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INVESTIGATION OF COMBUSTION CHARACTERISTICS OF LOW-MEDIUM
HEATING VALUE GASES IN AN ATMOSPHERIC PRESSURE COMBUSTOR

by

© Maher Boshra KHALIL

B.Sc. (Honours) and M.Sc. (Mech. Eng.)

A thesis
submitted to the Faculty of Graduate Studies
in partial fulfillment of the requirements
for the degree of
Doctor of Philosophy

Faculty of Engineering
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The undersigned recommend to the Faculty of Graduate Studies acceptance of the thesis: "Investigation of Combustion Characteristics of Low-Medium Heating Value Gases in an Atmospheric Pressure Combustor," by Maher Boshra Khalil, in partial fulfillment of the requirements for the degree of Doctor of Philosophy in Mechanical Engineering.

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NOMENCLATURE

a	constant
A	area
A/F	air/fuel ratio (by mass)
b	parameter given by equation $b = 220(\sqrt{2} \ln \phi / 1.03)$; used in Eq. (6.3)
c	Discharge coefficient
C	ratio of activation energy to universal gas constant = E/R
D	combustor reference diameter, combustor diameter
d	combustor diameter at any section
D'	combustor constant
E	activation energy
f	friction factor
F/A	fuel/air ratio (by mass)
g	gravitational constant
G	momentum, thrust
k	constant
L	combustor length
m	mass, mass flow rate
M	molecular weight
m*	dimensionless compressor flow function
M _N	ratio of the mass velocity of the cooling air stream to the mass velocity of the mean stream

N air molal flow rate
 n^* ratio of inerts/oxygen in air (by volume).
 p, P pressure, operating pressure
 r radius
 r_c average radius of liner
 r_i inner radius of swirlers
 r_o outer radius of swirlers
 $\bar{R}$ universal gas constant, outer radius of swirler
 Re Reynolds' number
 s slot height
 S swirler number = $G_\phi / (G_x r)$
 S_N correlation parameter defined by Eq. (3.10)
 T Temperature
 u velocity
 u_c velocity of cooling air Eq. (3.11)
 u_m velocity of main stream Eq. (3.11)
 U velocity vector magnitude
 V combustor volume
 V_r radial velocity component
 V_z axial velocity component
 V_θ circumferential velocity component
 v' primary zone volume
 x distance, fraction of air extraction (from compressor)

- y distance measured from the lower wall at any section of the combustor
- Y parameter depends on the equivalence ratio
- z axial distance along combustor centerline measured from the swirler-injector piece

Greek Symbols

- β parameter defined by Eq. (6.9)
- ϵ fractional oxygen consumption
- ϵ effectiveness of film cooling = $(T_m - T_{s,ad}) / (T_m - T_c)$
- η combustor efficiency
- π $\log \log 100/\eta$
- θ vane angle (of swirler vanes), pitch angle of velocity vector
- ρ density
- σ factor for steam or water injection
- τ ratio of reaction time to residence time
- ϕ equivalence ratio, yaw angle of velocity vector
- ψ fuel loading = $m_f / v' P$

Subscripts

- a air
- A annulus
- ad adiabatic
- C cooling air, combustor, compressor
- E effective overall

f fuel
 F formula
 h open hole
 i inlet
 L liner
 L3 liner at exit section
 m mainstream
 n first stage of turbine nozzle
 o outlet
 r reference
 s static, slot
 S surface
 t stagnation
 T total
 x axial
 ϕ angular
 θ circumferential

Superscripts

. time rate
 * dimensionless parameter, normalized parameter
 n reaction order

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ABSTRACT

This study describes an experimental approach to determining the effect of various independent flow parameters on the performance of a test combustor burning gaseous fuel of low-medium heating value. The independent parameters the variation of which was studied include air mass flow rate, air distribution among the various zones of the combustor, air/fuel ratio, fuel heating value and the amount of swirl given to the primary combustion air (through changing the vane angle of the swirler). Of specific interest was the effect of these independent flow parameters on such dependent variables as temperature, velocity vector and gas composition. A test combustor was designed and fabricated specifically for the purpose of determining the influence of these independent parameters and the design procedure has been outlined. The combustor surface had access holes to facilitate measurements of the dependent flow parameters in the core of the flow. The air supplied to the combustor was admitted separately to the four longitudinal zones in order to permit tests that can determine the influence of the air distribution among the combustor zones. The low-medium heating value fuel was simulated by blending CO, H₂, and CO₂ from bottles. A range of heating values was obtained through the change of composition of the fuel. The combustion air was supplied through a blower driven by a variable speed DC motor.

Measurements of temperature, velocity vector and gas composition were taken to monitor the combustor performance over the range of operating conditions obtainable on the test apparatus. The temperature was measured with a suction pyrometer probe. The velocity vector was determined using a 5-hole probe with the aid of a 2-hole probe. The gas sample for composition analysis was drawn through the suction pyrometer into a container before injecting 1 cc into the gas analyzer (Fisher model 1200 gas partitioner). Plots to show results of these measurements and how they are affected by the independent parameters are given as well as a discussion of the results to interpret the configuration obtained. Analysis of these plots suggests several means for improving the performance of combustion equipment utilizing low-medium heating value fuel: A high proportion of combustion air should be

introduced in the secondary zone and the maximum possible air flow rate should be used; further, the appropriate swirl is needed to match the heating value of the fuel. A swirl equivalent to that obtained with a swirler having vane angle of about 45° is required with heating values in the range of 9.3-11.2 MJ/m³ (250-300 Btu/scf). A minimum permissible heating value of 6.62 MJ/m³ is necessary for stable combustion. The study also shows that the range of air/fuel ratios that could be ignited inside the combustor can be increased by reducing or eliminating the premixed air and reducing the swirler vane angle. However, the swirler vane angle should be maintained at 30° or more since otherwise stability problems will be encountered.

The results from the experimental testing regarding the combustor efficiency for the operating conditions used have been correlated in a global formula. This formula not only included such parameters as the Reynolds' number, mass flow rates and ratio of reaction time to residence time in the relations which have been correlated previously, but also includes such parameters as swirl, heating value and fuel/air ratio, which are missing from all other previously reported global formulae. The global formula predicts the efficiency at operating conditions different from those specifically used in developing the formula with an estimated accuracy of 4%.

The stability loops of the fuel in the test combustor have been determined experimentally and the effect of the swirler vane angle on their shape is presented and discussed. The shape factor of the temperature profile at the exit section was calculated and plots are given to show the effect of the independent parameters investigated in this study upon the profile of the shape factor.

CHAPTER 1

INTRODUCTION

The world's energy crisis and the fast depletion of oil and natural gas reserves make it necessary to develop other sources of energy. One important source is the huge amount of coal reserves in North America (about 31% of the total world's reserves [36]). Coal, however, in most cases contains about 1-5% sulfur by weight. Upon burning such coals, sulfur oxides will result with the consequent formation of sulfuric acid. These sulfur compounds pose a threat to all forms of life on this planet. While the most severely affected are the regions where combustion of coal takes place, the noxious fumes could be carried by the winds for miles and affect the neighbouring countries as is the case in North America and in Europe. Thus, one disadvantage of burning coal directly as a source of energy is that it causes environmental pollution. The other principal disadvantage is the large amount of combustion space and heat transfer area its equipment requires which in most cases involves a high capital cost. Fluidized bed combustion of coal would solve the pollution problem provided that the coal is used in a boiler. However, for producing power with gas turbines, it was realized that gasification of coal is the optimal solution for clean, efficient and economic energy production from coal [115]. This, however, is based on the assumption that a suitable combustor to burn the gas so produced from coal is available. Gasification of coal restricts the pollution effects to a smaller volume of gases which can be scrubbed clean more economically.

Through the years, many techniques have been developed for gasification of coal, but the chemical processes and reactions are very similar. Gasification of coal involves the introduction of coal (in a form and size depending on the technique to be used) in a stream of steam and oxygen (air could be used as a source of oxygen in some techniques). The chemical interaction of these materials results in what is termed "coal-derived" gas. The heating value and composition of the produced gas depend mainly on the proportions introduced

into the gasifier as well as the gasification technique employed [86].

The coal-derived gas which is considered in this work resembles the produced gas of the fluidized bed technique (so called Winkler gasifier). Typical heating values of gases from the oxygen-blown gasifier of this type is 9.3-18.6 MJ/m³ (~250-500 Btu/scf) compared to ~35.4 MJ/m³ (950 Btu/scf) for natural gas or CH₄. If air were to be substituted for oxygen, the resulting heating value would only be about 5.96-6.7 MJ/m³ (160-180 Btu/scf). For this reason, a range of heating values from 5.96-11.2 MJ/m³ (160-300 Btu/scf) will be dealt with in this work. This range is the so called low-medium heating value range [100] as compared to high heating value natural gas. The typical composition (by volume) of gas produced from a Winkler gasifier is 32.9% CO, 41.2% H₂, 20% CO₂, and 3% CH₄ if oxygen is used; whereas the composition when using air is 21.7% CO, 13% H₂, 6.9% CO₂, 0.6% CH₄, and 56.5% N₂ [101].

In this work, it was intended to study the combustion characteristics of the gas as produced by the oxygen blown Winkler gasifier. Due to the small proportion of CH₄ in this gas, and in order to simplify the analysis, its heating value was simulated by substituting CO and H₂ in adjusted proportions. The gas used was obtained by blending CO, H₂ and CO₂. Dilution of the gas to reduce the heating value was achieved through increasing the proportion of CO₂, which will be the situation if the gasifier is working on part load or inefficiently.

It should be noted that the results and conclusions of this research are not restricted to the gas produced from fluidized bed coal gasification. The gas composition and heating value dealt with in this study could result also from a variety of other sources. The gasification of wood or agriculture by-products would result in a gas similar to the one of concern to this study. Also, a similar composition gas is produced as a by-product during the coking of coal and during the processes involved in steel making. In the latter case, it is known as blast furnace gas.

1.1 General Overview of the Problem

If a new fuel is found, it does not imply that the energy crisis is solved. Since all combustion equipment has been developed and improved over the years to suit the application of a certain type of fuel, the combustion characteristics of any new fuel must be investigated. The different chemical composition and the consequent difference in heating value of low-medium heating value fuel as compared to fuels which have been in application for a long time (mainly natural gas, propane and city gas) would imply a certain variation of combustion characteristics. If this new gaseous fuel is to be burned in conventional combustion equipment, some changes to this equipment would be needed. For example, the lower heating value would require lower air/fuel ratio for the same output of energy (if the air flow rate is to be kept constant). A change of chemical composition would imply changes in ignition characteristics. This might also lead to differences in stability as well as in blowout characteristics [84].

