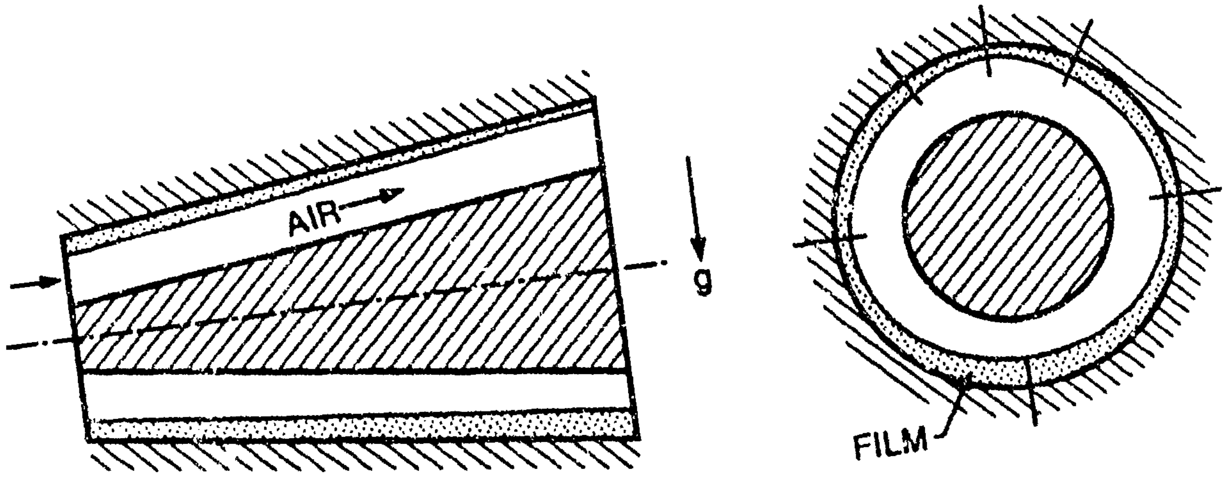


Figure 51. Water Fil Flow in a Prediffuser

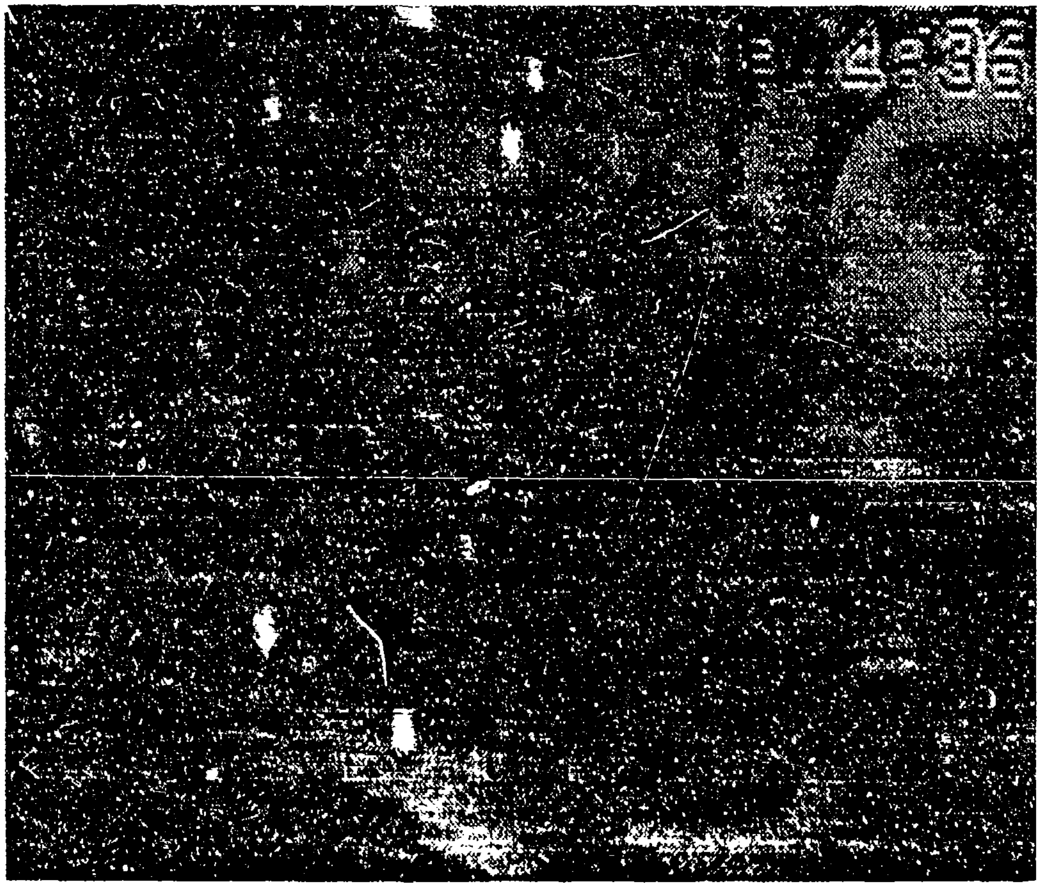


Figure 52. Water Droplet Distribution in the Primary Zone

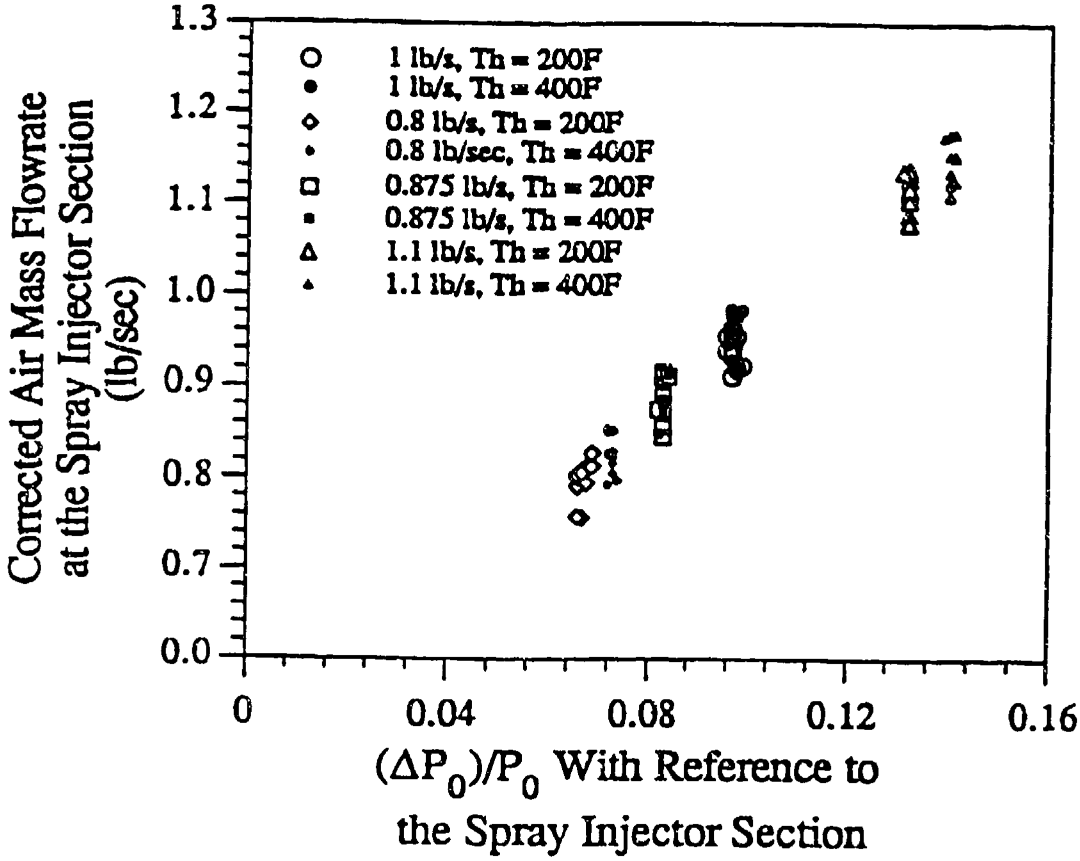


Figure 53. Pressure Loss Across the Combustor

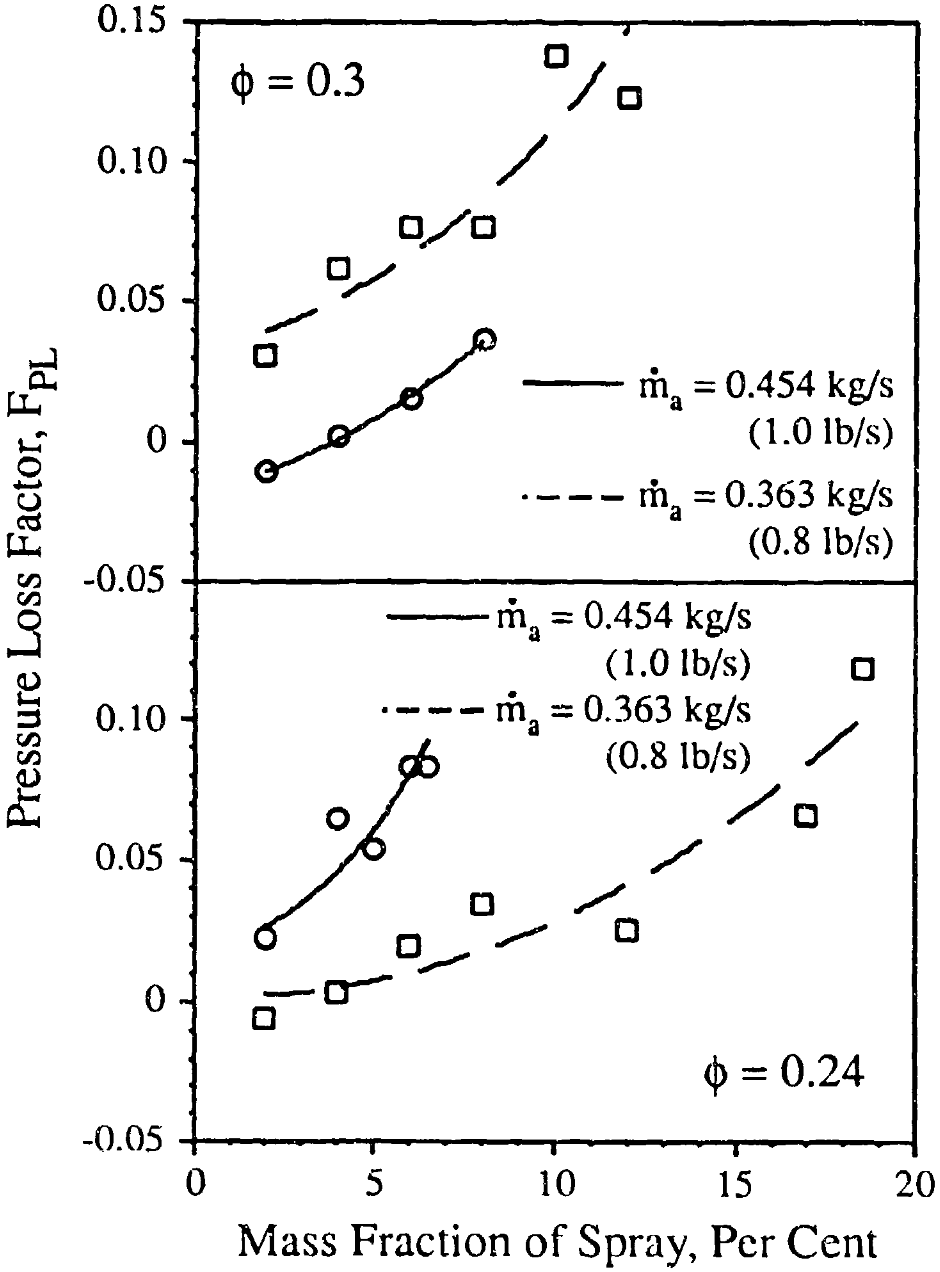


Figure 54. Case 1: Effect of Mass Fractions of Injected Water in Spray Form on Pressure Loss Factor

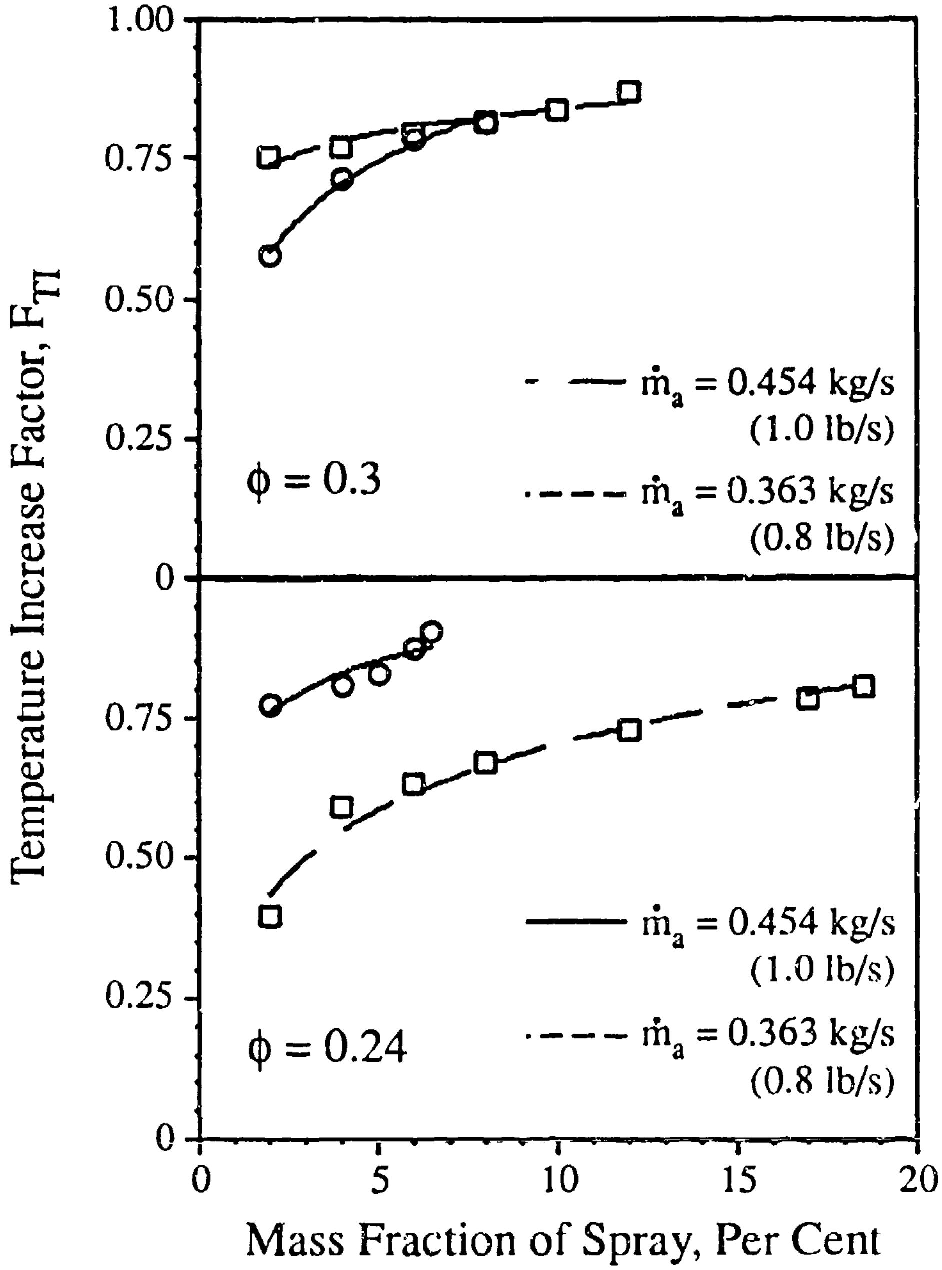


Figure 55. Case 1: Effect of Mass Fractions of Injected Water in Spray Form on Temperature Increase Factor