1.2 Statement of the Problem

The problems to be dealt with in this study are related to the burning of a gaseous fuel in the low-medium heating value range (6.0-11.2 MJ/m³). These problems first arose during an attempt to couple a power generating gas turbine unit to a coal gasification unit [91,100], utilizing as a fuel a mixture of 23% CO, 14.9% H₂, 3.4% CH₄, 8.4% CO₂ and 50.3% N₂ blended from bottles to simulate the operation of an air blown fluidized bed gasifier. In later studies, Khalil and Plett [55] as well as Odgers [84] identified various problems expected with burning low-medium heating value fuel in equipment using conventional gaseous fuels due to the variation in composition and heating value. The following sections present an analysis of problems that would be encountered if low-medium heating value fuel replaced conventional fuels in combustion equipment.

1.2.1 Flame Length and its Emissivity

The length and emissivity of flames are very important characteristics in any combustion equipment. The cooling load is affected greatly by the flame emissivity in the case of a gas turbine combustor. The flame length can be related to the stability of combustion. It was found with jet flames that the flame length decreases and becomes "noisy" (increasing turbulence) before blowout and that the flame becomes bushy and long before it was lifted from the nozzle rim [63a]. The length of a flame is a clear indication of the degree of mixing between fuel and oxidant as well as the rate of fuel consumption. The mixing and therefore the flame length can be altered by changing the amount of swirl imparted to the combustion air [68]. The velocity at which the fuel is injected into the combustion space also influences the mixing and therefore the flame length [9]. On the other hand, the difference of flame length for low-medium heating value gas from that of a conventional gaseous fuel could be explained by their difference in flame speed. The flame speed of CO or H_2 , the main combustible constituents of low-medium heating value fuels, are higher than such gaseous constituents as CH_4 , C_2H_2 , C_3H_8 and C_4H_{10} which constitute the major proportions in most gaseous fuels. This implies that shorter flames should be expected when burning low-medium heating value fuel. However, as experienced by Shofstall [10], inefficient mixing and very low heating value (below 4.3 MJ/m^3 in the experiment) may result in a longer flame.

1.2.2 The Difficulty of Mixing Fuel and

Due to the lower energy density of low-medium heating value fuels compared to conventional gaseous fuels, it is estimated that for producing the same output energy, the volume required of this type of fuel will be 3-7 times that required if natural gas is used. This in turn means that the injector area must be enlarged accordingly to handle the increase in fuel flow rate if the fuel injection velocity is to be maintained the same. This increase of the injector diameter would necessitate enlargement of the combustor head in order to supply the necessary combustion air. The resulting larger primary

combustion zone diameter makes it more difficult to generate jets of combustion air strong enough to penetrate the core, for good mixing. This, in addition to the resulting change in ratio of fuel mass flow rate to air mass flow rate, which affects the fuel-air mixing in the combustion zone, will result in imposing mixing difficulty inside the combustor. The change in mass flow ratio will also affect the size and position of the recirculation eddy [9]. Stability and efficiency of combustion are strong functions of the degree of mixing and of the size and position of the recirculation eddy. In the recirculation zone the flame speed is matched by the flow speed, hence combustion stability is affected by the width and extent of the recirculation zone. Also, in the recirculation zone, the fresh mixture of fuel and air is heated by the recirculated products to prepare the fresh mixture for ignition and thus it affects the combustion efficiency.

1.2.3 Emission of Unburned Products in the Exhaust

The main combustible components of low-medium heating value gas are CO and H_2 . The oxidation rate of such components is much faster than the oxidation rate of conventional gaseous fuels (which causes them to produce, in most cases, shorter flames). Normally one would expect no emission of unburned products in the exhaust when burning low-medium heating value fuels. However, inadequate mixing may result in a longer flame or even in the emission of unburned products in the exhaust (mainly CO).

1.2.4 Formation of Nitrogen Oxides

The formation of NO_x in any combustion system, using air as a source of oxygen, depends on the maximum temperature inside the system, and the residence time at the maximum temperature [103]. NO_x can also be formed from fuel-bound nitrogen or ammonia in the case of coal-derived gas [29]. The ammonia could be formed in trace amounts during the gasification process and then carried with the produced gas to the combustion system. For constant air flow combustion equipment, the amount of fuel in the case of low-medium heating value fuel must increase (if compared with propane or natural gas) in

order to obtain the same average temperature at the exit section. Hence the air/fuel ratio in this equipment will be lower if low heating value fuel is used, which implies lower availability of nitrogen, although more than required to form undesirable amounts of NO_x if other conditions favor such formation. As for the maximum temperature inside the combustion space, it is expected to be lower, in general, when burning low-medium heating value gas. However, this may not be true if in order to improve the overall efficiency, the sensible heat of the gas leaving the gasifier is utilized by feeding the gas directly into the combustion space. In this case the resulting temperature might be higher.

1.2.5 Compatibility and Competition with Conventional Fuels

In order for a fuel to be considered acceptable, it must be able to replace other fuels in an emergency or be replaced by them without significant loss of performance and combustion efficiency. With the differences outlined above regarding the combustion characteristics of low-medium heating value gases, they will not be considered acceptable substitutes for other fuels until combustion systems are developed and improved to handle such a variety of fuels [84]. In terms of competition, low-medium heating value gas must show economic advantage or it will not find demand. Low-medium heating value gas is produced in small quantities as a by-product of certain processes (e.g. steel making) which means that it is available almost free of cost to be utilized at these sites. However, in the event that the gas is to be produced specifically to be used as a fuel, by a gasification process, the economics of the process must show an advantage over alternatives. The cost of electricity generated in a system burning low-medium heating value gas produced by gasification techniques is comparable to the cost of generating electricity via direct burning of coal or petroleum produced gas [24,86].

The above mentioned problems will mainly arise when low-medium heating value gas replaces natural gas (or any high heating value gas) in burners or combustors. These problems reveal the immediate need of investigations in

order to better understand the combustion characteristics of this new fuel. Such contributions would be valuable to developing combustion equipment to suit this gas or improving the existing ones.

1.3 Objectives of This Work

This work has the general objective of increasing the knowledge and information available for the application of low-medium heating value fuel. As revealed by the problems outlined in previous sections as well as those to be discussed in the literature review of the following chapter, the need is to further investigate the combustion characteristics of this gas. It was decided in this study to conduct the tests on a combustor whose dimensional proportions and configuration are typical of industrial gas turbine combustors. The development and design of the combustor used will be discussed in a later chapter. This combustor was designed to operate at a pressure of up to 5 atm. However, while it was initially hoped that the combustor would be connected to the bench scale fluidized bed gasifier in the same laboratory, trials to get steady flow and consistent composition of gas from the gasifier for a long enough period to carry out the needed combustor tests were not successful. At that stage, it was decided to use a gas mixture from bottles to simulate the expected composition of the gas produced from an oxygen-blown gasifier (neglecting the effect of CH_4 due to its small proportion but replacing its energy content by adding more CO and H_2). Inefficient operation of the gasifier or part load operation were simulated by increasing the proportion of CO_2 to reduce the heating value. The very high cost of equipment to provide high pressure air plus the cost of fuel (especially CO), made the study economically possible only if all the work were to be carried out at close to atmospheric pressure. To summarize, the objectives of this work are given as follows:

1. The development of a combustion facility suitable for the objectives of this work in terms of facilitating measurements of such dependent variables as temperature, velocity vector and gas composition as well as for the variation of independent parameters of importance to the

combustion process, such as air distribution among combustor zones, air flow rate, fuel flow rate (F/A ratio), fuel heating value and swirl of primary air.

2. Investigation of the operating conditions of the combustor required for satisfactory performance as measured by:
 - a. Stability of combustion.
 - b. Temperature profiles throughout the combustion space and at the exit section.
 - c. Profiles of the three velocity components in the combustion space.
 - d. Gas analysis.
 - e. Blowout(stability) loops.
3. To determine the required parameters (design features and gas properties) for stable combustion and the range of these parameters.
4. Comparison of this work with recent theoretical investigations.
5. To recommend future steps for research towards the goal of achieving a simple design procedure for building combustion facilities for low-medium heating value gas.

CHAPTER 2

REVIEW OF PREVIOUS WORK

Substantial effort has been devoted to the study of low-medium heating value fuels in North America and in other industrial nations due to the important influence that this type of fuel is bound to have upon the energy schemes in these countries [75].

The objectives of most of the previous work that has dealt with low-medium heating value gas, has been to determine the feasibility of its application in equipment that has been developed for conventional gaseous fuels. Through such feasibility studies relating to low-medium heating value fuel, combustion or performance problems have been identified and hence solutions for these problems or recommendations for improvements or modifications of the design of combustion equipment have been made. An obvious concern was the impact this alternate fuel would have on the environment. The desire to achieve electricity production economically and in a "clean" manner from coal was a major motivating factor for this type of research.

2.1 Review of Work Related to Utilizing Low-Medium Heating Value Gas in General

The interest in utilizing coal for power generation was the first motive for utilizing low-medium heating value gas. A popular concept was to use a high temperature gas turbine in combination with a heat recovery boiler to operate a steam turbine [60,78]. The approach was developed because of the need to produce electricity economically from coal without violating the emission laws. Interest in this concept was promoted by the continuous increase in allowable turbine inlet temperature which now approaches about 1650°C [78]. A full description of the whole combined cycle plant is given by Montgomery and Lemezis [78] together with the coal gasification and clean-up equipment. Control systems for matching the flows from appropriate points in the process and monitoring the entire plant are also described in the same

reference. At the time that paper was presented, the testing had not been started. The effect of integration of the gasification system into the combined power plant was estimated by Kydd [60] and was found to increase the gasification efficiency from 73% to 85.5% while increasing the overall cycle efficiency from 34.3% up to 43%. Kydd [60] also outlined a thermodynamic analysis to estimate the different component efficiencies as well as the total plant net efficiency.

The plant described by Ahner et al. [2] utilized a fixed bed pressurized Lurgi gasifier. The gas was then cooled in a shell and tube heat exchanger. The cooling water was recycled to recover the sensible heat contained in it while the gas was directed to the clean-up process to scrub off H_2S & NH_3 . Then the gas was burnt in a high temperature combustor that feeds to the high temperature gas turbine ($1650^{\circ}C$ inlet temperature). The exhaust from the gas turbine was directed to a heat recovery boiler which generates steam to drive a steam turbine. Various coals were considered for the plant. The performance of the integrated gasification plant was found to be comparable to nuclear or conventional coal-fired plants.

Recent events related to energy have led to some important observations; construction and licensing for nuclear reactors is at a virtual standstill, not only in the United States but also in Canada. The oil crisis has caused the price of oil to rise by a factor of ten in the last 10 years which implies a rise of more than 25% annually. This percentage increase surpasses the annual inflation rate which was less than 10% in the past ten years. Lehman et al. [63] have observed that steam plants which are oil-fired are now faced with either of two options: to shut down or to switch to coal which can only be done within the strict constraints of emission limits. According to Lehman et al., one alternative to this situation which can meet economic and environmental constraints is the "repowering" of existing plants. "Repowering" of plants could be described as follows: Through the use of coal gasifiers, fuel gas of low heating value can be scrubbed clean to meet environmental restrictions before burning it in a high temperature gas turbine combustor. The exhaust from the gas turbine will raise steam in a heat recovery boiler to run the steam turbines. It was estimated that there is a

gain of 50% in power upon the repowering of existing plants [63]. Several gasification techniques have been investigated and they all seem to be feasible for use in repowering oil-fired plants. Repowering was found to also increase the original overall thermal efficiency. The final overall thermal efficiency will mainly depend on the gasification technique used to provide low Btu gas.

Robson and Thurlaw [94] have carried out an experimental study on a combined plant unit with low heating value gas. The gasifier used was a fluidized bed gasifier with a complete gas clean up facility. The gas produced was mainly carbon monoxide, hydrogen and a small contribution of methane as the combustible components. In addition to the coal gasifier, a fluidized bed combustor was included in the plant. The steam to operate the steam turbine was generated in a heat exchanger that utilizes the energy in the exhaust from the gas turbine. The steam was then superheated in the fluidized bed combustor. Part of the exhaust from the fluidized bed combustor was mixed with the gas produced from the fluidized bed gasifier. After cleaning the gas, it was fed into a combustion chamber which provided the gas to run the gas turbine. The bed in the gasifier was fluidized with a mixture of steam and the remaining part of the exhaust from the fluidized bed coal combustor. The results of this work showed that production of a clean gas of low heating value suitable for gas turbine combustors is feasible and that the changes in the gas properties or composition with variation of operating parameters are tolerable for this application.

The work with a combined cycle plant by Davidson & Moore [27] showed very good success with low Btu gas (generally referred to as LBG) produced by gasification techniques. The fluidized bed gasifier was reported to result in a 0.5% increase in overall cycle efficiency compared with entrained or fixed bed gasifiers. Oxygen-blown gasifiers showed 1.5-2% lower overall cycle efficiency compared to air-blown gasifiers. A thermal efficiency of 40% or higher could be achieved with this combined cycle plant. This was almost the same range of efficiency achieved by Brown Boveri [89] utilizing fixed bed air-blown gasifiers.

The economic evaluation of power plants that utilize low-medium heating

value gas has been investigated by Corman and Fox [24]. The performance and economics were evaluated for seven energy conversion systems utilizing coal. The systems considered were: advanced steam (atmospheric fluidized bed coal combustor), advanced steam (pressurized fluidized bed coal combustion), closed cycle helium gas turbine, potassium topping cycle, combined cycle gas turbine (water cooled) burning coal-derived liquid fuel, combined cycle gas turbine (air cooled) with integrated low Btu gasifiers and open cycle MHD. The advanced steam cycles referred to in the study correspond to plants which were built from state-of-the-art components designed and fabricated to conform to all emission levels. The water used to cool the gas turbine in the combined cycle (water cooled) is used to clean the fuel of alkali metals and then to heat the fuel before introducing it to the burner. The cooling air used for the gas turbine in the combined cycle (air cooled) was used as a source of oxygen for the gasification process by mixing it with steam and feeding the mixture to the gasifier. The objective of the study was to have a consistent datum for the evaluation of these various systems. This was achieved by comparing their performance and economics to that of a reference steam power plant. All the plants were required to have thirty years useful service life and 90% plant availability. The results of this study were very encouraging to those interested in utilizing low heating value gas. The plants that gave the best overall efficiency as well as the lowest cost of electricity were those using an MHD cycle. The low Btu gasifier combined power plant was similar to pressurized fluidized bed combustion in terms of economics, capital cost and cycle efficiency. The other candidate plants did not compare favorably regarding either efficiency or cost of electricity. Ahner et al. [2] found that the cost of producing electricity from a combined plant burning LBG was lower than in the case of nuclear or conventional coal-fired plants.

2.2 Review of Previous Work Related to the Combustion of Low - Medium Heating Value Gas in Various Equipment

The problems to be dealt with upon switching to low-medium heating value gas from conventional fuels were outlined by Pillsbury et al. [91] based on an experimental investigation at Westinghouse Corp. and also by Schwieger [100].

The potential problems are related to flameout, low flame temperature, flame length, ignition, gas volume, mixing with air and compatibility with alternate conventional fuels. The problems related to the impact of burning low-medium heating value fuel on the environment in terms of pollutant emissions, will be discussed in the following section. Experimental tests carried out [91] on a full-scale and a half-scale combustor showed that a gas of heating value as low as 4 MJ/m^3 (108 Btu/scf) could be burned without flameout. Tests with heating value below 4.8 MJ/m^3 (130 Btu/scf) generally suffered from stability problems. However, a low heating value gas of 3.7 MJ/m^3 (100 Btu/scf) was successfully burned in a gas turbine combustor in another study by Crouch et al. [25]. The need was recognized for an oversized combustor with a rich primary zone and for heating of the fuel gas (if the heating value is less than 4 MJ/m^3) and for usage of auxiliary fuel in conjunction with fuel gases of very low heating value. For boiler applications, tests with low heating value gas [100] indicated the need for modification of the ignition system, installation of new fuel-gas piping system, replacement or enlargement of burners and windbox, uprating of forced and induced draft fans and modification of the convective heat transfer surfaces as well. Dual fuel burners were recommended in general for this type of fuel, that is, to start with natural gas or #2 distillate [5] and then gradually switch to low Btu gas either partially or fully.

The magnitudes of the minimum heating value of a fuel that can be used in a combustor were found to be at variance from one research to another. Battista and Farrell [5] have presented a plot of the heating value of fuel versus the combustor temperature rise. For conventional combustors, the minimum heating value from the plot was 2.07 MJ/m^3 (56 Btu/scf). However, this heating value is not suitable for use in gas turbines because it corresponds (in the study of Ref. [5]) to a condition where all the compressor air needs to be extracted for the purpose of gasification and where only fuel would be added in the combustor. For the purpose of high temperature stoichiometric combustion, the minimum heating value that could be practically burned was 3.33 MJ/m^3 (90 Btu/scf). Battista and Farrell [5] had also found through this development program that combustion stability with low Btu gas

depends on H_2/CO ratio as well as CO concentration. Two combustor designs were made and tested for low heating value fuel; one with constant cross-sectional area and the other with a sudden variation in the cross section after the primary zone. The performance of the combustors was evaluated with #2 distillate oil as well. The effects of combustor exit temperature and fuel heating value on the combustor performance were measured. The variable cross section combustor showed more flexible operation and it also gave satisfactory operation on # 2 distillate fuel.

In conjunction with the development of combustion facilities for high temperature turbines, some development work was attempted by Vogt [116] with a double walled sectoral combustor to utilize low Btu coal gas. The sectoral combustor considered in that study eliminated the transition ducting entirely by completely immersing the combustion system in the compressor discharge. The sectoral combustor was double walled to isolate thermal strains from pressure forces while the liner is cooled through the countercurrent heat exchange in the space between the double walls. The engine, in this case, was made of sixteen sectoral combustor chambers arranged in a closely packed circle around the turbine inlet annulus. In order to obtain satisfactory combustion performance of low Btu gas, modifications were made to the fuel handling system which needs to mix an unusually large volume of fuel having a very narrow flammable range. Two stage fuel injection was introduced to provide mixture variation from rich to lean to accommodate power or load demand. In the case of plants with air-blown gasifiers, the air needed for gasification was extracted from the compressor before the combustion system, with the amount of air extracted increasing with the increase of the fuel produced. This creates a paradoxical situation since increasing the amount of fuel produced decreases the amount of air available to burn it.

The other difference of burning LBG as compared to conventional fuels is in the sensible heat content of the fuel. While it is very negligible in the case of conventional fuels due to the small mass of fuel compared to air, the case is quite different upon switching to LBG. For very low heating value, the gas may need to be heated in order to achieve the required turbine inlet temperature. Also, in the case of LBG, the term "combustion margin" is of

prime importance. It is defined as the difference between the adiabatic cup-mixed mean gas temperature at the leading edge of the first rotating stage of the turbine, which is known as the firing temperature, and the maximum temperature the combustor can deliver to that point. In the case of gas turbines burning conventional fuels, this combustion margin is quite high, and hence it is of little concern in this case. However, for gas turbines burning LBG, this combustion margin will be very low especially with high temperature turbines.

Another criterion is the flammable range of LBG as compared to natural gas and oil. LBG (that contains 11-17% H_2 , 19-20% CO and more than 60% inerts, mainly N_2 and H_2O) will burn (as a mixture with air) in a narrower range compared to conventional oil and natural gas. It was found that a solution for the problem of combustion margin, burning limits and narrower allowable mixture ranges is to raise the fuel temperature before it enters the combustor. One other aspect is to have variable air flow splits through the burning zone which could be achieved by variable geometry hardware that opens or closes off all or part of the air flow entering the combustor and by staging fuel injection.

A similar effort to develop a high temperature gas turbine for operation with LBG was carried out by the Curtiss Wright Corporation in cooperation with the United States Department of Energy [56,120]. The objective was to develop a combustor having a wrap-around configuration which would burn LBG and deliver a high temperature gas of $1650^{\circ}C$. The combustor was to start at that temperature and sustain it for the period of operation. The performance of the combustor was found to be dependent on the fuel/air ratio. By heating the fuel (or utilizing the high temperature fuel from the gasifier) a lower fuel flow rate could be used, thereby freeing more air for film cooling of the combustor liner. In general, the combustor surface was made small to reduce the need for cooling. The fuel was introduced through a central fuel tube which allowed other types of fuel to be injected for start-up or emergency. The combustor had not been tested when the work was reported. A setup using simulated coal-derived gas was ready for testing during which the exit section of the combustor would be sampled and analyzed.