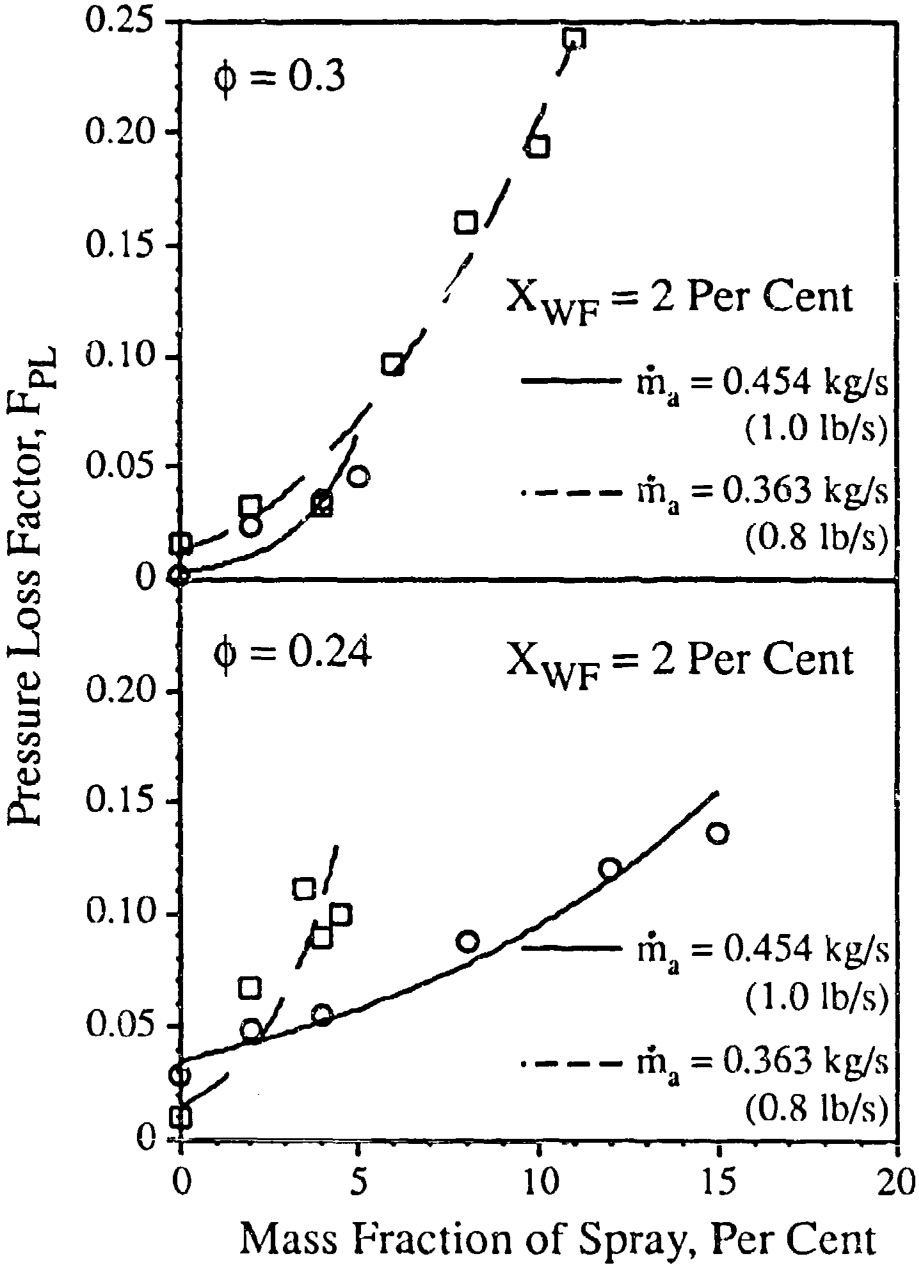


Figure 56. Case 1: Effect of Mass Fractions of Injected Water in Film and Spray Form on Pressure Loss Factor

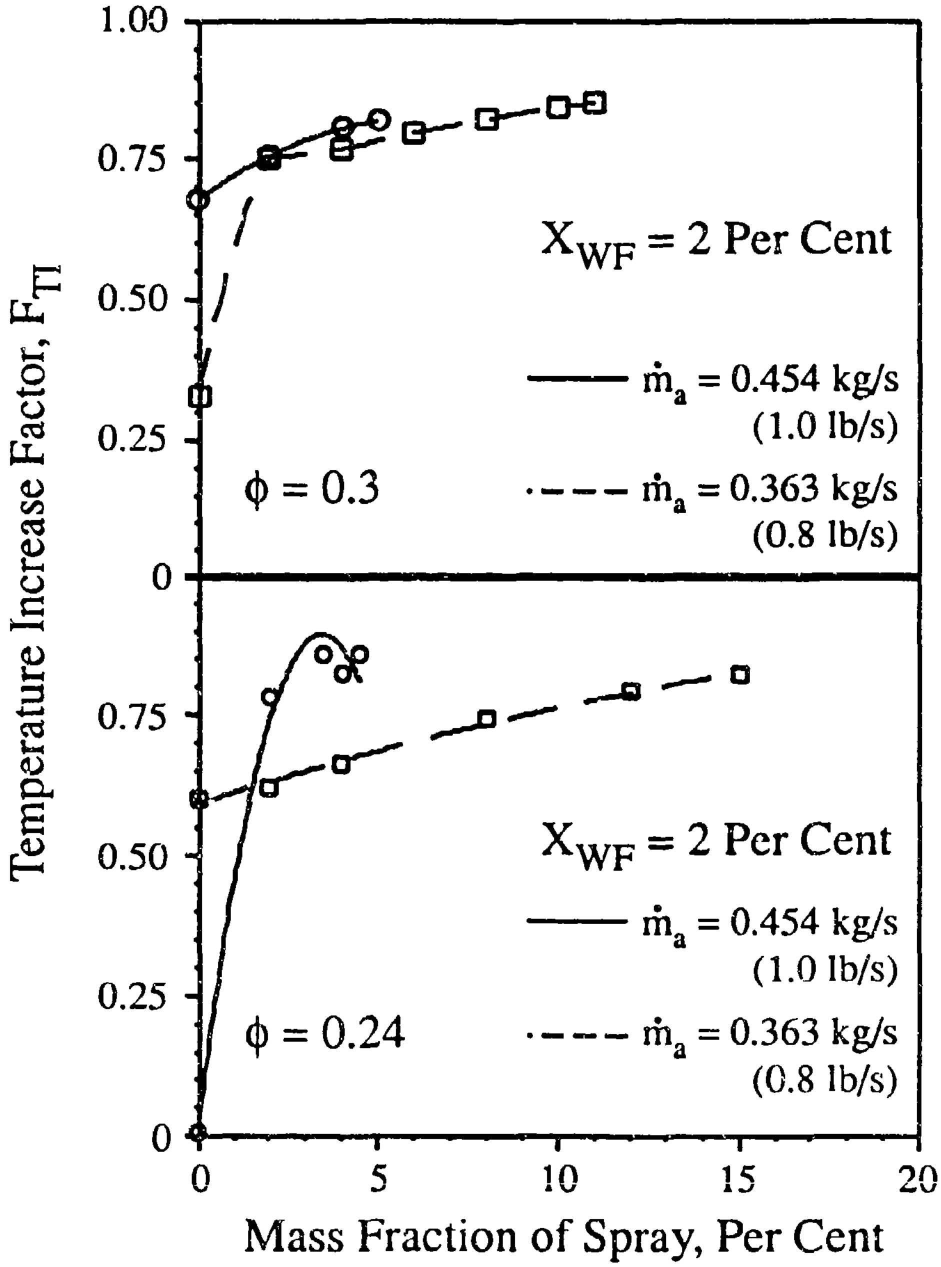


Figure 57. Case 1: Effect of Mass Fractions of Injected Water in Film and Spray Form on Temperature Increase Factor

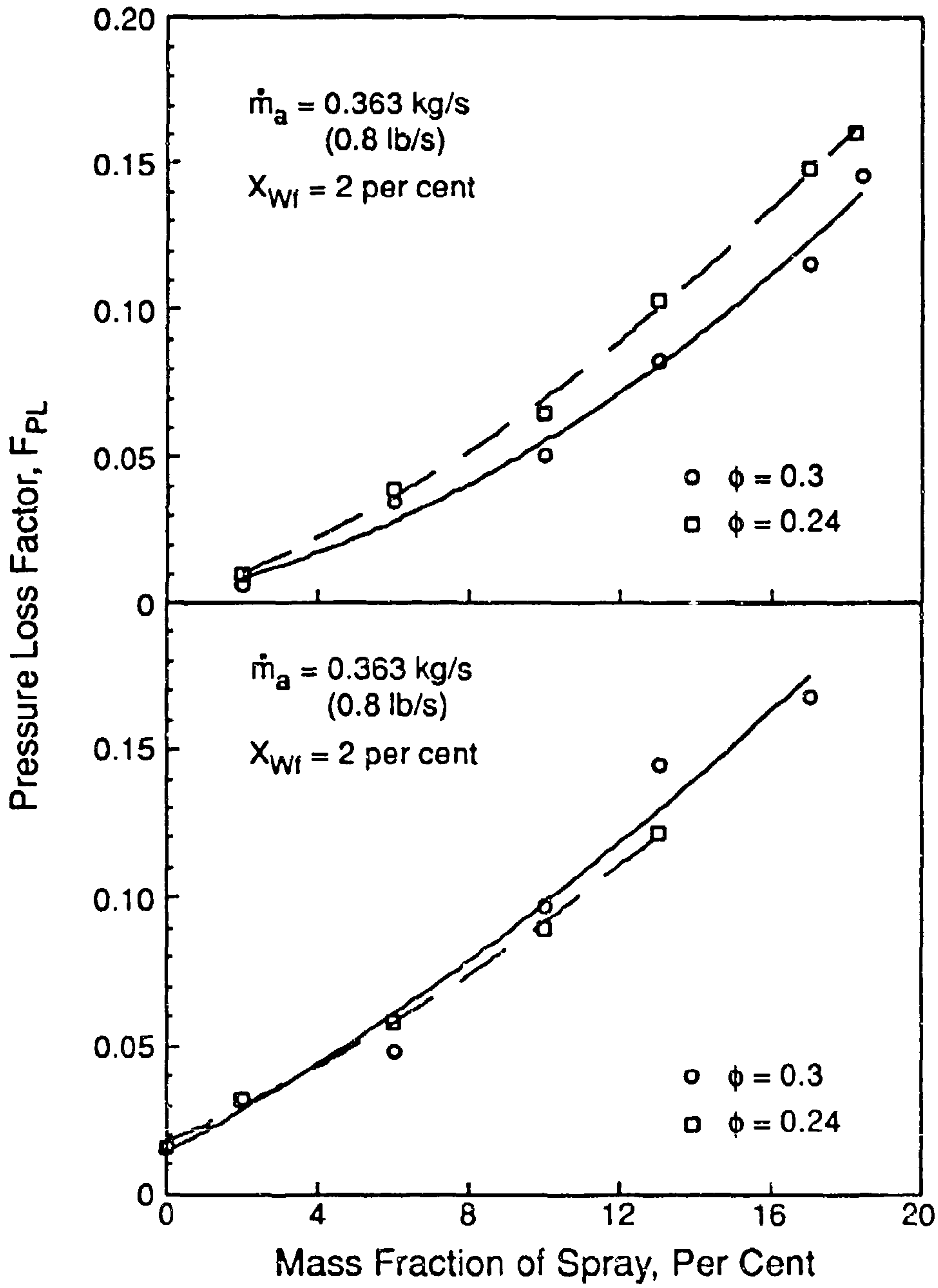


Figure 58. Case 2: Effect of Mass Fractions of Injected Water in Film and Spray Form on Pressure Loss Factor

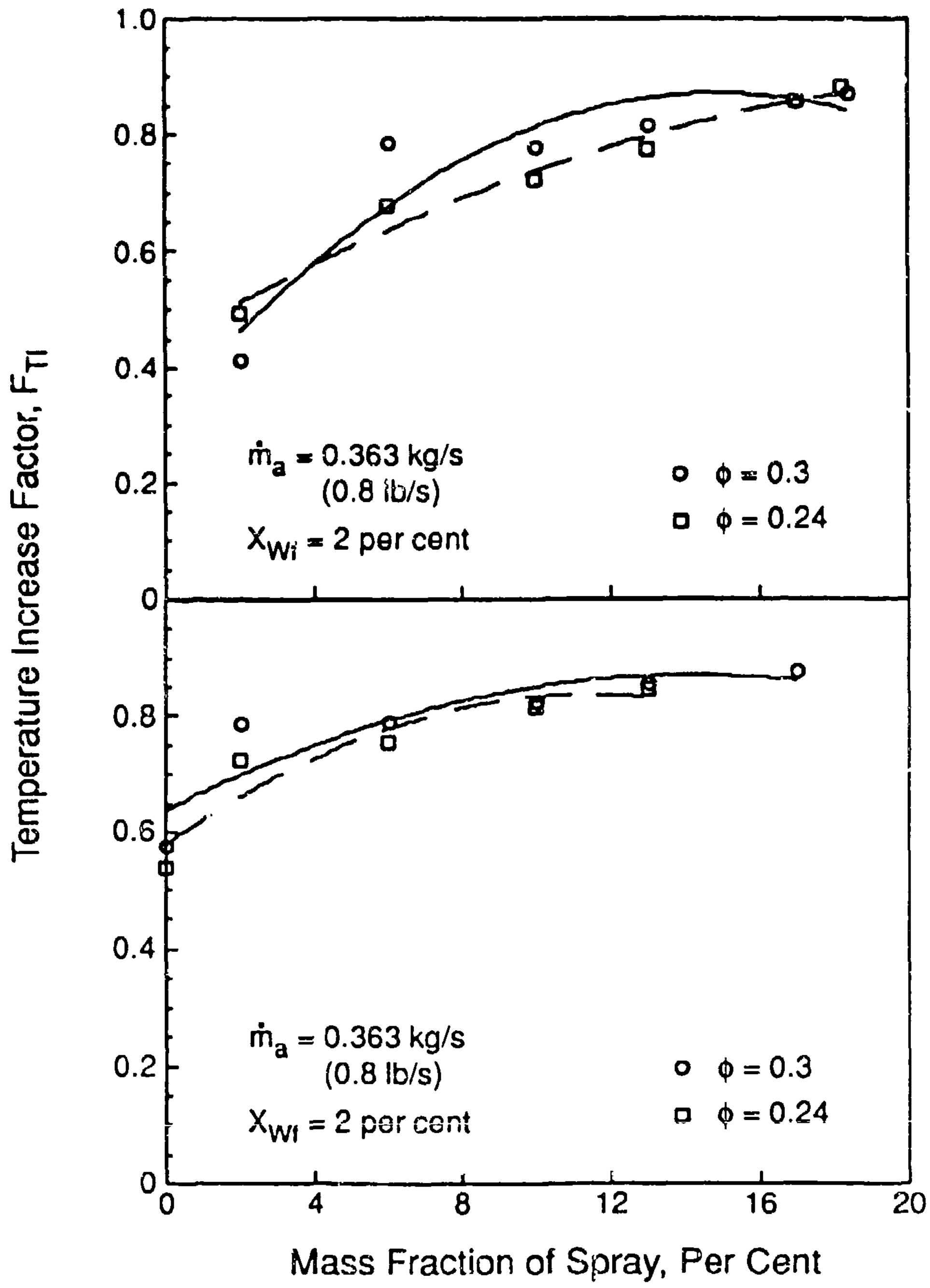


Figure 59. Case 2: Effect of Mass Fractions of Injected Water in Film and Spray Form on Temperature Increase Factor

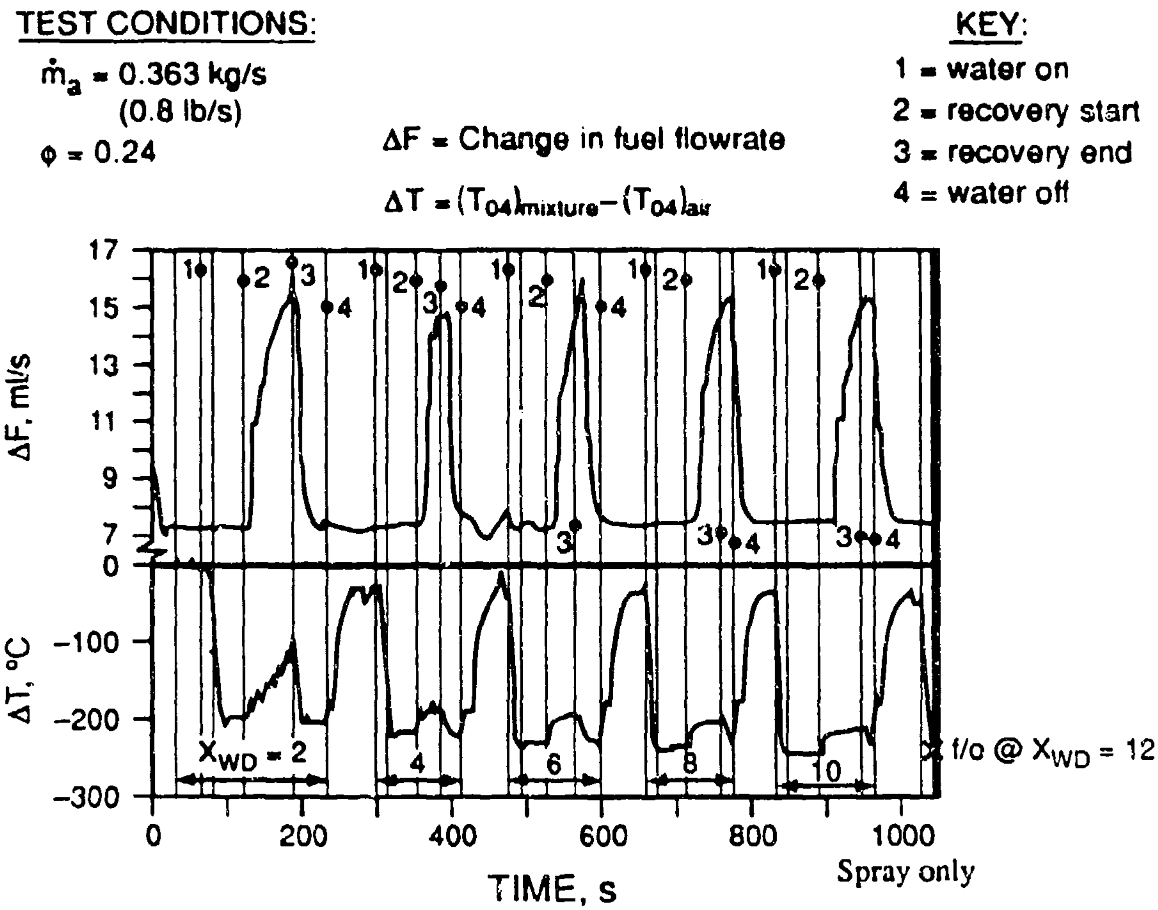


Figure 60. Reduction in Combustor Exit Temperature vs. Time

(A) Spray only (continued).

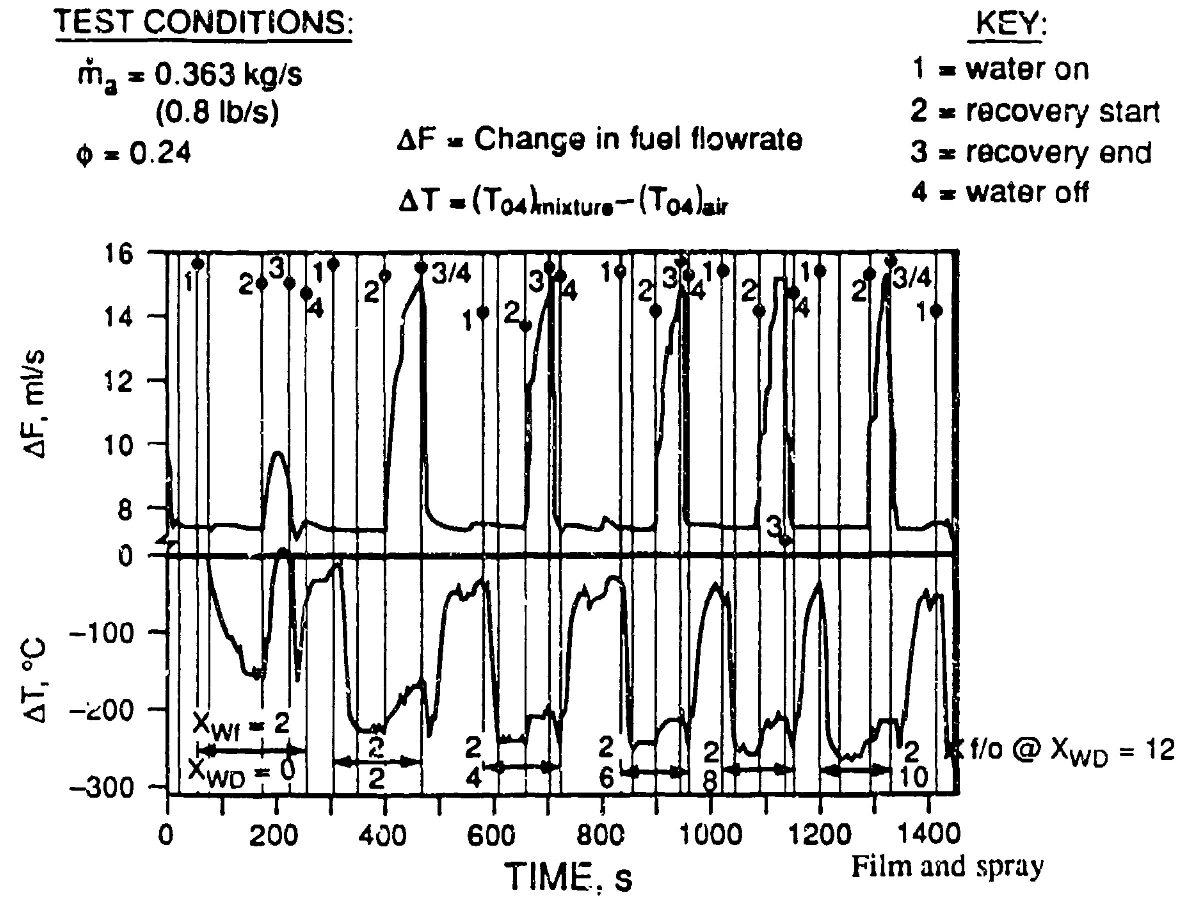


Figure 60. Reduction in Combustor Exit Temperature vs. Time
(B) Film and Spray (concluded).

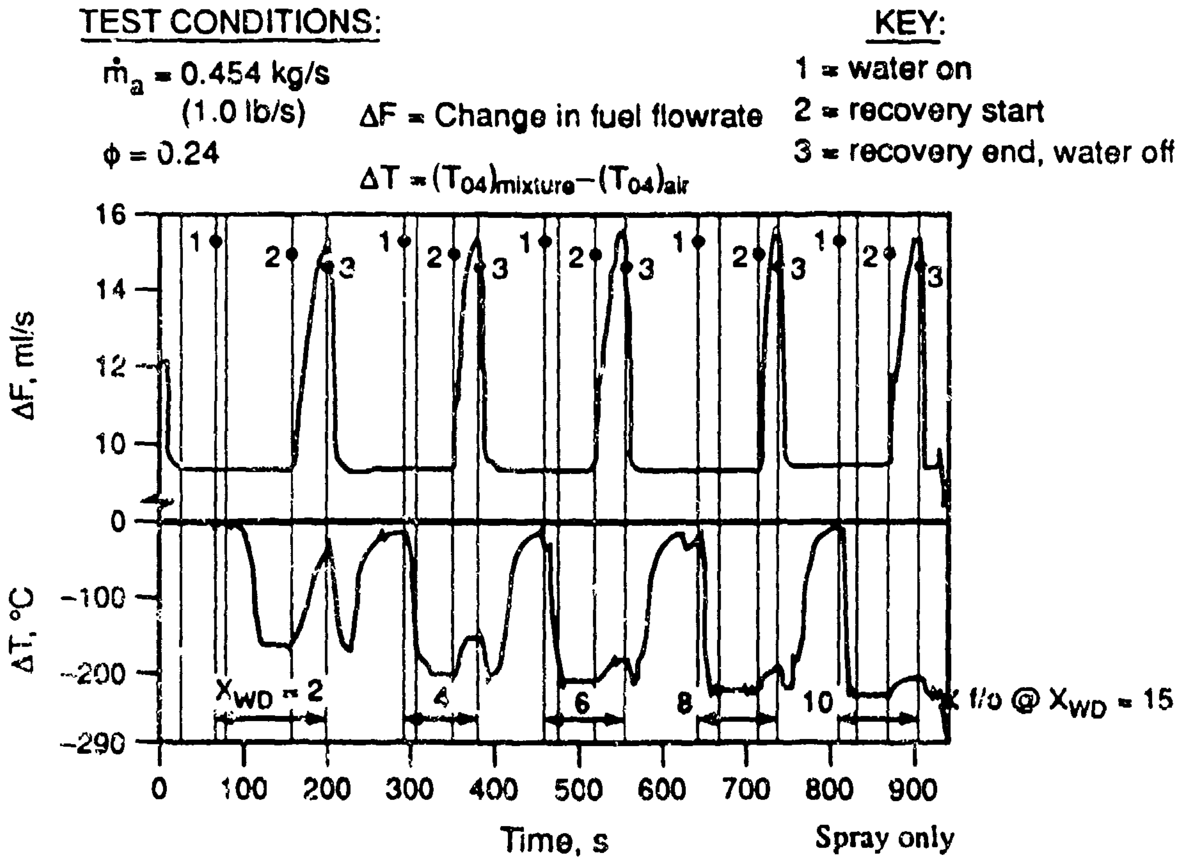


Figure 61. Reduction in Combustor Exit Temperature vs. Time

(A) Spray only (continued).