For more than 25 years, Brown Boveri have been designing gas turbine installations using blast furnace gas [90,122]. The gas turbines designed were equipped with a combined burner that would allow oil or gas to be burned. The burner allows the blast furnace gas to be mixed with air and admitted, with some swirl, at the head of the combustor while the fuel oil is centrally injected. In their experience, they achieved satisfactory operation of gas turbines utilizing gas with heating value as low as 3.2 MJ/m^3 (86.4 Btu/scf) and possibly even lower but with added auxiliary fuel or by preheating the fuel.

Results of work on a movable vane boiler burner utilizing low-medium heating value gas [101] showed some important findings. The tests were carried out on a variety of fuels that simulate those derived from different gasification techniques. The flame length was shorter as compared to that resulting from natural gas in all low-medium heating value fuels tested except when the heating value of the fuel was 4.3 MJ/m^3 (116 Btu/scf) and a swirler vane angle of 45° was used. This heating value was the lowest among fuels tested and resulted in a longer flame compared to the flame from natural gas. It was found that there is an optimum velocity of 30 m/s for fuel injection to obtain a stable flame. Similar results were reported for a gas turbine combustor for which a fuel injection velocity range of 45-120 m/s was found necessary for stable burning of low heating value fuel [25]. The emissivity of the flames resulting from burning the various fuels, produced from coal gasification techniques, showed very little variation compared to one another or with respect to flames produced from natural gas.

Low heating value gas may be the by-product of some industry or process. Investigations have been carried out to enhance the ignition characteristics of such fuels and to obtain stable combustion [11,114]. In these attempts, gas of heating value as low as 1.4 MJ/m^3 (37.8 Btu/scf) was successfully burnt through the use of a cross wind flow or cyclone combustor.

2.3 Review of Work Related to the Impact of Low-Medium Heating Value Gas on the Environment

When dealing with pollutants from any combustion system, the emphasis is directed to four major components, which are: sulfur oxides, nitrogen oxides, carbon monoxide and unburned hydrocarbons. In most cases, also included is soot or carbon particles recognized as dense smoke, but for the present case only the first four components are of concern.

Experimental tests by Pillsbury et al. [91] on natural gas combustors burning low-medium heating value gases showed that the concentration of CO emission at the exit section tends to rise with the decrease of combustor exit temperature. The emission of CO increased with increased temperature at the combustor exit to a peak at a temperature of 1000°K and then began to drop with further temperature increase. In contrast, Battista and Farrell [5] found in two different combustors that a drop of CO emission occurred with increase of exit temperature. These investigators [5], also found that the CO emissions were higher for a constant cross section combustor than for a combustor of variable cross section. In the same study [5], it was found that CO emissions increased for both combustors as the fuel heating value decreased. CO emissions were found to be largely affected by cycle pressure. Higher pressures resulted in higher CO emissions. The concentration of CO at thermodynamic equilibrium increases with exit temperature (as the mixture becomes richer in fuel) and reaches a peak at stoichiometric mixture and then falls again as the fuel content continues to rise in the mixture [100]. Equilibrium CO concentration is the lowest value achievable while real values could be higher and the observed behaviour may be quite different from that under conditions of thermodynamic equilibrium. It was found by Schiefer and Sullivan [95] that a higher H_2/CO ratio in the fuel would result in a reduction of the emission of CO. Also, the increase of residence time, by intensifying the recirculation zone would result in lower CO emission. Crouch et al. [25] have suggested that the emissions of CO could be brought to the limits obtained by firing natural gas through modifications to the air distribution and the burner design.

It is important to mention here that in practice, the CO emissions were

found to be much higher than predicted from equilibrium calculations and also behaved differently. Equilibrium calculations indicate that CO emission will rise with temperature. However, as was outlined in this section, CO emissions were noticed to decrease with the increased exit temperature or reach peak value with increasing temperature and then decrease upon further increase of temperature. This may be caused by one or more of the following [62a]:

- a) Inadequate burning rates in the primary zone due to low fuel/air ratio or insufficient residence time (for low exit temperature).
- b) Inadequate mixing of fuel and air at lean fuel/air ratios which produces some regions in which the mixture strength is too weak to support combustion and others in which overrich combustion yields high local concentration of CO.
- c) Quenching of the post flame products by entrainment with the liner wall cooling air.

The pollutant which has received most attention is the general category of nitrogen oxides. It should be expected that lower NO_x emission will be generally obtained with the burning of low-medium heating value gas due to the lower flame temperatures which result in this case. However, other factors may interfere and cause this situation to be changed. Since the concentration of nitrogen in the fuel is higher for low heating value gas, this may lead to the formation of nitrogen oxides early in the combustion stage. These gases remain stable until combustor exit due to the near stoichiometric combustion of LBG. The other factor is the possibility of ammonia formation in the gasifier which would also remain with the gas leaving the gasifier, and this could react to form nitrogen oxides during combustion. The experimental results at Westinghouse [91,123] showed lower NO_x from all low-medium heating value gases tested in comparison to natural gas. The expected trend of increasing NO_x emission with combustor exit temperature was also clear in all tests. A comparative study of NO_x formation due to the combustion of a variety of fuels in a gas turbine combustor has been outlined by Hung [47] utilizing a kinetic diffusion limited model proposed in an earlier study [45].

The NO_x emission values were averaged for each fuel compared to # 2 distillate oil. Results concerning coal-derived gases indicated NO_x emission levels ranging from 0-1.1 times the emission levels from # 2 distillate oil, depending on the gasification technique (or heating value). It was also pointed out that while a concentration of 100 ppm bound nitrogen in liquid fuel would have negligible effect on NO_x emissions, 100 ppm of ammonia in a 7.46 MJ/m^3 (200 Btu/scf) gaseous fuel can have a significant effect due to the larger quantity of fuel required when burning low Btu gas. Hung [46] also found higher NO_x emissions from the burning of medium heating value gases that were obtained from oxygen-blown gasifiers compared to low heating value gases. This result was in agreement with the findings of Shoffstall [101] who studied combustion of a variety of low-medium heating value fuels in a variable vane boiler burner. The resulting NO_x concentration was higher than from burning natural gas for all medium heating value gases obtained from oxygen-blown gasifiers [101]. The modelling work of Heap et al. [41] on NO_x emission for burning low Btu gas showed some interesting results. This model, was based on the kinetics of 100 reactions in addition to some empirical constants and data from previous work. The model also included the effect of presence of ammonia in the fuel. The results showed that NO_x emission dominates for a lean mixture with a peak at an equivalence ratio of 0.9. The NO_x emission was found to decrease continuously with the increase in equivalence ratio after this peak to reach a minimum at a rich mixture of 1.4 equivalence ratio before it slowly rises again. The NO_x emission was found to decrease with the increase of residence time for LBG fuels containing ammonia. The work recommended that a rich primary zone should be used for low Btu gas with a residence time of 200 ms at least, while gradual air distribution and mixing should follow downstream. This concept was affirmed by Folsom et al. [31] who outlined the effect of LBG composition on NO_x emissions resulting from burning LBG in three laboratory-scale atmospheric pressure flame reactors. The tests showed that hydrocarbon concentrations in LBG had a significant effect on NO_x formation from NH_3 while CO/H_2 ratio was of minimal influence. Further testing on bench scale reactors [22] showed a slight effect of fuel hydrocarbons on NO_x emission. Increased swirl also reduced NO_x emission while

pressure and Reynolds' number had minimal effects. Corman and Fox [24] conducted a comparative analysis on different power plants utilizing coal and showed that lower NO_x emissions occurred when burning LBG than with coal or liquid fuel produced from coal liquefaction. The study by Davidson & Moore [27] suggested that the fuel gas temperature should not exceed 538°C at the entrance to the combustor in order to limit NO_x emissions.

The emissions of sulfur oxides when burning low-medium heating value gas have not been considered by most investigators, probably due to the belief that sulfur can be removed either during the gasification process or by scrubbing the produced gas. The study of Corman & Fox [24] showed, however, that slightly higher emissions of sulfur oxides resulted with LBG combustion than with other power plants using coal directly or a liquid fuel produced from coal.

The emission of unburned hydrocarbons when burning low-medium heating value gas, also has received very little attention. This may be due to the small proportion of hydrocarbons in the composition of low-medium heating value gas produced from most gasification techniques (see Table 4.1). The tests of Pillsbury et al. [91] showed negligible hydrocarbon emission during coal gas combustion in all cases except near flameout cases. Emissions were lower for full-scale combustors than for small scale units. The tests of Battista & Farrell [5] showed that unburned hydrocarbon emission follows the same trend as CO emission in that increase of the combustor exit temperature decreases the emission levels of hydrocarbons.

In summary, this literature survey has revealed that many investigations have been made on the combustion or the utilization of low-medium heating value gas, but little attention has been given to the combustion characteristics inside the combustion space and the influence of design or operating parameters on these characteristics. While a few attempts have been made to modify the combustion equipment to suit the combustion of low-medium heating value gas, none have investigated the change in properties occurring inside the combustion space due to the modifications or changes in operating or design parameters. The main objectives of those studies was to determine the the feasibility of using low Btu gases. If the information about the

combustion space and the effect of operating or design parameter variations is not determined, the optimum solution for overcoming any problem particularly those due to load variation or variation of heating value may never be obtained. Measurements at the exit section may indicate the suitability of the high temperature gases for one application or another but they are not capable of describing what could be achieved through modifications or which modifications would be most effective.

In this work, a test combustor suitable for research relating to the combustion of fuels especially in the low-medium heating value range was designed and fabricated. The test combustor is equipped with measuring holes which give access to the combustion space. By means of various measurements, a complete picture can be obtained of the effect of varying operating or design parameters such as air mass flow rate, heating value of fuel, amount of swirl given to the primary combustion air or air distribution among the various zones of the combustor. Also given are results of the measurements taken, with discussion of how they relate to the application of low - medium Btu gas in various combustion equipment. It is expected that the conclusions and recommendations of this work will provide both insight and direction regarding the utilization and combustion of low-medium heating value gas which clearly is of great practical importance.