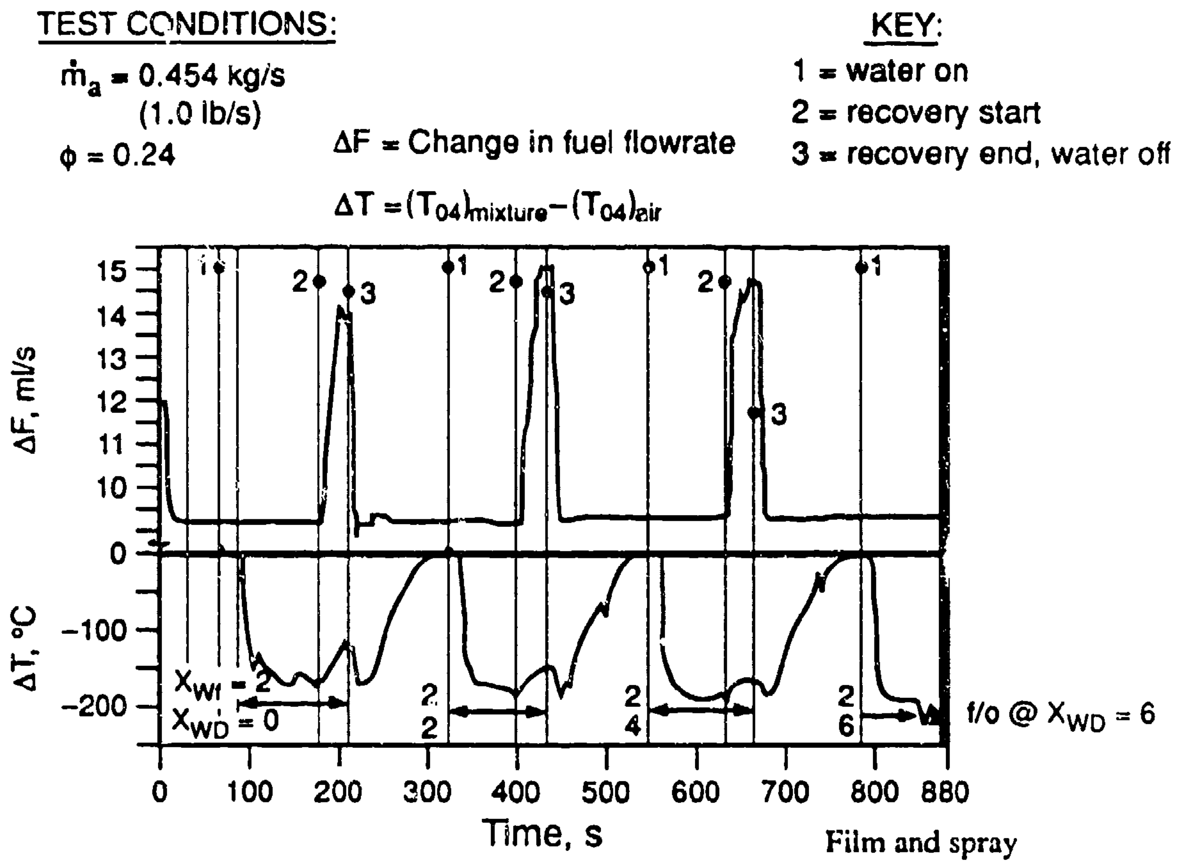


Figure 61. Reduction in Combustor Exit Temperature vs. Time

(B) Film and Spray (concluded).

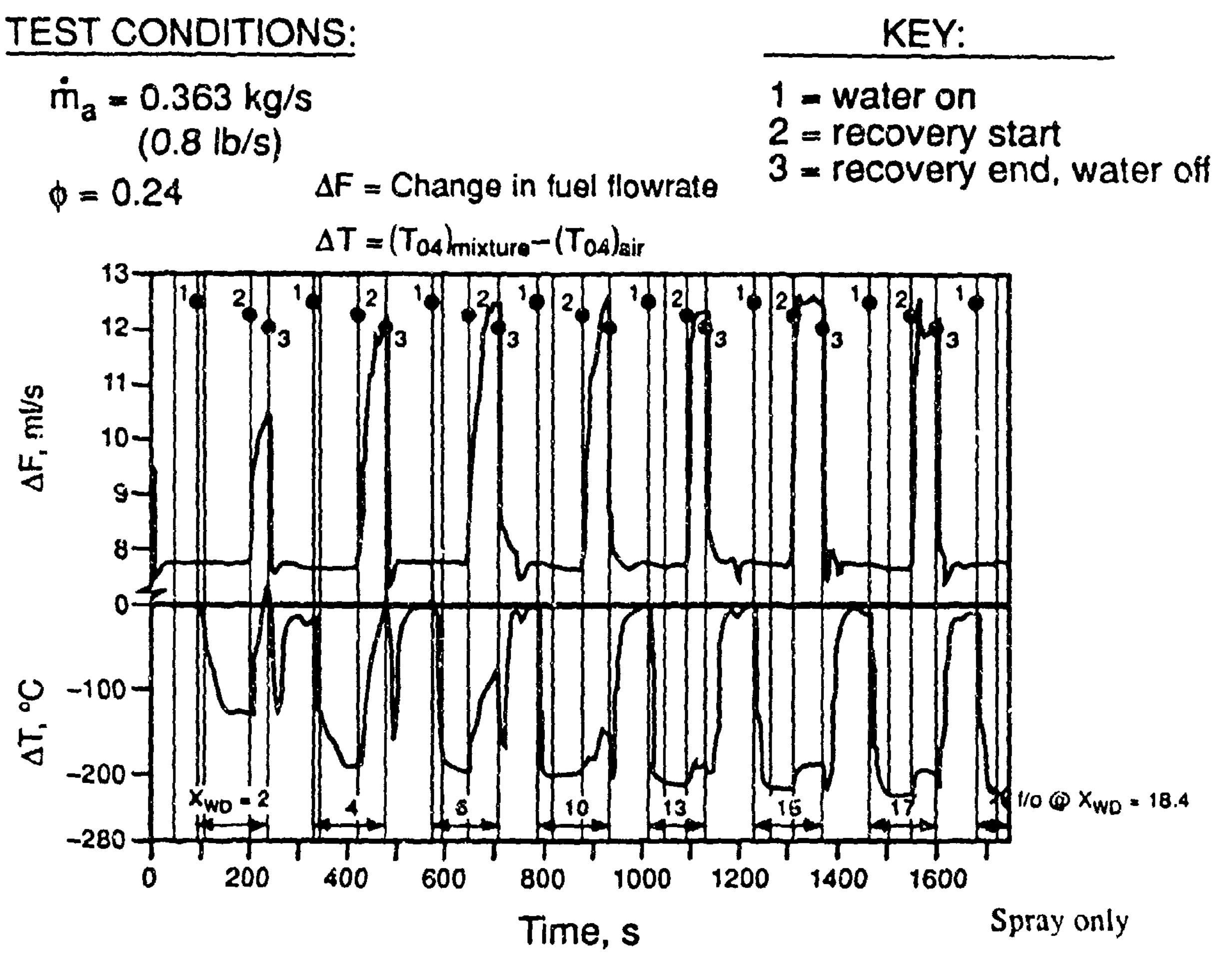


Figure 62. Reduction in Combustor Exit Temperature vs. Time

(A) Spray only (continued).

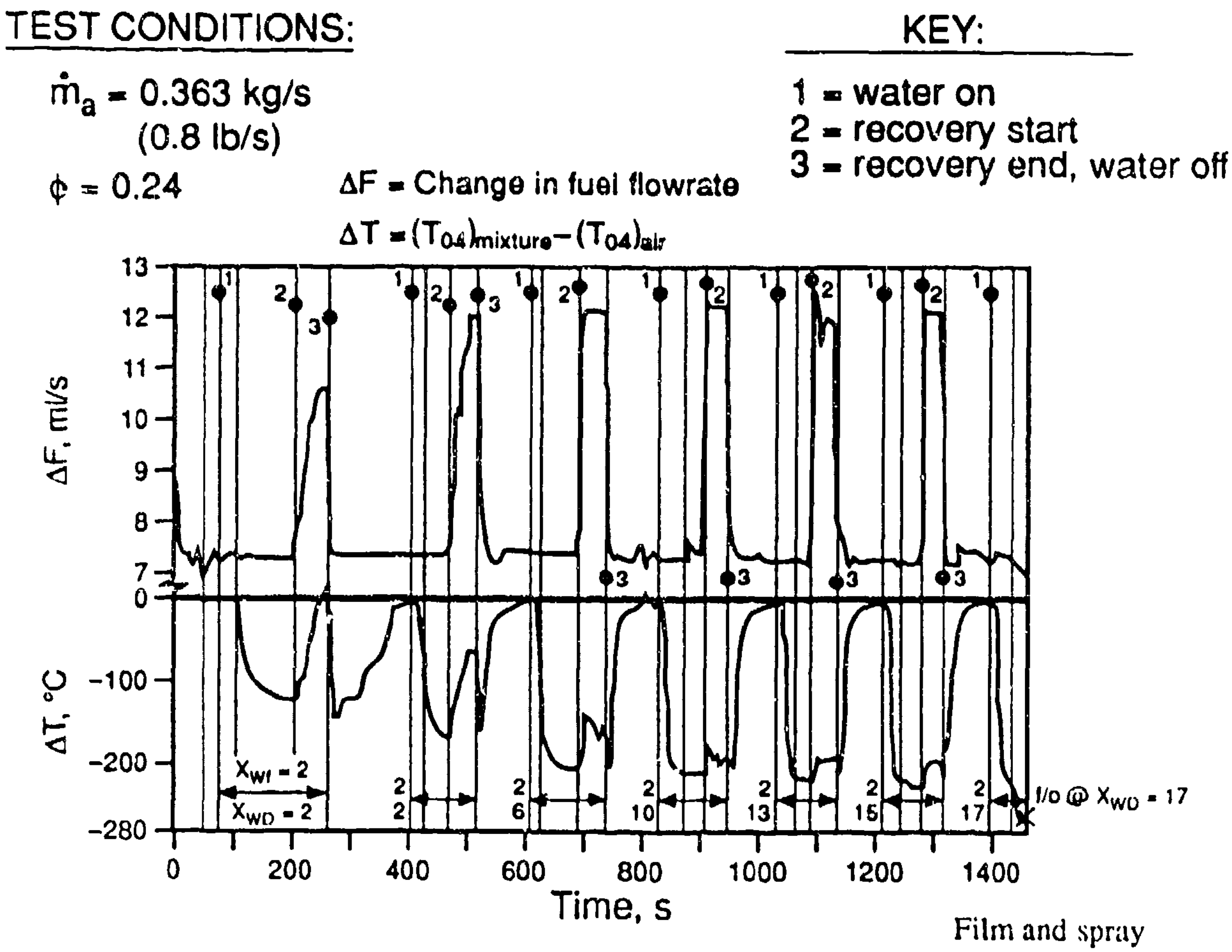


Figure 62. Reduction in Combustor Exit Temperature vs. Time

(B) Film and Spray (concluded).

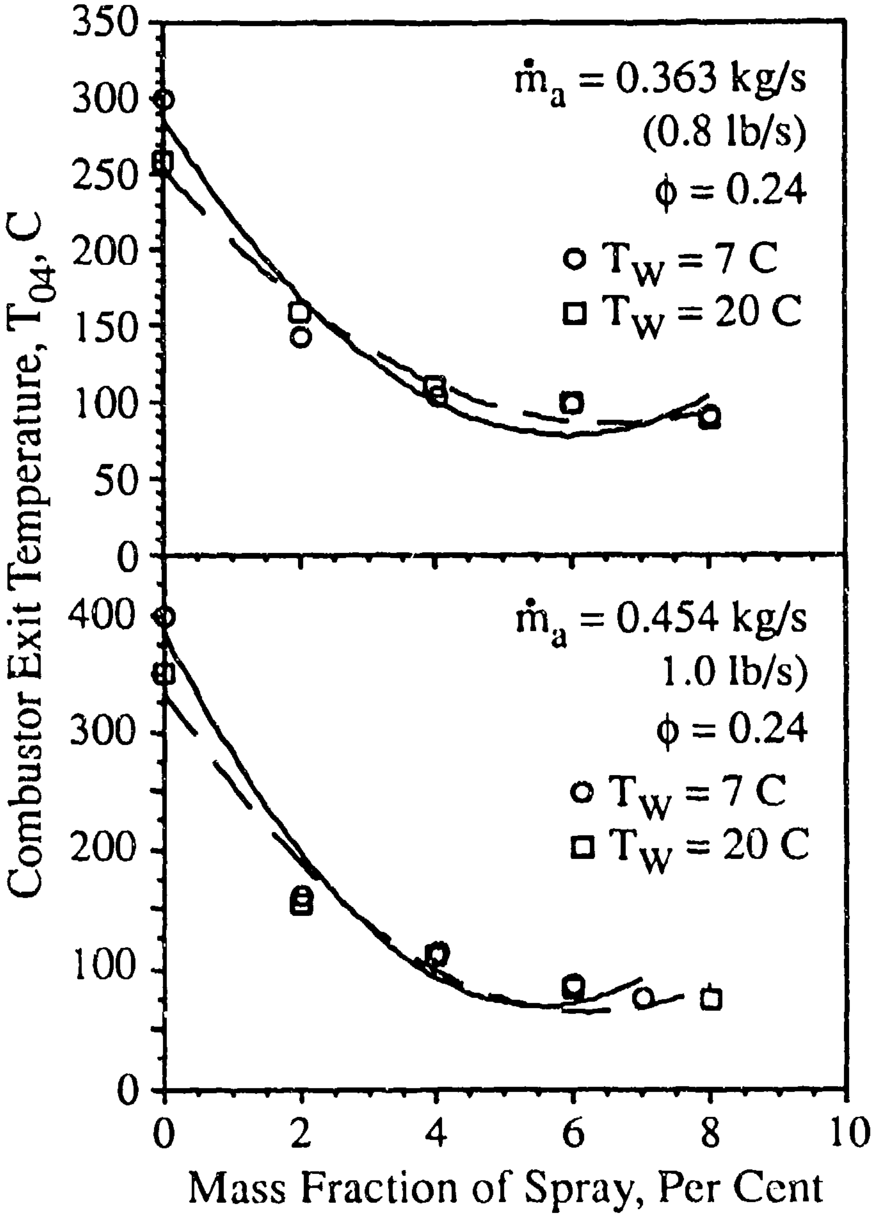


Figure 63. Effect of Injected Water Spray Temperature on Combustor Exit Temperature