CHAPTER 3

DEVELOPMENT OF A TEST COMBUSTOR

The need to carry out experimental testing inside combustors which provide access to the combustion space was explained in the earlier chapter. It was therefore desired to fabricate a combustor that would satisfy this requirement and also facilitate the variation of operating parameters to determine their effect on the performance of the combustor. Access to the combustion space was made possible by the provision of holes in the combustor walls through which measuring probes could be inserted at right angles to the combustor axis at numerous axial locations. The combustor liner was divided into separate compartments to allow the total combustion air to be allocated with varying proportions along the combustor as desired. The air flow rate to each compartment was metered. A burner-swirler combination was designed to allow the primary air to gain some swirl where the fuel gas was admitted. The burner and its attachments were designed in such a way that would also allow the use of a separate inlet line to introduce another fuel, either gas or oil (through an injector), at the center of the burner. It was thought that this might be needed in the case of very low heating value fuels or when a starting fuel is used in real systems. This would allow the combustor to be integrated with a gasifier producing the gas on-line from a primary fuel such as coal.

In sizing this combustor, certain factors were considered to be of prime importance. For example, the combustor should be large enough so that insertion of probes for various measurements inside the combustor will not disturb the flow appreciably. At the same time, the combustor should not be so large that the air and fuel requirements cannot be economically met.

The combustor was designed to operate at a pressure of 413.5 kPa (60 psi); however, due to limitations in available equipment and funding, tests were carried out at near atmospheric pressure using an air blower to provide the air necessary for combustion (more details about the test rig are given in a later chapter).

The function of any combustor, according to gas turbine manufacturers, is to provide a high temperature gas stream which has temperature and velocity profiles suitable for turbine design. The combustion efficiency should be close to 100% with minimum pressure drop (less than 5%). The combustor should provide stable operation over a wide range of conditions and be able to handle a variety of fuels. Further, due to recent awareness of pollution problems, emissions of the various pollutants should be within acceptable limits at all operating conditions.

The steps that were taken for sizing this combustor had followed the approach outlined by Hazard [40]. The procedure that was followed in sizing this test combustor, together with the various verifications of performance wherever possible, or applicable, is given in the next sections. Recommendations or experiences outlined in earlier work relating to utilization of low-medium heating value gas [5,25,54,56,75,90,122] were taken into consideration wherever possible. A summary of the approach outlined here has been given in an earlier presentation [55].

3.1 Design Parameters of the Combustor

3.1.1 Material

Materials suitable for combustors have been continuously developed and many are now available which would give the required life and performance for most applications. In the case of a test combustor, the life duration is of relatively little concern and in this type of application, stainless steel (AISI 304) provides acceptable performance and life. The stainless steel sheets should be thick enough to allow welding by conventional techniques but it should be noted that excessive thickness can promote fatigue failure [48].

3.1.2 Reference Velocity

The reference velocity, in combustor design, is the mean velocity across the plane of maximum cross-sectional area of the casing, in the absence of a liner, for the flow of combustor inlet air. Upon referring to different

designs of industrial gas turbine combustors [40,103], it was found that the value for the reference velocity ranges between 6.56-30.5 m/s (21.5-100 ft/s). The higher the reference velocity, the higher the pressure losses in the combustor will be. Fig. 3.1, taken from [40], shows the relation of minimum theoretical pressure drop, in percentage of static pressure, to reference velocity based on a total combustor loss coefficient ($\Delta P / \frac{1}{2} \rho v_{ref}^2$) of 15. A value of 15 m/s (50 ft/s) was chosen for this combustor, which corresponds to a minimum static pressure loss of 2%. Lower pressure loss (less than 2%) causes severe asymmetric flow distribution.

3.1.3 Diameter of Outer Case

In practical combustors, this is fixed from the volumetric air flow rate of the compressor and the reference velocity as chosen in step 3.1.2. However, in this case, the combustor size must be adequate to allow the introduction of probes for measurements without severe disturbance of flow. At the same time, it is limited by the laboratory air supply and space. Generally, the combustor size must be as small as possible to save on fuel consumption and also to permit the research program to be conducted within a reasonable cost. The maximum flow rate from the air blower that was available for this research was 1857 m³/hr. However, the amount that could actually be obtained was much less as a result of reducing the exit area of the blower. Tests to calculate the flow rate with a flange containing ten 12.7 mm diameter holes to carry the air to the various inlets to the combustor showed that only about 35% of the maximum air flow from the blower could be obtained. At a reference velocity of 15 m/s and 0.35×1857 m³/h (maximum air flow rate), the diameter of the combustor was calculated to be 12.4 cm. This diameter was approximated to 12.7 cm (5 in.) outside diameter to allow for wall thickness (1/16 in). This case diameter or outer diameter is also known as reference diameter and the cross section area based on this diameter is the reference area.

3.1.4 The Liner Diameter

Grobman et al. [37], used a tubular combustor with constant annulus cross section and applied incompressible cold flow relations to obtain the effects of the ratio of the annulus area to reference area on pressure losses, as presented in Fig. 3.2. The figure indicates clearly that the ratio of liner area to reference area should be about 0.5-0.6 for minimum pressure losses. In this design, a value of 88.9 mm (3.5 in.) was chosen to be the average liner diameter as estimated from the reference diameter and the optimum ratio of annulus area to reference area. In order to allow for film cooling of the liner, the diameter was stepped up between sections in the axial direction using thin rings (Fig. 3.3). These rings contained 24 axial holes around their periphery which would allow a portion of the air (in the compartment between the liner and outer case of the combustor) to flow tangentially over the inner liner surface..

3.1.5 The Combustor Length

The combustor length affects two important performance aspects. If too short, it will not permit sufficient time for mixing and hence, the combustion efficiency will be degraded and the exit section profiles will not be fully developed. If too long, the pressure losses will be excessive. Therefore, short combustors must provide some means to enhance mixing and stabilize the flow. Based on observations of many previously developed combustors, the lengths of the various zones in the present combustor were chosen as follows. The liner has a primary zone (which extends from the swirler exit plane to the plane of the first set of secondary air admission holes) of a length equal to the air inlet diameter permitting an undisturbed recirculation zone which assists in flame stabilization and mixing. The secondary zone has a length equal to the liner inlet diameter. In order to avoid any flame chilling or reaction quenching, a length of about the maximum liner diameter was allowed between secondary and dilution zones without any additional air admission. The dilution zone follows and extends for a length of about one diameter (reference diameter). Between this dilution zone and the combustor exit is a

section to stabilize the flow and to create a suitable temperature profile. The combustor length (flame tube) totals 431.8 mm (17 in.). The resulting length/outer diameter of the combustor (3.4) falls within the range for industrial combustors as given by Hazard [40] to be 2.1-5.8. The liner volume is 2.3 l (140 in³) and the average cross-sectional area is 53.3 cm² (8.26 in²). Test probe locations are distributed along the combustor length (Figs. 3.3 & 3.4).

3.1.6 The Air Admission Holes

Fig. 3.2 indicates that the pressure drop in the combustor depends upon the ratio of the area of holes used for air admission to the reference area. Fig. 3.5 (taken from Hazard [40]) shows that the discharge coefficient for plain holes increases with the increase in velocity ratio (admission velocity/annulus velocity). Hence, combining the two figures for the lowest pressure drop and largest discharge coefficient, the ratio of hole area to the reference area should be about 0.6. This hole area was interpreted in terms of a set of holes, each having 12.7 mm (0.5 in.) diameter as indicated in Fig. 3.3. The air admission holes were simply drilled normal to the combustor axis. This was found by Odgers [82] to give a high discharge coefficient for the holes and the largest possible air penetration which leads to better mixing. Fig. 3.3 shows the details of the liner assembly isolated from the combustor while Fig. 3.4 is a complete longitudinal cross section of the combustor assembly.

3.1.7 Swirlers

One efficient method to create a recirculation zone at the combustor inlet is by imparting some swirl to the combustion air. The importance of a recirculation zone and swirl of combustion air have been investigated by many researchers [15,57,65,97,98,99]. Swirl is believed to give stable combustion, delays blowout and is a means to control the formation of some pollutant species during combustion. The swirlers for this combustor were made as one integrated part with the fuel-air mixture injector. Four swirlers were made.

with each having a different blade angle (30° , 45° and 60°) and the fourth swirler containing only straight axial blades (0° swirler). This allowed investigation of the swirl needed for best performance. For a given swirler, a low-medium heating value gas will result in a shorter flame compared to a high heating value gas [101]. Hence, variation of swirl may be needed to determine the optimum swirl with this new fuel. Fig. 3.6 shows a schematic drawing of the swirler-injector piece. It should be mentioned here that a fine mesh screen made of stainless steel was fitted inside the injector passage that carries the fuel-air mixture in order to prevent the flame, in case of a flashback, from travelling back to the gas bottles.

3.1.8 Fuel Injection

The fuel, being a medium (or low) heating value gas, needs special care and attention to ensure a high mixing rate with air. In the fuel injection system designed for this combustor, the fuel is allowed to mix with part of the combustion air before injection into the combustion space. The fuel issues from the injector in such a way as to meet the primary air coming from the swirler at 90° . This accelerates mixing of fuel and air utilizing lateral shear stresses [9]. Also, this arrangement allows the introduction of higher volumes of fuel with respect to air at the combustor head. With this arrangement, mixing the fuel with another fuel is possible if needed. This arrangement and its details represent the experience of all the previous investigators in the design of fuel injectors for low-medium heating value gas [89,90,122].

3.1.9 Ignition

The fuel was ignited using a standard spark plug which had shown satisfactory performance in another bench scale combustor. The spark plug was placed in the combustor in such a way that its spark gap was at the tangent

point of a circle drawn to touch the combustor centerline, the liner inclined surface and the liner's first segment*. The spark was powered by a high voltage electric transformer (20,000 volts).

3.2 Parameters Related to the Performance and Effectiveness of the Test Combustor

Having decided upon the dimensions of the combustor, it is important to check and verify that the performance of the combustor will be adequate. The combustion intensity, cooling effectiveness as well as the pressure drop were estimated for the operating conditions of the combustor, as outlined in the following sections.