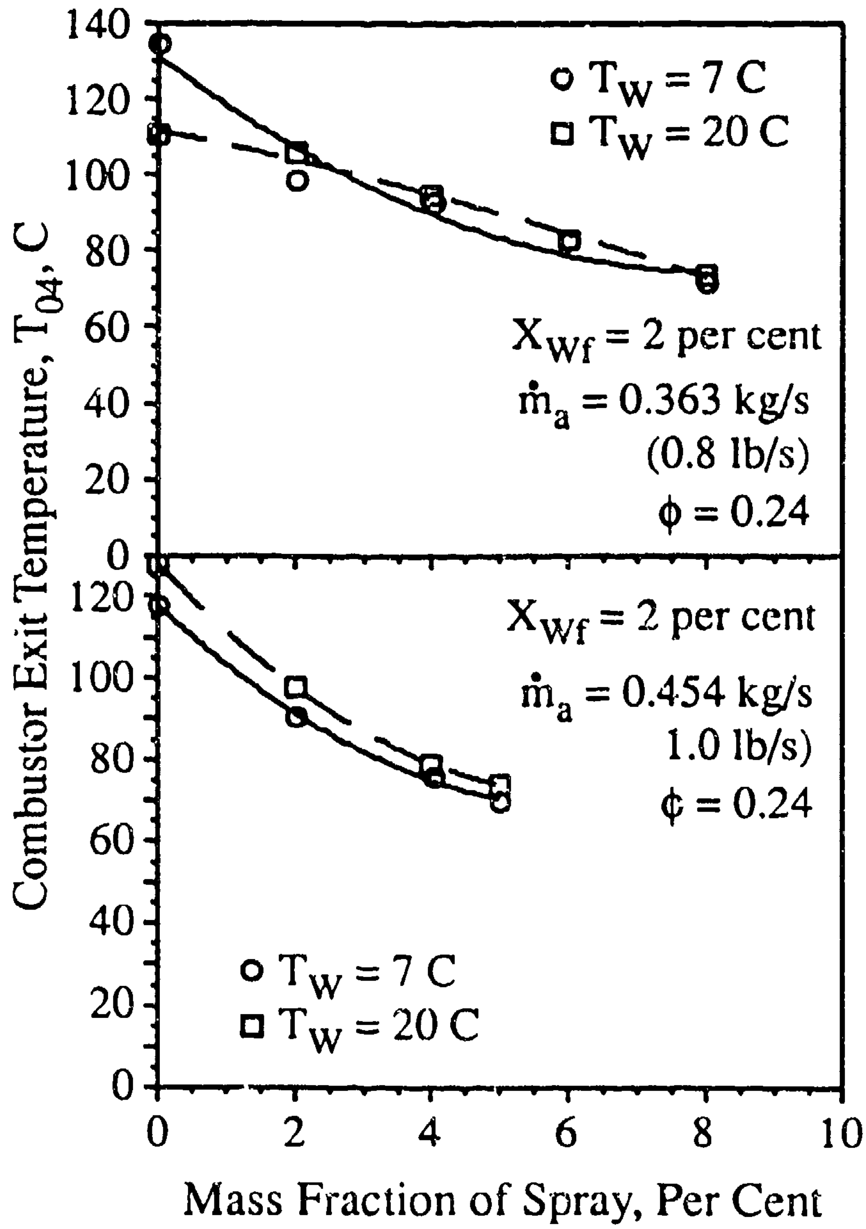


Figure 64. Effect of Injected Water Film and Spray Temperature on Combustor Exit Temperature

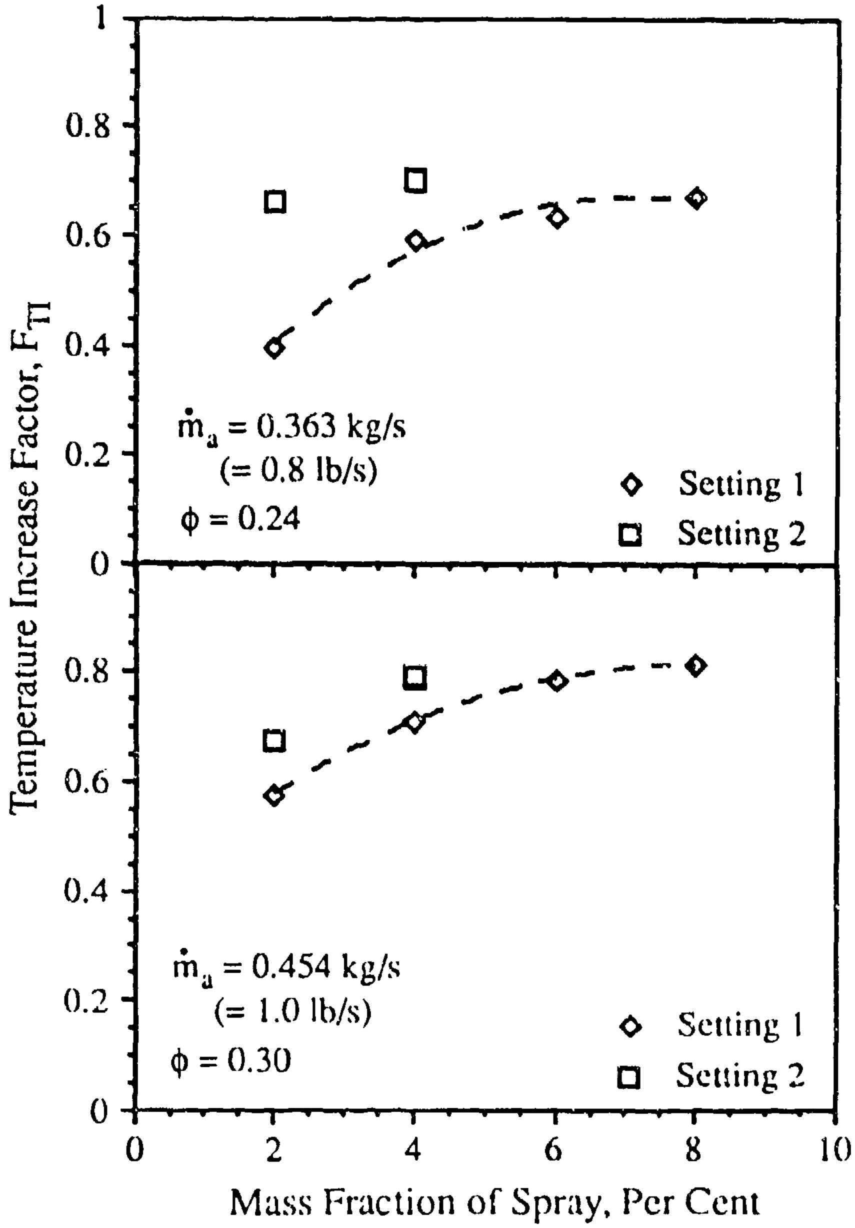


Figure 65. Effect of Test Article Orientation on Temperature Increase Factor (Water Spray Injection)

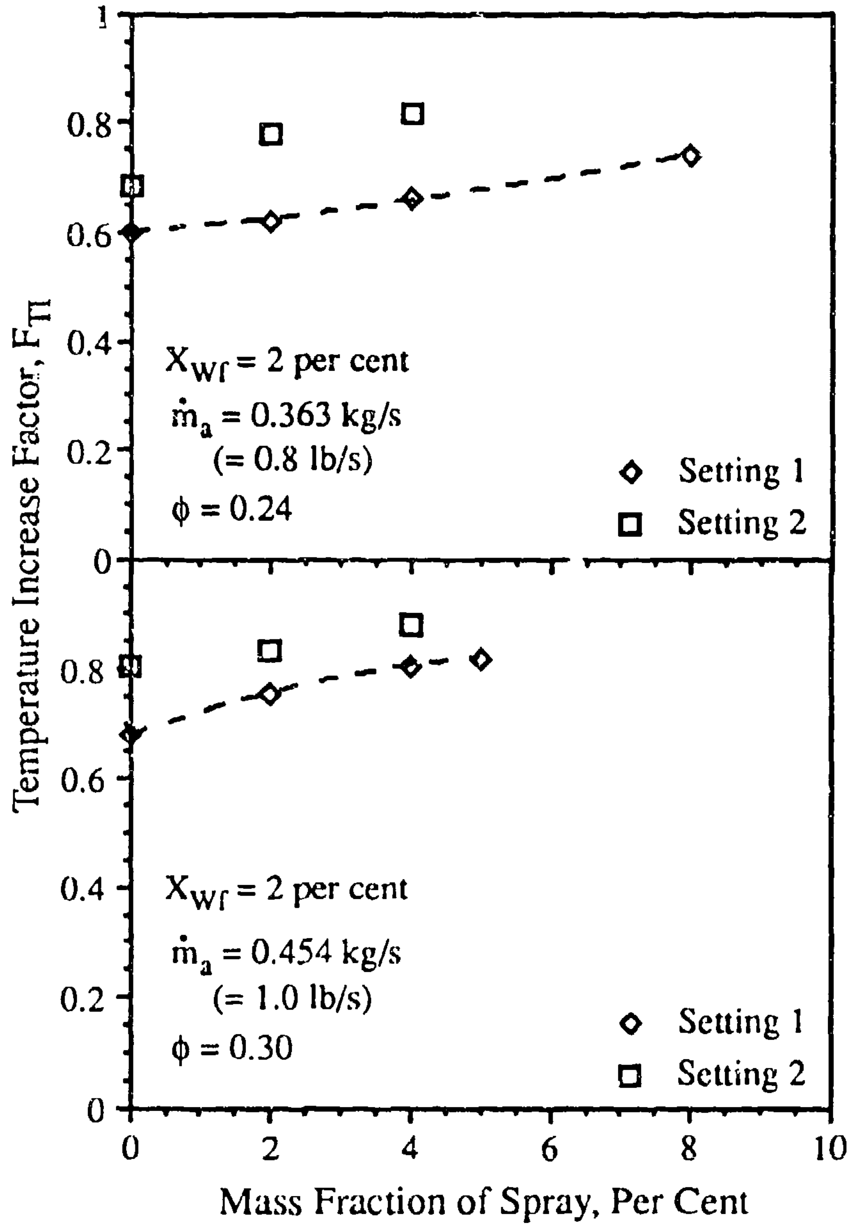


Figure 66. Effect of Test Article Orientation on Temperature Increase Factor (Water Film and Spray Injection)

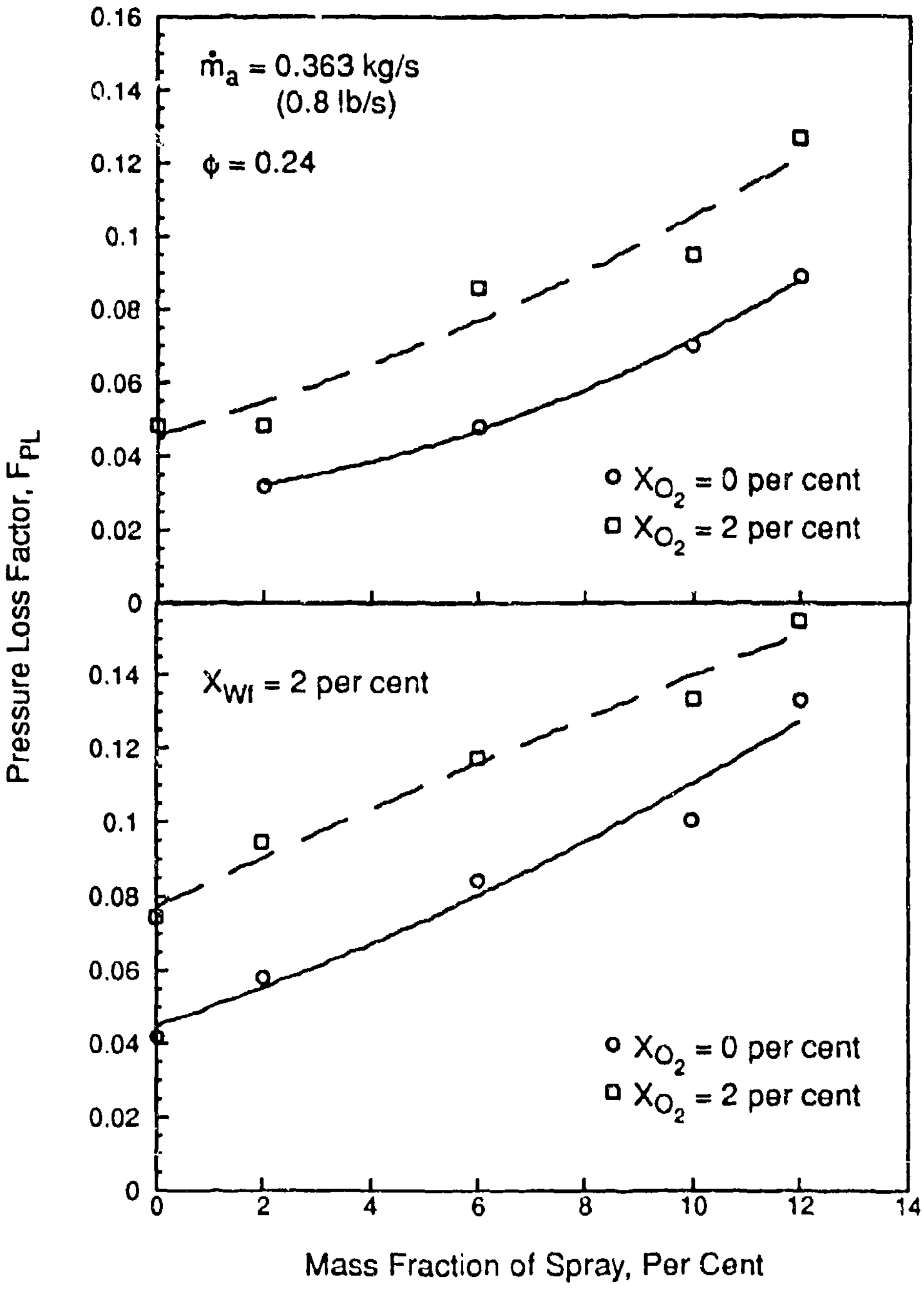


Figure 67. Case 1: Effect of Mass Fractions of Injected Water in Film Form and Spray of Small Droplet Size on Pressure Loss Factor (X_{O2} = 2.0, ϕ = 0.24)

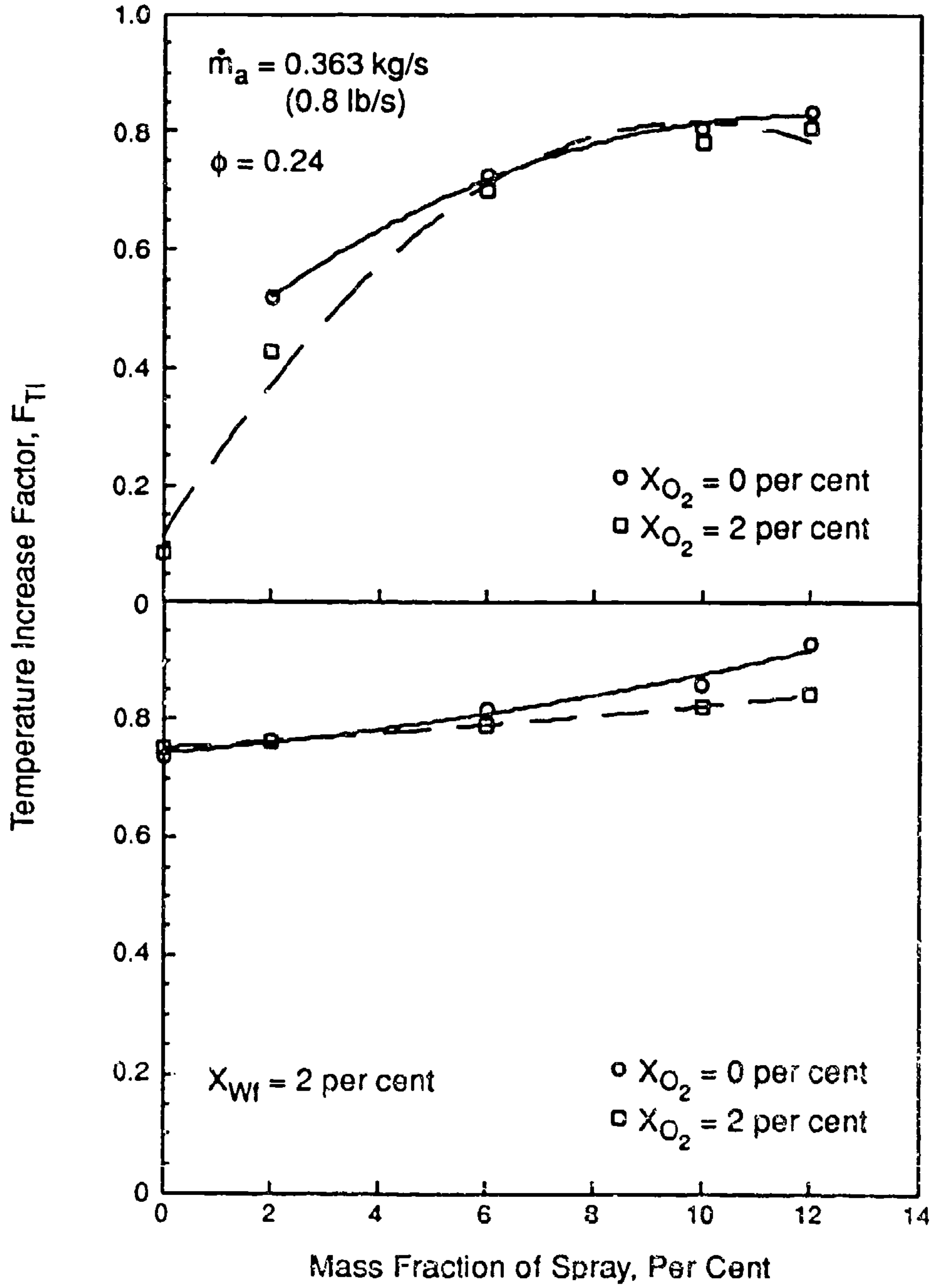


Figure 68. Case 1: Effect of Mass Fractions of Injected Water in Film Form and Spray of Small Droplet Size on Temperature Increase Factor (X_{O2} = 2.0, ϕ = 0.24)

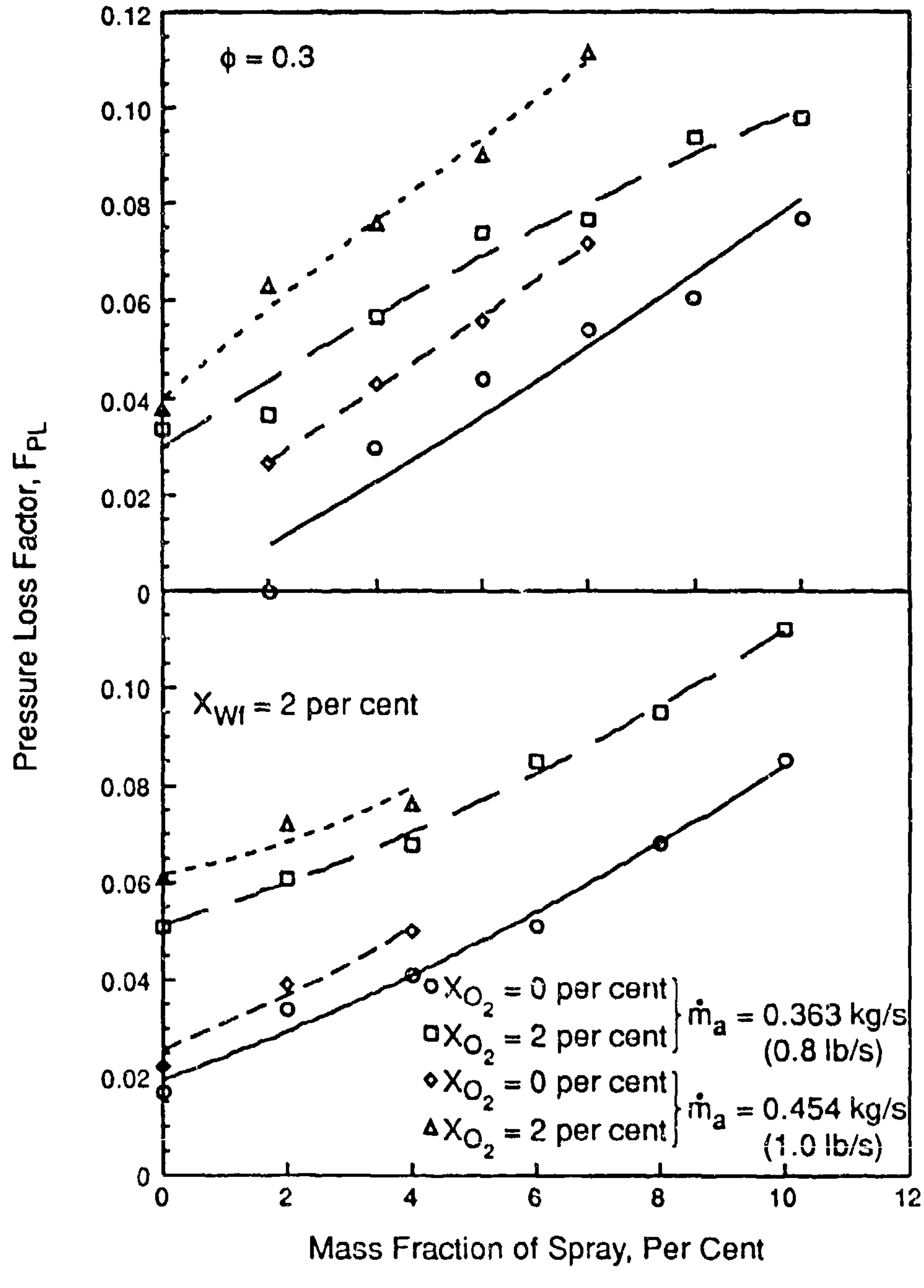


Figure 69. Case 1: Effect of Mass Fractions of Injected Water in Film Form and Spray of Small Droplet Size on Pressure Loss Factor (X_{O2} = 2.0, ϕ = 0.3)

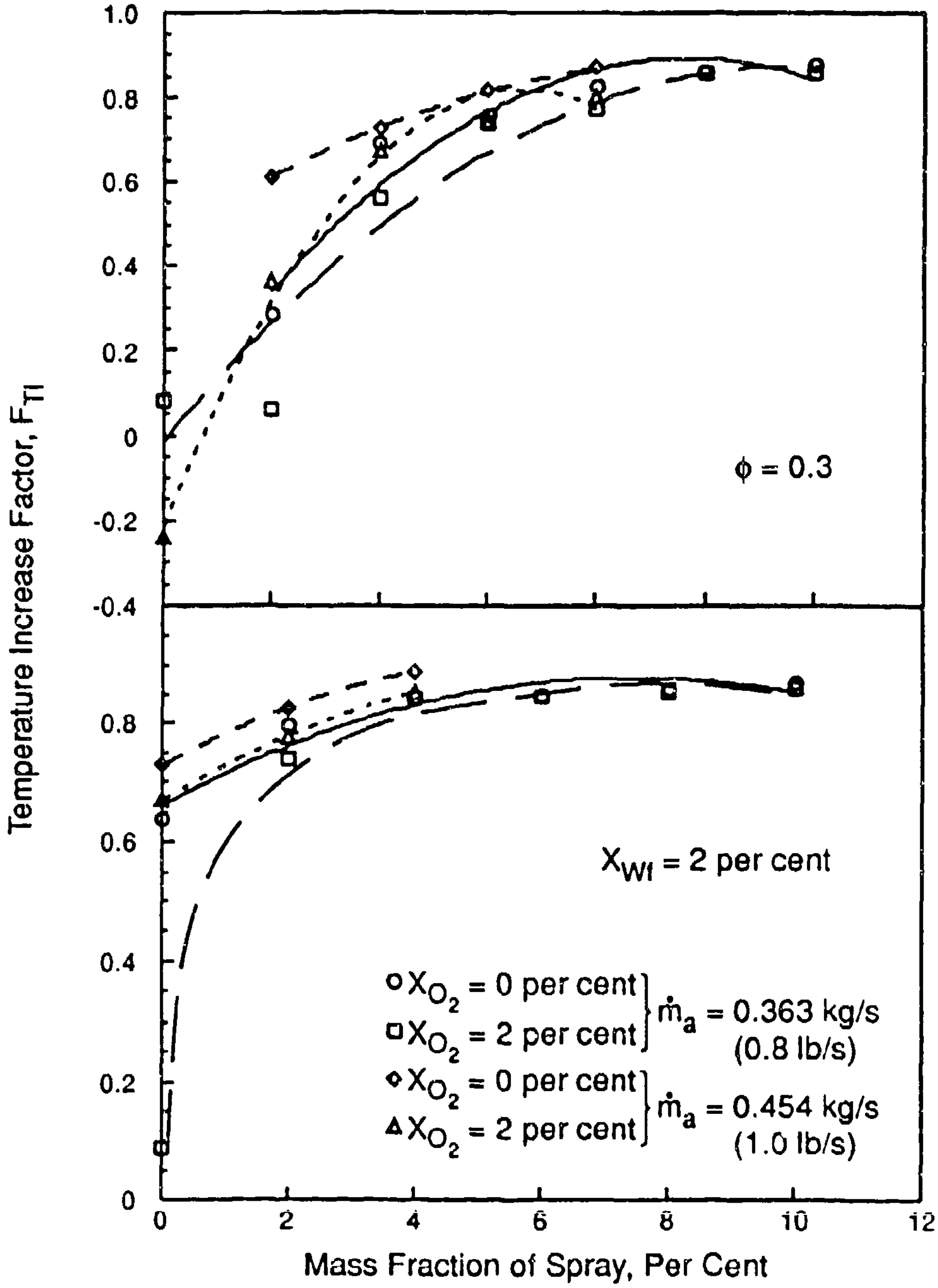


Figure 70. Case 1: Effect of Mass Fractions of Injected Water in Film Form and Spray of Small Droplet Size on Temperature Increase Factor (X_{O2} = 2.0, $\phi = 0.3$)

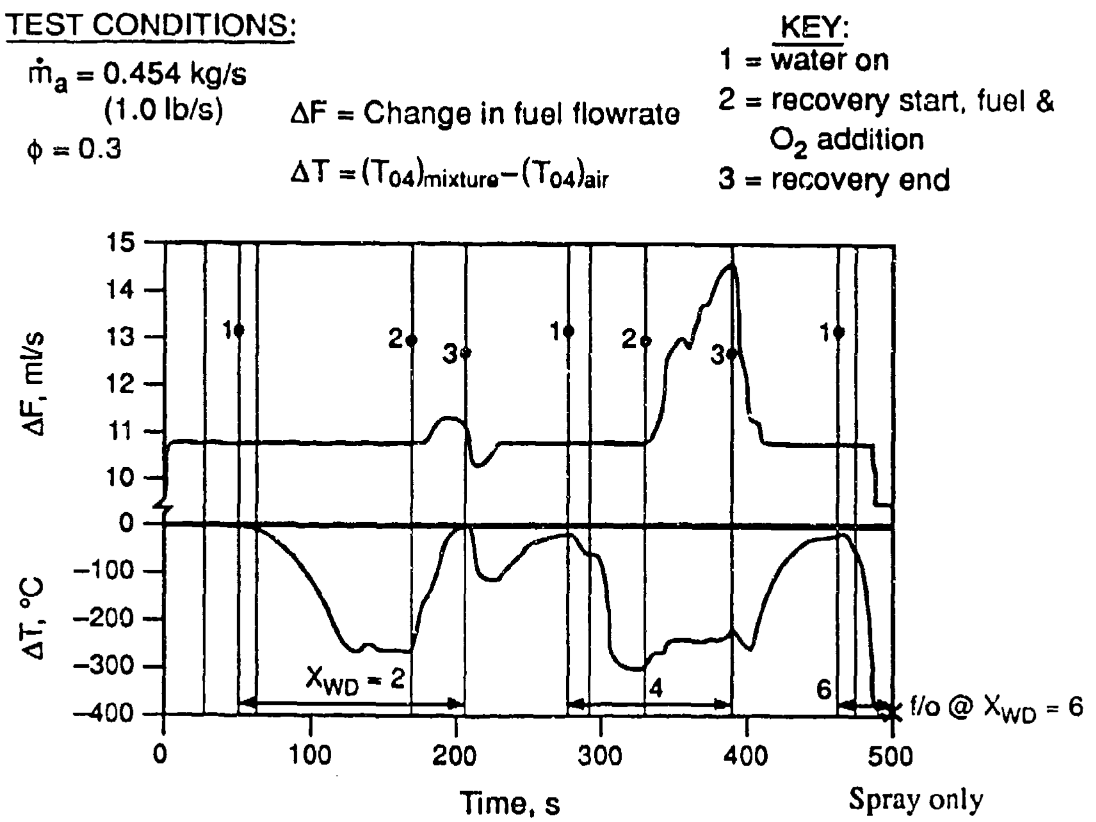


Figure 71. Reduction in Combustor Exit Temperature vs. Time (A) Spray only (continued).