3.2.1 Combustion Intensity

The combustion intensity is a quantity introduced for the design of combustors. It includes the effect of the combustor dimensions, pressure and the heat release rate, the latter as an indicator of the temperature inside the combustor. The combustion intensity is related to the rate of reaction in the combustor. Two definitions are generally used for combustion intensity. One is the volumetric combustion intensity obtained from the heat release rate divided by the product of the liner volume and pressure. The second definition of combustion intensity is given by the ratio of the heat release rate to the product of the liner cross-sectional area and pressure. These values should be kept within a range of $2.98 \times 10^{10} - 3.73 \times 10^{10} \text{ J/m}^3 \text{ hr atm}$ ($0.8 - 1.0 \text{ MM Btu/ft}^3 \text{ hr atm}$), for the volumetric combustion intensity, and $1.02 \times 10^{10} - 1.36 \times 10^{10} \text{ J/m}^2 \text{ hr atm}$ ($0.9 - 11.2 \text{ MM Btu/ft}^2 \text{ hr atm}$), for the combustion intensity based on liner cross-sectional area. However, these values could reach as high as $30.2 \times 10^{10} \text{ J/m}^3 \text{ hr atm}$ ($8.1 \text{ MM Btu/ft}^3 \text{ hr atm}$) or $8.4 \times 10^{10} \text{ J/m}^2 \text{ hr atm}$ ($7.4 \text{ MM Btu/ft}^2 \text{ hr atm}$) for short combustors [23,40].

For the test combustor of this study, values of air and fuel flow rate were estimated based on reference velocity, outer case diameter and the

* Based on private discussion with Prof. J. Odgers of Laval University, Que., Canada.

temperature rise required across the combustor. Fig. 3.7 obtained from Merrill [74] shows the flow diagram of the ANJ1 two-shaft industrial gas turbine manufactured by Westinghouse Canada Limited. The temperature at the compressor exit was given as 540°K (971.8°R) and the temperature from the combustor at the turbine inlet was given as 1056°K (1900°R), the inlet to the heat exchanger was 711°K (1280.3°R). If a heat exchanger effectiveness of 80% is assumed, the temperature at the combustor inlet could be estimated to be 621°K (corresponding to 610°K at exit from heat exchanger). This results in a temperature rise across the combustor of 435°K . This is the temperature rise that was utilized to calculate the required flow rate of fuel into the combustor and also the temperature rise that was achieved during the experimental work. Based on the estimated fuel flow rate and the dimensions of the combustor, the combustion intensities are calculated to be:

$$\begin{aligned} \text{volumetric combustion intensity} &= \text{heat release rate /} \\ &\quad (\text{liner volume} \times \text{pressure}) \quad (3.1) \\ &= 3.55 \times 10^9 \text{ J/m}^3 \text{ hr atm} \\ &\quad \text{or } 9.54 \times 10^4 \text{ Btu/ft}^3 \text{ hr atm} \end{aligned}$$

$$\begin{aligned} \text{cross-sectional combustion intensity} &= \text{heat release rate /} \\ &\quad (\text{liner cross-sectional area} \times \text{pressure}) \quad (3.2) \\ &= 1.53 \times 10^9 \text{ J/m}^2 \text{ hr atm} \\ &\quad \text{or } 1.35 \times 10^5 \text{ Btu/ft}^2 \text{ hr atm} \end{aligned}$$

The above values were estimated based on a combustor operating pressure of 413.5 kPa (60 psi). This is the value on which the combustor parts' mechanical design was based. If the combustor is to operate at atmospheric pressure and produce the same temperature rise, the above values would become $1.45 \times 10^{10} \text{ J/m}^3 \text{ hr atm}$ ($3.9 \times 10^5 \text{ Btu/ft}^3 \text{ hr atm}$) and $6.24 \times 10^9 \text{ J/m}^2 \text{ hr atm}$ ($5.5 \times 10^5 \text{ Btu/ft}^2 \text{ hr atm}$) respectively. Utilizing the fuel flow rates and the heating value used during the experimental investigation showed that the combustion intensities during the operation of the combustor were very close

to those estimated above (operating volumetric combustion intensity was $\leq 1.57 \times 10^{10}$ J/m³ hr atm and operating combustion intensity based on cross-sectional area of liner was $\leq 6.80 \times 10^9$ J/h m² atm).

3.2.2 Pressure Loss

Pressure loss has an important role in both combustion stability and efficiency of the gas turbine, downstream of the combustor. Since every attempt has been made during the design stage to reduce pressure losses and keep them at minimum values, it was expected that the pressure drop in this combustor will be within acceptable limits. Three methods to calculate the pressure drop are presented as follows, and applied for the case of this combustor.

3.2.2.1 Compressor, Combustor and First Row of Turbine Nozzle Analysis

This procedure is based on a study on gas turbine engines by Sullivan [110]. He considered a model in which the compressor, combustor, and first row of turbine nozzles are one unit. The following non-dimensional compressor mass flow function was defined as:

$$m^* \triangleq \frac{m_c}{p_1 A_n} \sqrt{\frac{\bar{R} T_1}{M_a g}} \quad (3.3)$$

where,

- m^* = dimensionless compressor flow function
- m_c = mass flow rate
- p_1 = pressure at compressor inlet
- A_n = area at first stage turbine nozzle
- $\bar{R}$ = universal gas constant
- M_a = molecular weight of air
- g = gravitational constant

The pressure drop inside the combustor is then given by:

$$\frac{\Delta P}{P_2} = \frac{\sigma}{2} \frac{T_2}{T_1} \left[\frac{(1-x)m^*}{\frac{A_1}{A_n} \frac{P_2}{P_1}} \right]^2 \quad (3.4)$$

where, σ = factor for steam or water injection
 (=1 for no injection)
 T_2 = air temperature at compressor exit
 T_1 = air temperature at compressor inlet
 x = fraction of air extraction

Applying the values of this combustor, $m^* = 0.6273$ and the resulting pressure drop $\Delta P/P = 1.7\%$ when operating pressure is 413.5 kPa (60 psi). However, if the combustor is to operate at near atmospheric pressure, the resulting values would be $m^* = 0.241$ and $\Delta P/P = 2.9\%$.

3.2.2.2 A Formula Based on Flow Analysis

This analysis is based on an investigation carried out by Williams [119] on aircraft engines. It is given in empirical form as follows:

$$\Delta P = \frac{\dot{m}_a^2}{2g\rho A_{eff}^2} \quad (3.5)$$

where, $\dot{m}_a$ = mass of air flow rate
 ρ = air inlet density
 A_{eff} = effective area
 g = gravitational constant

Williams outlined a procedure to calculate the effective area of the combustor which is similar to adding electrical conductances in parallel. By applying this procedure to the configuration of this combustor, the effective area was calculated to be 30.91 cm^2 (4.7913 in^2). Substituting this value of A_{eff} in equation (3.5), yields a pressure drop of $\Delta P = 5.66 \text{ kPa}$ (0.8215 lb/in^2) or $\Delta P/P = 1.36\%$ of combustor operating inlet pressure of 413.5 kPa (60 psi). When the combustor operates at atmospheric pressure, the above value becomes $\Delta P/P = 2.1\%$.

3.2.2.3 Tubular Turbojet Combustion Analysis

This study was carried out by Grobman et al. [37] using a tubular

turbojet combustor at NACA and is based on fluid mechanics laws and discharge coefficients. One dimensional flow was assumed, ignoring any mixing loss or delay of the jet flow and the main stream. The result is given in the form of two equations to be solved simultaneously to obtain the combustor pressure drop. The equations are given as follows:

$$\frac{A_{hT}}{A_r} = \frac{m_L}{m_T} \int_C^1 \frac{d\left(\frac{m_L}{m_T}\right)}{C \sqrt{\frac{P_A - P_L}{q_r}}} \quad (3.6)$$

$$\frac{P_A - P_L}{q_r} = \frac{P_1 - P_{L3}}{q_r} - \frac{P_1 - P_A}{q_r} - \left(\frac{A_r}{A_L}\right)^3 \left[\frac{T_{L3}}{T_1} - 2 \left(\frac{T_L}{T_1}\right) \left(\frac{m_L}{m_T}\right)^2 \right] + \left(\frac{A_r}{A_L}\right) \left(\frac{A_r}{A_A}\right) \left(1 - \frac{m_L}{m_T}\right)^2 \quad (3.7)$$

where,

- P_A = annulus pressure
- P_1 = combustion inlet pressure
- P_L = liner pressure
- P_{L3} = liner pressure at exit section
- A_{hT} = total liner open hole area
- A_r = reference area
- A_L = liner cross-sectional area
- A_A = annulus cross-sectional area
(area between case and liner)
- T_1 = combustion inlet temperature
- T_{L3} = combustor exit temperature
- m_L / m_T = fraction of air admitted at combustor inlet
- m_L = weight of air flow rate through liner
- m_T = total weight of air flow rate passes through combustor

$C =$ discharge coefficient ~ 0.4

The annulus total pressure loss term $(P_1 - P_A)/q_r$ could be calculated from the relation

$$\frac{P_1 - P_A}{q_r} = \frac{4f \frac{L}{D}}{\left(\frac{A_A}{A_r}\right)^2 \left[1 - \left(\frac{A_A}{A_r}\right)^2 \right]} \int_0^1 \left(1 - \frac{m_L}{m_T}\right)^2 d\left(\frac{x}{L}\right) \quad (3.8)$$

An approximate value for $m_L/m_T = (x/L)^2$ if the air fraction entering at the combustor head is neglected.

where, x = distance measured from combustor head

L = combustor length

f = friction factor = 0.005

Substituting and performing the integration of Eq. (3.8) gives $(P_1 - P_A)/q_r = 0.488$. If it is assumed that $C\sqrt{(P_A - P_L)/q_r}$ does not vary with m_L/m_T , which is a reasonable assumption for this combustor as the flow is divided at the secondary and dilution zones and admitted separately, the integration of Eq. (3.8) gives $(P_A - P_L)/q_r = 39.06$. Substituting into Eq. (3.7) results in the combustor's total pressure loss being computed to be $(P_1 - P_{L3})/q_r = 37.43$ or $\Delta P/P = 3\%$

The results obtained from the three methods are in good agreement but the last method required more assumptions to arrive at this range. However, the results show that the pressure drop in this combustor would probably be within the acceptable range.