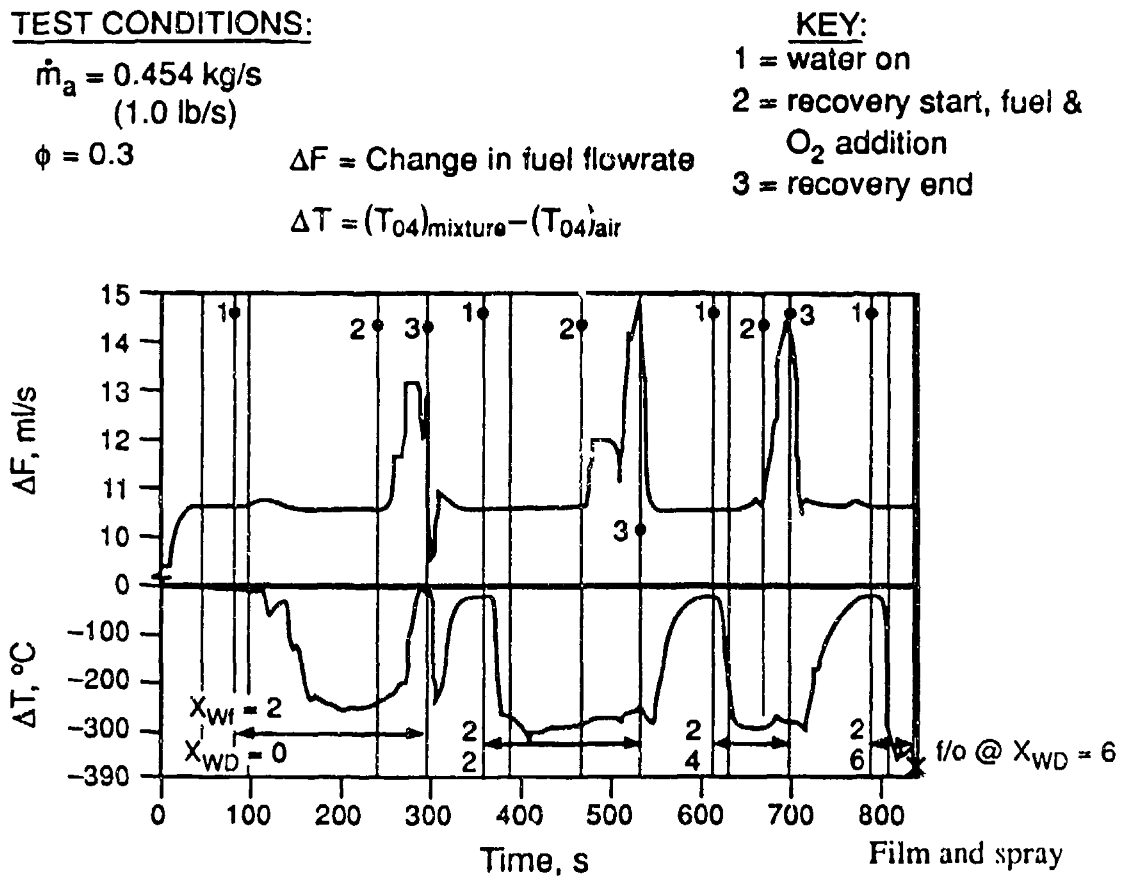


Figure 71. Reduction in Combustor Exit Temperature vs. Time (B) Film and Spray (concluded).

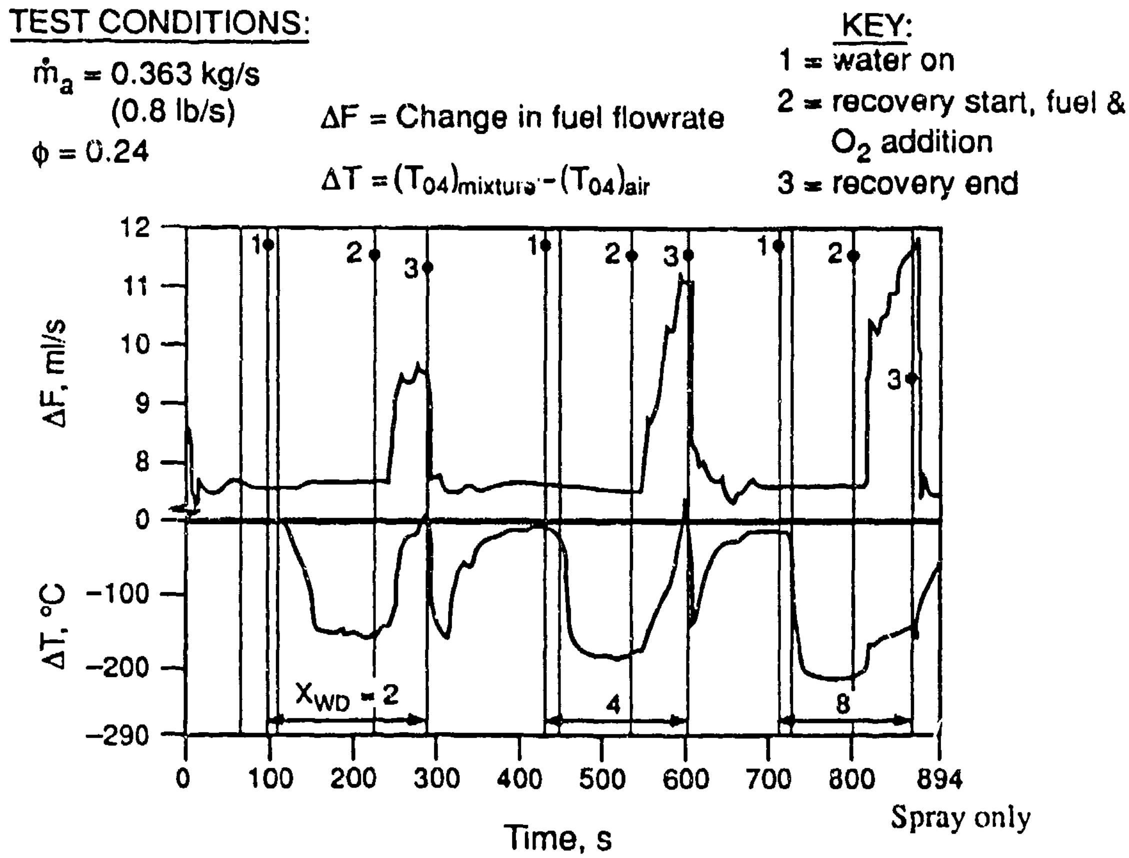


Figure 72. Reduction in Combustor Exit Temperature vs. Time
(A) Spray only (continued).

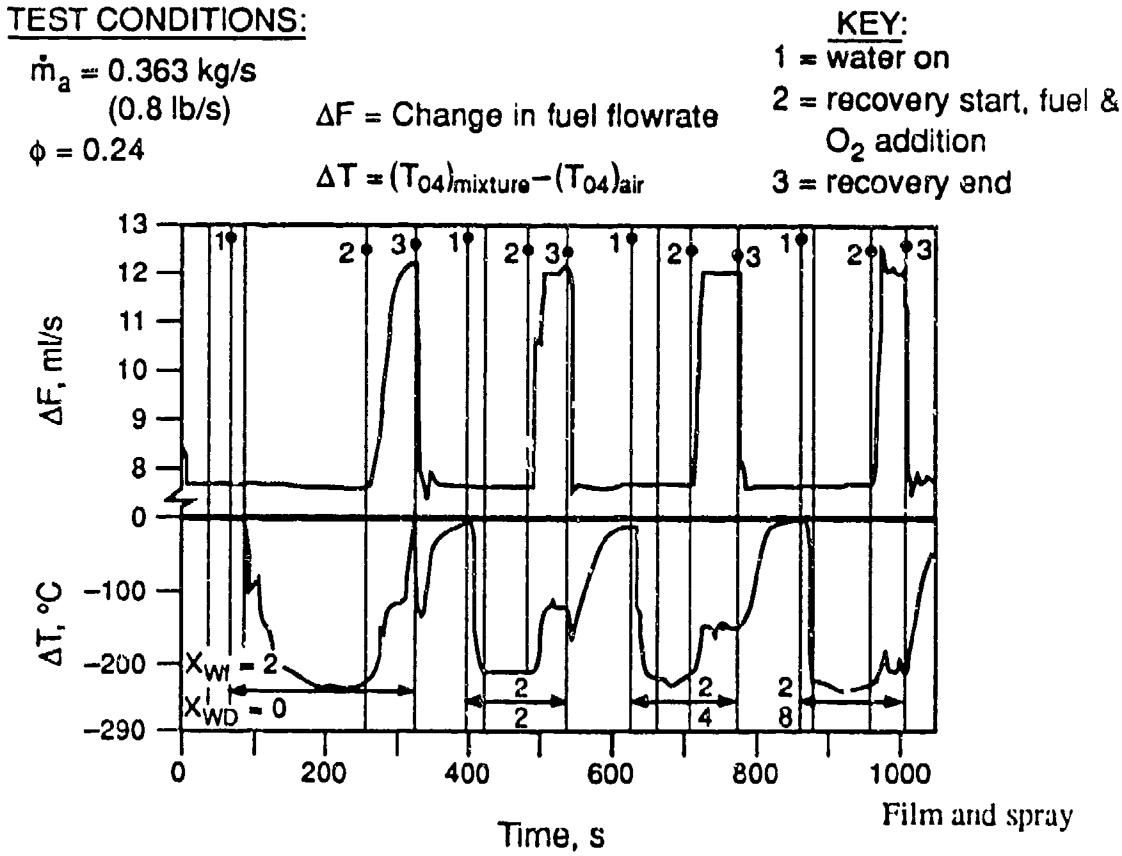


Figure 72. Reduction in Combustor Exit Temperature vs. Time

(B) Film and Spray (concluded).

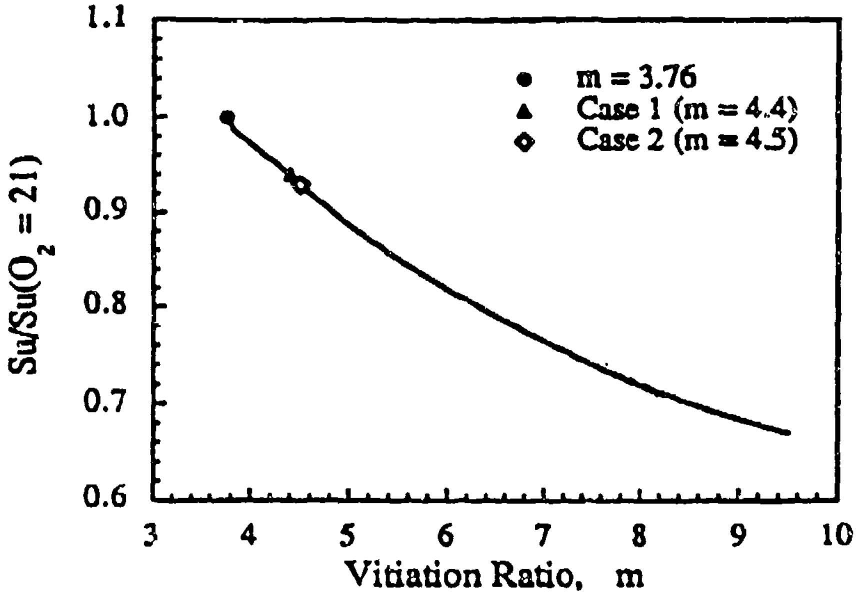


Figure 73. Effect of Quantity of Vitiation on Flame Speed

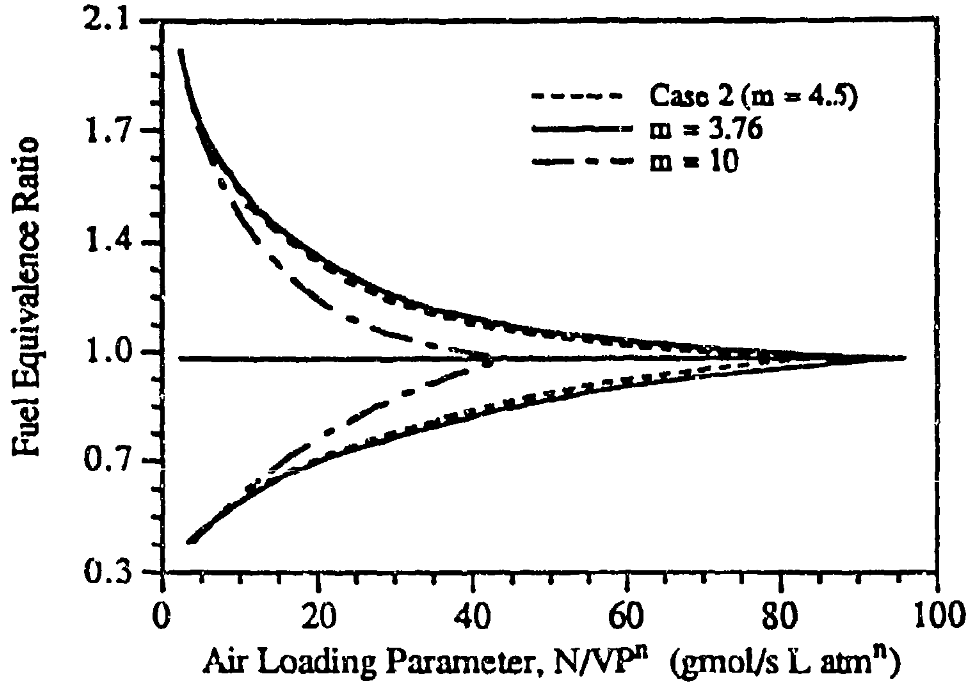


Figure 74. Effect of Vitiation on Flameout Characteristics

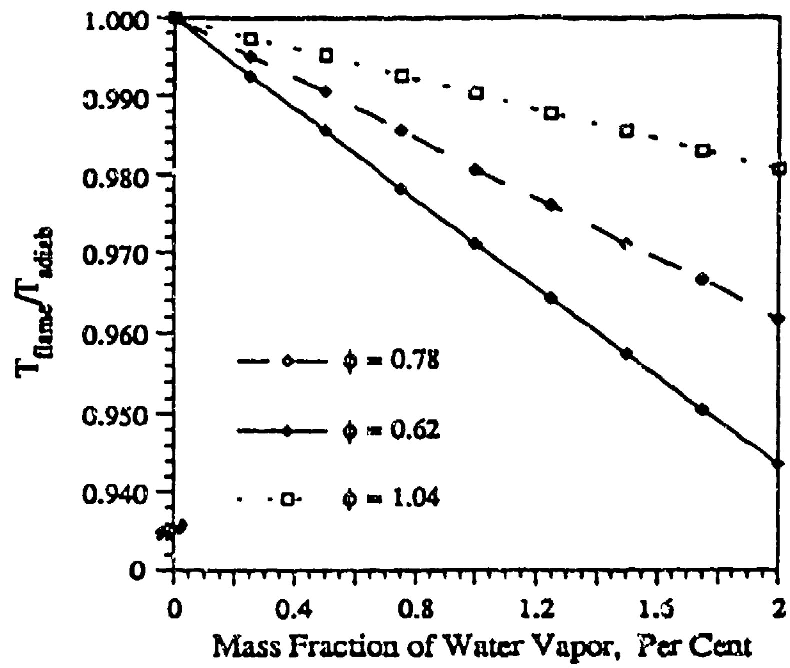


Figure 75. Effect of Vitiation on Flame Temperature

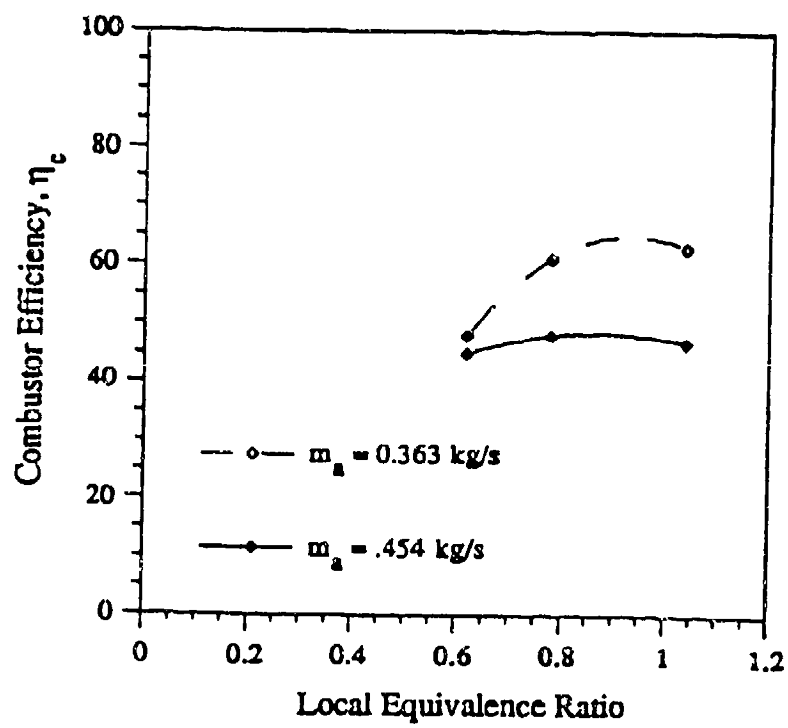


Figure 76. Overall Combustor Efficiency

APPENDIX A. SPRAY NOZZLE CHARACTERISTICS

Details are provided in the following on the two models of spray nozzles utilized in this investigation. The nozzles have been acquired from Spraying Systems, Co., Wheaton, Ill. One of 'he models is a standard pressure nozzle while the other is an air-assist nozzle, utilized for generation of small droplet sizes. Table A-1 includes the overall features of the two nozzles. Table A-2 provides some operational conditions of the nozzles.

TABLE A-1. SPRAY NOZZLES USED IN THE INVESTIGATION

Nozzle Type	Spray Pattern	Drop Size Classification	Spray Angle (degrees)
Veejet 2510 (pressure)	Flat	Small-Medium	15 to 0
Casterjet 19880 (air-blast)	Flat	Small	50

TABLE A-2. CHARACTERISTICS OF THE SPRAY NOZZLES

Nozzle Type	Water Pressure (kPa)	Air Pressure (kPa)	Capacity (Lpm)	Mean Volumetric Droplet Size (microns)
Veejet 2510 (pressure)	207	-	2.7	1250
	276	-	3.8	
	414	-	4.5	
	552	-	5.3	
	689	-	6.1	875
Casterjet 19880 (air-blast)	up to 689 kPa	310	4.9	20 to 25

APPENDIX B. FUEL SUPPLY SYSTEM CALIBRATIONS.

The fuel equivalence ratios utilized in the burning tests ranged from 0.24 to 0.4. The air mass flow rates ranged from 0.363 to 0.454 kg/s. With these operating conditions, the required fuel flow rate ranged from approximately 7.0 to 15.0 mL/s. Calibration tests have been carried out on the three fuel injector bits, both individually and collectively (connected to the manifold). The objective of the tests were several-fold:

1. To calibrate the micro flowmeter in situ,

2. to verify the performance of the individual fuel injector bits, and

3. to ensure that, for a given fuel flowrate to the manifold, the flowrates of the three injector bits (all attached to the manifold) are similar (i.e. within an error band of 1.0 per cent, as specified by GE).

The results of items 1 and 3 are presented in tables B-1 and B-2, respectively.

TABLE B-1. FUEL FLOW METER CALIBRATION

Test Conditions:		Fuel Pressure	App. Fuel Flowrate
		450 kPa	10 ml/s
Test Number	Volume (mL)	Counts	K (pulses/mL)
1	1,392.5	11,964	8.59174
2	1,343.8	11,490	8.55070
3	1,356.3	11,602	8.55447
4	1,341.3	11,486	8.56365
5	1,362.5	11,681	8.57321
6	1,543.8	13,227	8.56810
7	1,571.3	13,484	8.58170
8	1,558.8	13,345	8.56135
9	1,592.5	13,663	8.57959
10	1,557.5	13,376	8.58812
11	1,513.8	12,948	8.55359
12	1,530.0	13,101	8.56275
13	1,415.0	12,150	8.58657
14	1,547.5	13,246	8.55961
15	1,550.0	13,308	8.58581
16	1,576.3	13,560	8.60270
	Average K	=	8.57273
	σ_{n-1}	=	0.01566

The procedure for the calibration of the flow meter is as follows: (i) The flow meter display unit (which counts the number of pulses outputted from the flow meter, and determines the frequency) is zeroed. (ii) fuel is flowed through the meter at a flow rate of approximately 10 mL/s until about 1.5 L has been collected. (iii) The flow of fuel is then stopped, the volume of collected fuel is measured, and the number of pulses recorded. (iv) The calibration constant of

the flow meter, K (pulses/mL), is then calculated by dividing the number of pulses by the volume of fuel. The procedure has been repeated 16 times and an average K has been determined.

The procedure for item 3 is as follows: (i) fuel is flowed through the three injectors (which were connected to the fuel manifold) at an approximate flow rate of 10 mL/s for about 120 sec. (ii) The volume of atomized fuel leaving each injector is measured. (iii) the three volumes are then compared with each other. The test is repeated 13 times.

TABLE B-2. FUEL INJECTOR AND MANIFOLD PERFORMANCE

Test Conditions:	Fuel Pressure	App. Fuel Flowrate		Test Duration
	450 kPa	10 ml/s		120 sec
Test Number	Volume After 120 sec (ml)			Total volume (ml)
	Injector 1	Injector 2	Injector 3	
1	403	405	405	1213
2	407	410	410	1227
3	405	407	407	1219
4	404	406	406	1216
5	406	409	410	1225
6	403	405	406	1214
7	405	406	408	1219
8	402	404	405	1211
9	400	402	404	1206
10	405	407	410	1222
11	405	407	406	1218
12	402	401	404	1207
13	399	401	401	1201
Average	403.5	405.4	406.3	1215.2
σ_{n-1}	2.3	2.8	2.7	7.3
% Error	0.4	0.08	0.3	-

APPENDIX C. SAMPLE ϕ AND ϕ_O ANALYSIS CALCULATIONS

Ideal Heat Release: The ideal heat release was calculated by multiplying the mass flow rate of fuel entering the combustor by the lower heating value of the fuel as follows:

$$\begin{aligned}\text{Ideal Heat Release} &= \dot{m}_a \phi \phi_s \Delta H_f \\ &= 325\text{kJ}\end{aligned} \tag{C-1}$$

where $\dot{m} = 0.36\text{kg/s}, \phi = 0.31, \phi_s = 0.0676, \text{and } \Delta H_f = 42.8\text{MJ/kg}$

Actual Heat Release: The actual heat release is established as the difference between the enthalpy of the inlet air supply and the combustor exhaust gas as follows:

$$\begin{aligned}\text{Actual Heat Release} &= (\dot{m}_a + \dot{m}_f)Cp_t T_{t4} - \dot{m}_a Cp_c T_{t3} \\ &= 118\text{kJ}\end{aligned} \tag{C-2}$$

where $\dot{m}_a$ = air mass flow rate (0.36 kg/s), $\dot{m}_f$ = fuel mass flowrate $(7.603 \times 10^{-3}\text{kg/s})$, Cp_tT_{t4} = stagnation enthalpy at the combustor exit (1051 J/kg K $\times$ 578 K), and Cp_cT_{t3} = stagnation enthalpy at the combustor inlet (1004 J/kg K $\times$ 293 K).

Heat for Vaporization of Water: The heat required to vaporize the water in the combustor consisted of (i) the heat required to raise the temperature of the water from 20° to 100°C (335 KJ/kg), and (ii) the heat required to vaporize all of the water at 100°C (2257 kJ/kg). In all of the cases considered, the mass fraction of injected water, X_W, is 2.0 per cent, therefore, the heat required to vaporize the water is calculated as follows:

$$\begin{aligned}\text{Heat Required for Vaporization of Water} &= X_W \dot{m}_a (335 + 2257) \\ &= 18.8\text{kJ}\end{aligned} \tag{C-3}$$

Overall Efficiency, η_c: The overall efficiency was calculated by means of the following equation:

$$\eta_c = \frac{\text{Actual Heat Release} + \text{Heat for Vaporization of Water}}{\text{Ideal Heat Release}} \tag{C-4}$$

$$= \frac{118 + 18.8}{325}$$

$$= 42 \text{ per cent}$$

Estimated Mass of Fuel Burned: Finally, an estimate of the amount of fuel that could be expected to have undergone combustion is calculated as follows:

$$\begin{aligned} \text{Mass of Fuel Burned} &= \eta_c \dot{m}_f \\ &= 0.42 \times 7.6\text{g/s} \\ &= 3.2\text{g/s} \end{aligned} \tag{C-5}$$

APPENDIX D. EFFECTS OF COMBUSTOR EXIT TEMPERATURE AND PRESSURE ON ENGINE THRUST

Considering a simple jet engine with a single spool and a nozzle following the turbine driving the compressor (figure D.1), one can write

$$\frac{P_{O5}}{P_{O4}} = \left[1 - \frac{\Delta T_{O45}}{T_{O4}}\right]^{\frac{\gamma}{\gamma-1}} \tag{D-1}$$

and

$$\Delta T_{O45} = T_{O4}\left[1 - \left(\frac{P_{O5}}{P_{O4}}\right)^{\frac{\gamma-1}{\gamma}}\right], \tag{D-2}$$

where P_O and T_O correspond to the stagnation pressure and temperature, respectively. The subscripts 4 and 5 refer to the combustor and turbine exit stations, as indicated in Fig. D-1. the ratio of specific heats is denoted by γ.

Next, denoting the thrust by F, one can write

$$F = \dot{m}V \tag{D-3}$$

and

$$V^2 = 2C_pT_{O5}\left[1 - \left(\frac{P_{O6}}{P_{O5}}\right)^{\frac{\gamma-1}{\gamma}}\right], \tag{D-4}$$

where $\dot{m}$ is the air mass flow rate, and V is the air velocity at the nozzle exit.

Now, Eq. D-4 can be rewritten as follows using Eqns. D-1 and D-2, and Eqns. 2 and 4 in section 5:

$$V^2 = 2C_pT_{O3}\left[\left(1 + \frac{\Delta T_{O43}}{T_{O3}}\right) - \frac{\Delta T_{O45}}{T_{O3}}\right]\left[1 - \left(\frac{P_{O6}}{\left(1 - \frac{\Delta P_{O34}}{P_{O3}}\right) - \Delta P_{O45}}\right)^{\frac{\gamma-1}{\gamma}}\right] \tag{D-5}$$

As an illustration of the effects of a reduction in the combustor exit temperature and pressure on the engine thrust, one can consider a case of operation with constant air mass flow rate, constant engine rpm, and constant compressor pressure ratio.

Since the power required to drive the compressor is assumed to remain constant, a drop in the combustor exit temperature (T_{O4}) would lead to a corresponding drop in the turbine exit temperature (T_{O5}). Referring to Eq. (D-4) it can be seen that a drop in T_{O5} would lead to a reduction in the engine thrust. Concerning the two combustor performance parameters, namely $\Delta T_{O43}/T_{O3}$ and $\Delta P_{O34}/P_{O3}$, with reference to Eq. (D-5) it can be seen that a reduction in $\Delta T_{O43}/T_{O3}$, and an increase in $\Delta P_{O34}/P_{O3}$ will result in a reduction in the engine thrust. For the case of operation with an air-water mixture, it has been found that $\Delta T_{O43}/T_{O3}$ drops, and $\Delta P_{O34}/P_{O3}$ increases when compared to the case of operation with air only. Thus operation with an air-water mixture will result in a reduction in the engine thrust. It is worth pointing out here that operation with air enriched with oxygen (and no water injection) leads to an increase in both combustor performance parameters, which can be expected to result in a slightly higher value of thrust.

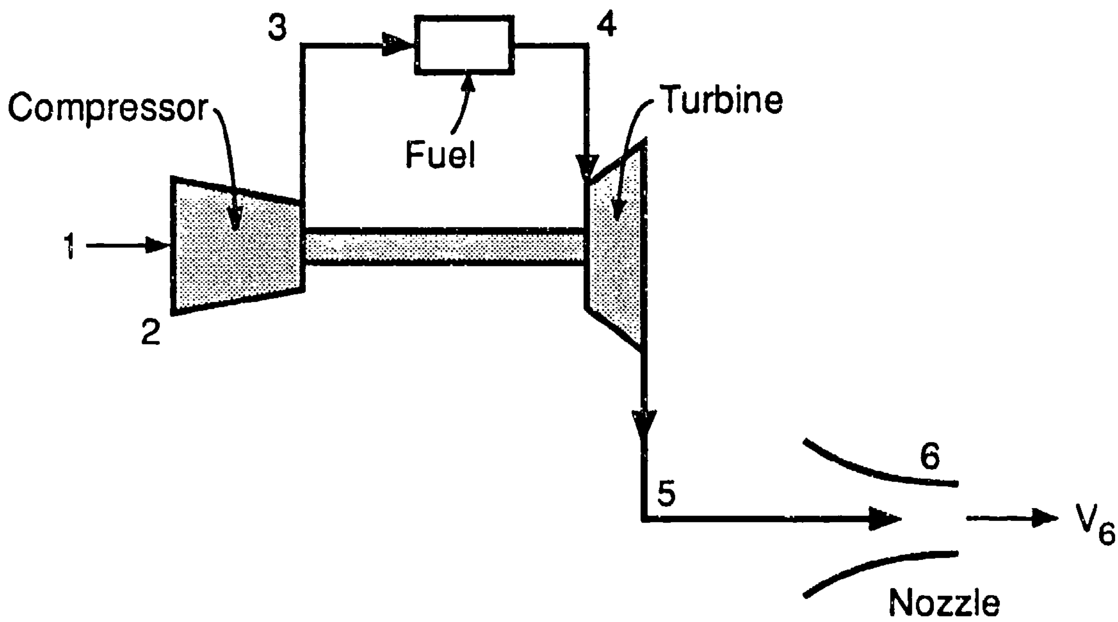


Figure D.1 Single Spool Jet Engine

*U.S. GOVERNMENT PRINTING OFFICE: 1994-604-081-00157