3.2.3 Film Cooling of the Combustor Liner

Film cooling of combustor liners is a means by which the heat transferred to the liner by radiation from the high temperature gases is removed by convection. Cooling of liners is important to extend their life as well as to avoid any deformation therein due to overheating or thermal stresses. Also, liner cooling should be just enough to avoid flame chilling or

reaction quenching. In this design, the segments of the liner overlapped on each other and thin rings were used to step-up the diameter between segments. These rings contained axial holes around the periphery to allow some of the air to be introduced tangentially to the liner surface. The choice of this arrangement has provided effective operation when used in aircraft engines [111].

In order to determine the effectiveness of using this type of film cooling, an empirical correlation was used. Sturgess [109] used the analysis by Stollery et al. [107,108] which was based on a turbulent boundary layer model, to develop a correlation for the effectiveness of film cooling. This correlation was based on adiabatic liner walls as the optimum condition. The formula for the geometry of this combustor is given as:

$$\epsilon = 1.0082 \{1 - 0.12025 S_N^{0.65}\} \quad (3.9)$$

where, S_N = correlation parameter given by

$$S_N = \left\{ \left(\frac{x - x_p}{M_N s} \right) \left(Re_s \frac{u_c}{u_m} \right)^{-0.15} \frac{A_o}{A_E} \right\} \quad (3.10)$$

$$M_N = \rho_c u_c / \rho_m u_m \quad (3.11)$$

ϵ = effectiveness of film cooling

$$= (T_m - T_{s,ad}) / (T_m - T_c)$$

T_m = main stream temperature

$T_{s,ad}$ = adiabatic surface temperature of liner

T_c = temperature of cooling air

ρ_c = density of cooling air

u_c = velocity of cooling air

ρ_m = density of main stream

u_m = velocity of main stream

x = distance measured downstream from injection plane

x_p = potential core length

Re_s = Reynolds' number based on slot height

s = slot height

A_o = slot outlet area

A_E = slot overall effective area

Assuming the combustion efficiency at each section inside the combustor and assuming also constant discharge coefficients for all the holes, the temperature and velocity were approximately estimated at each film cooling ring. Using Eqs. (3.9-3.11) and the combustor dimensions, the effectiveness at each ring was estimated. Table 3.1 shows the values calculated and the effectiveness at the four rings.

The values of effectiveness obtained are acceptable and hence the resulting liner temperature will be within the allowable limit.

Most of the known empirical formulae used for initial sizing of a combustor have been summarized by Odgers [83]. However, upon applying these relationships to size the test combustor of this study, a wide range in combustor dimensions was obtained. The relation by Bragg [13] was an exception and was found to give reasonable values for the combustor dimensions. This relationship was based on pressure drop in the combustor.

This chapter summarized the steps taken in designing a test combustor which can be used to test low-medium heating value gas. The behaviour of this fuel was included in considerations leading to the design. Included in the design was provision for holes at regular intervals along the combustor to give access to the combustion space such that through measurements at these sections, combustor and combustion evaluation is possible. Premixing of some air with the fuel to lower the pollution effect was permitted within the design conditions. Air was supplied separately to primary, secondary and dilution zones of the combustor. This was mainly included due to the narrow flammable range of low-medium heating value gas. Widening of the flammable range might be possible through the change in the primary air swirl of the combustor. Critical performance parameters were evaluated to ensure the suitability of the design for practical applications.

Table 3.1 Effectiveness of Film Cooling at Each of the Rings

Ring #	Re _s Reynolds number	M_N $\frac{\rho_c u_c}{\rho_m u_m}$	S_N Correlation parameter	ϵ % Effective- ness
1	11,497	3.14	6.02	61.85
2	10,470	3.14	2.74	77.46
3	10,470	3.068	2.757	77.38
4	10,470	3.21	2.606	78.22

Fig. 3.1 Relation of minimum combustor pressure loss to reference velocity

Fig. 9.2 Effect of liner open-hole area and liner cross-sectional area on combustor total pressure loss coefficient (Ref. 37)

Fig. 3.5 Variation of discharge coefficient with velocity ratio for rounded orifices

Fig. 3.6 Swirler-injector piece
(details of parts #37+38 of Fig. 3.4)

CHAPTER 4

EXPERIMENTAL SETUP, INSTRUMENTATION AND ACCURACY

This chapter describes the various components of the experimental test rig. Also included is a description of the probes and instruments which were used to make quantitative measurements. This is followed by an analysis of the accuracy of the instrumentation used and the measurements made in this study.

4.1 Test Rig

Fig. 4.1 shows a schematic diagram of the arrangement used as the experimental setup. It consists of a fuel supply, air supply and the test combustor. Fuel was supplied by blending its main constituents from bottles of commercial gases. The main components of the fuel as obtained from fluidized bed coal gasification plants are given in Table 4.1 obtained from Ref. [101]. In order to simulate the operation of an oxygen-blown fluidized bed gasifier, it was decided to consider only CO , H_2 and CO_2 in the fuel mixture and neglect CH_4 due to its small proportion while replacing its heating value by increasing the proportions of CO and H_2 . Lower heating values were obtained by increasing the proportion of CO_2 in the gas mixture which resembles inefficient, or part load operation of the gasifier. Each gas line leading from the respective compressed gas bottle was fitted with a rotameter to measure the flow rate before the gases were all mixed together and admitted to the combustor. This provided a means of control of the fuel flow rate and its composition (which determines the heating value). The heating value was varied, if required, by changing the fuel composition via the proportion of CO_2 in the fuel. Although very low heating values could be achieved, the highest possible heating value was limited by the energy content of both CO and H_2 (heating value of $\text{CO} = 11.853 \text{ MJ/m}^3$ (318 Btu/scf), gross heating value of $\text{H}_2 = 11.988 \text{ MJ/m}^3$ (320 Btu/scf)) [105]. In order to obtain heating values higher than 11 MJ/m^3 , CH_4 (36 MJ/m^3 or 1000 Btu/scf) should be

blended into the fuel but this was not attempted in this study. The carbon monoxide bottles used were of technical grade with a guaranteed purity of 99% while those for hydrogen were extra dry with a minimum purity of 99.95%, 5 ppm maximum moisture content and 3 ppm maximum oxygen content. The carbon dioxide used was provided from commercial grade bottles which had a minimum purity of 99.5%. The three gas lines led into a mixing chamber before leading to the combustor. This chamber was stuffed with steel wool to prevent the flame from travelling to the gas bottles in case of flame flashback.

The combustion air was supplied by a centrifugal air blower of capacity up to $1857 \text{ m}^3/\text{hr}$ at a speed of 3100 rpm. The blower was driven by a variable speed DC motor. In order to measure the air delivered to the combustor and also to vary the flow rate of air to each section, flow metering devices were inserted between the blower and the combustor. The total air supply was measured via a venturi meter mounted at the blower exit using an inclined water manometer. The air was then divided into two parts, one being directly fed to the combustor at either the secondary or dilution zone, while the remaining air flowed through six rotameters before being admitted to the combustor. Normally, this second part feeds the combustion air at the premixed, primary and secondary zones of the combustor. Variation of the total air supply to the combustor was achieved by changing the speed (rpm) of the driving motor. The exit end of the combustor was connected to the laboratory's exhaust system.

4.2 Instrumentation and Equipment

4.2.1 Suction Pyrometer

A radially traversing aspirated pyrometer probe was used to measure the temperature at each section of the combustor. Fig. 4.2 shows a detailed cross section of the probe. The probe was water cooled to maintain its surface at a temperature below 400°C (the safe operating temperature of stainless steel for this application) when inserted inside the combustion space. Platinum versus platinum-13% rhodium (type R) thermocouple was used with a wire diameter of 0.2 mm. A suction pump was used to draw the high temperature gases over the

probe junction at a velocity within the average axial velocity at the cross section where the measurements were made. It should be noted that the cooling of the suction pyrometer occurred in the portion of the probe that had the stainless steel jacket, and that the sensing junction of the probe was in a region which was not cooled, but was surrounded by a ceramic radiation shield. Therefore, the probe should give a true reading of the local gas temperature without being affected by the cooling of the probe or the radiation to the liner walls.

4.2.2 Two-Hole Probe

Fig. 4.3 shows a cross-sectional drawing of the two-hole probe designed for this work. In order to determine the flow direction and identify the recirculation zone, a traverse was made with the 2-hole probe before the velocity vector was determined by the 5-hole probe. This probe was also water cooled to withstand the high temperature inside the combustor.

4.2.3 Five-Hole Probe

A water cooled 5-hole probe (type DA-375) manufactured by United Sensors and Control Corporation was used to obtain the velocity vector at selected measurement points in the various cross sections along the combustor. This type of probe has the advantage that its tip is fixed at the point of measurement while the probe is rotating. Detailed description of the 5-hole probe and its use in combination with the 2-hole probe is given in Appendix A.

4.2.4 Gas Chromatograph

A Fisher Model 1200 gas partitioner chromatograph was used for the purpose of analysis of combustion gases at sampling points in the combustion space. At these points, a gas sample was drawn (using the suction pyrometer probe) into a specially made gas sampling tube. Prior to analyzing the samples, the gas chromatograph was calibrated for the gases of concern to this study, mainly CO , CO_2 , H_2 , N_2 , O_2 . A syringe was used to inject 1 cc of the sample gas to be analyzed in each case. Due to the presence of fuel in the

first three measuring sections, two separate gas chromatographs were used for analysis of samples taken from these sections. However, for the remaining seven sections along the combustor, only one chromatograph was used because only combustion products are expected to exist in these sections. One chromatograph performed the analysis for the hydrogen content in the sample and used nitrogen as a carrier gas. The other chromatograph used helium as a carrier gas and carried out the analysis for the rest of the sample components. Nitrogen was used as a carrier gas for hydrogen analysis due to its high sensitivity [102], as a result of the large difference between the thermal conductivities (on which the detecting and measuring systems of the gas chromatograph are based) of these two gases. On the other hand, the rest of the gases contained in the sample could be detected more accurately if helium were used as a carrier gas. Helium, however, is not suitable as a carrier gas for hydrogen analysis. The analysis could have been performed on one chromatograph by using the commercially available mixture of 8.5% hydrogen in helium as a carrier gas [51]. However, this carrier gas results in a negative peak for hydrogen on the chromatogram which would require that the baseline of the chromatogram be in the middle of the page. This would mean that the amplitude of all components of the sample should be reduced and since this would lower the accuracy of measurements, such a procedure was not adopted.

4.3 Accuracy of Various Instruments and Measurements

In the following paragraphs, a brief discussion is given of the accuracy achievable on the instruments used in this study.

4.3.1 Flow Meters

Flow meters were used to measure the flow rate of fuel and air supplied to the test combustor. The total air flow rate provided by the blower was measured by a venturi meter. Ten air flow lines were mounted at the exit section of the blower following the venturi meter. Six of them passed through flow meters before reaching the combustor, while the other four lines led

directly to the combustor. The total air flow through these four lines was calculated as the difference of the total air flow calculated from the venturi meter and the sum of the readings of the six flow meters.

Model RMC-SSV flow meters made by Dwyer Instruments Inc. were used for the high flow rate regime of this research. These were chosen primarily for their long, clear scale. When carrying out the stability loop tests at very low fuel flow rate, model RMB-SSV was used due to its small flow range. In such flow meters, the flow rate is determined from the position of a float which can move freely along the length of a tube of variable cross-sectional area. Most of these flow meters operate in the turbulent flow range. The RMB series are accurate within $\pm 3\%$, while RMC series are accurate within $\pm 2\%$ [125]. The accuracy range given by Linford [67] for using rotameters for flow measurement is also in the range of 2 to 3%. Careful observation of the flow meters during the run and readjustment for any changes in the flow kept the reading within the accuracy limits mentioned.

4.3.2 Manometers

A variety of manometers were used to measure the pressure differences needed. An inclined manometer with a range of 0-0.25 inches of water was used on the venturi meter. A U-tube manometer was used to measure $P_1 - P_{atm}$ in the five-hole probe. The pressure difference $P_2 - P_3$ was measured by an inclined manometer with a range of 0-0.25 inches of water. The pressure difference $P_1 - P_2$ was measured by an inclined manometer with 0-0.25 or 0-1.0 inches of water range depending on the value measured. The pressure difference $P_4 - P_5$ was measured by a Hook gauge which was equipped with a micrometer for more accurate reading of the pressure difference. The accuracy of the reading of the inclined manometer depends on the precision with which the manometer is levelled as well as the size of division which can be read on the scale. A polished reflective chrome-plated scale mounted alongside the tube allows alignment of the meniscus with its image for accurate consistent readings. The accuracy of the U-tube manometer is within 2 mm of water. Considering an average reading of 130 mm of water during the tests, an accuracy of $\pm 1.5\%$ could be estimated. For the inclined manometers of 0.0 - 0.25 in. of water

range, an accuracy within $\pm 2\%$ is estimated based on a minor division of 0.005 in. of water. The other inclined manometer of range 0-1.0 in. water is accurate within $\pm 4\%$. The Hook gauge manometer is accurate to within 0.001 in. of water, hence its accuracy depends upon the value to be measured.

4.3.3 Suction Pyrometer

The temperature indicated by a thermocouple placed in a gas stream is the resultant of the various heat transfer modes at the thermocouple junction [9]. Errors in the measured temperature arise due to radiation and conduction exchange between the junction, the thermocouple wires and the surroundings. The aspirated suction pyrometer overcomes these errors by compensating for radiation and conduction losses with heat transferred by convection from the gas as it passes over the hot junction, and also by shielding the hot junction to reduce the temperature difference between the junction and the surroundings. In order to reduce conduction losses, a fine wire (0.2 mm diameter) was chosen for the thermocouple [14,53]. It was shown by Chedaille & Braud [14] that the error due to the suction velocity in measuring the static temperature (rather than the total temperature) is limited to 7°C at a suction velocity of 250 m/s. The temperature measured by the suction pyrometer is representative of the temperature at the position of the junction. It was found that the effect of suction at the probe tip is limited to a sphere of 3 mm diameter at a suction velocity of 250 m/s [52,53]. The temperature measured by a suction pyrometer could be accurate to within $\pm 2^{\circ}\text{C}$ [14]. However, depending on the measured temperature, an error of 10°C could be expected in some cases when particles of fuel are deposited on the suction line. Aspiration thermocouples have a very fast response (~ 0.5 min.), provided that a fine wire is used for both legs of the thermocouple. In contrast with calorimetric thermocouples they do not require calibration at every point [72,96]. Also, aspiration thermocouples can be made less bulky than calorimetric thermocouples and hence result in less flow disturbance where inserted for measurements.

4.3.4 Two-Hole Probe

The two-hole probe was used to give an indication of the flow direction, i.e., to know if the velocity vector is pointing upstream or downstream. For this purpose, an inclined manometer was found to be sufficient at positions where the velocity magnitude could be indicated by 0.005 in. of water or more. However, at very low velocities, the direction could be determined either by the Hook's gauge or by measuring the direction at a point above the one under consideration and a point below it. Through interpolation, the flow direction at the desired location could then be defined.

4.3.5 Five-Hole Probe

Three-dimensional probes have been used in connection with fluid mechanics analysis of flows for several decades now. Due to the lack of sensitivity of the conventional Pitot tube to flow direction, attention has turned to the use of multi-hole probes to facilitate the determination of the velocity vector in a flow field. The first exhaustive description of the application of the 5-hole probe was given by Lee & Ash [62]. Application of multi-hole probes have been used since then in a wide variety of investigations [3,7,10,14,28,32,43,61,71].

Probes, by their very nature, produce error in the flow properties due to the effect of their insertion. Area changes due to the presence of the probe result in error in measuring the static pressure. Also, in recirculation flows the disturbance resulting from the probes' presence may travel upstream and cause a behaviour different from that of the undisturbed flow. In combustions flows, some probe materials might act as a catalyst for the chemical reaction. All these disadvantages, however, do not reduce the usefulness of probes as a reasonably accurate source of experimental measurements, since they are easy to use and are affordable.

Experience in applying multi-hole probes for flow measurements has resulted in improved design and therefore, improved accuracy of their measurements. It is generally recommended that the smallest possible probe be

used and that materials which might react with the flow be avoided [34]. The accuracy of the probe is increased by rotating the probe until $P_2 - P_3$ is zero to determine the yaw angle [17]. This simplifies the calculations as the yaw angle need not be calibrated for. A major source of error in the 5-hole probe is caused by the blockage of its tubes by particles, dust or condensation of water vapor (formed from combustion of hydrogen in the fuel) on the water cooled passages. Frequent purging of the holes with compressed air or nitrogen should avoid this problem [16]. The disturbance of flow caused by the insertion of the probe can be monitored to some degree by observing the change of the wall static pressure [16]. In this work, an aspirated thermocouple was used for this purpose and the oscillation in the temperature reading at the wall in a region where the flow temperature was around 1000°C was limited to $\pm 10^{\circ}\text{C}$ upon the insertion of the 5-hole probe. Error analysis of the 5-hole probe was calculated by Gouldin [34] and found to be 2° in yaw angle, 0.2° in pitch angle and 3.5% in the total pressure. Five-hole probes [18,26] showed accuracy within 2-5% for static pressure and dynamic pressure coefficients, about 2° for pitch angle and $3-10^{\circ}$ for yaw angle. The probe used by Rhode et al. [93] was estimated to have an accuracy within 5% for most measurements which could reach 10% in regions of low velocity. Due to its small value, the accuracy of the circumferential velocity is expected to suffer the most [117]. Comparison of axial velocity measurements by the 5-hole probe and those measured by a hot-wire anemometer [117] were found to agree within $\pm 2\%$ in regions of low turbulence level and low velocity gradients. The difference, however, increased to $\pm 10\%$ where turbulence and velocity gradients were higher.

The accuracy of the manometer readings contributes significantly to the accuracy of the measurements of the 5-hole probe. Accuracy of $\pm 5\%$ was estimated on this basis for flow measurements past the recirculation zone and downstream to the combustor exit. Error could be larger in the recirculation zone particularly due to the small magnitude of circumferential velocity and because of the limitation of the probe sensitivity for yaw or pitch angles beyond $40-60^{\circ}$. Another source of error may arise due to the fact that the probe was calibrated in a uniform and homogenous flow unlike the measurement

environment which is non uniform, of variable composition at the combustion space and with a high degree of turbulence.

4.3.6 Gas Analysis

Accuracy of gas analysis using gas chromatography depends mainly on the extent of separation of the peaks for various gases as well as upon the ability to obtain sharp, clear peaks with the maximum possible amplitude. The carrier gas flow rate and column temperature determine to a great extent the separation of peaks of various gases [49]. Several trials were carried out with a standard gas sample to determine the optimum carrier gas flow rate and column temperature. The chromatograph for hydrogen analysis was calibrated using only hydrogen as a sample gas.

The other factors which contribute to the accuracy of the gas analysis are the effectiveness with which the sampled gas is quenched from further reaction [12], calibration accuracy for the various gases to be analyzed as well as the reactivity of the gas sampling containers. Quenching of the sample is necessary mainly while analyzing for small fractions such as pollutant gases. Calibration was done several times before every gas analysis and the average of these analyses was considered. Also, analysis of the standard sample was carried out after the gas analysis of the samples collected to confirm the original calibration results. In order to establish that the sample containers were reaction free, standard samples were analyzed immediately after the container was filled and two days later. It was found that within the level of precision achievable, the same analysis was obtained in both cases which verified the reaction-free nature of the sampling containers.

Appendix B describes the procedure followed in taking the various measurements and the technique that was followed to perform some of the experiments.

Table 4.1 Species Concentration and Heating Value for
Medium and Low Btu Coal Derived Gases Ref. [101]

Gasification Technique	Species in Mole Percent					HV Btu/scf
	CO	H ₂	CO ₂	CH ₄	N ₂	
Lurgi O ₂	18.5	40.2	29.4	9.4	0.6	285
Winkler O ₂	32.9	41.2	20.0	3.0	1.0	269
Kopper- Totzek } O ₂	52.1	34.5	9.2	0.5	1.0	289
Wellman- Galusha } Air	26.3	14.3	2.4	2.6	46.9	160
Winkler Air	21.1	13.0	6.9	0.6	56.5	116
	CH ₄	C ₂ H ₆	C ₃ H ₈	N ₂	CO ₂	
Natural Gas	72- 94	2.1- 7.8	0.4- 3.6	0.- 18.	0.1- 2.8	934-1051

Fig. 4.1 Test rig layout.

Scale 1:1

All Stainless Steel Tubes are Type 304

Fig. 4.2 Details of suction pyrometer

Fig. 4.3 Two-hole probe

CHAPTER FIVE

EXPERIMENTAL RESULTS AND DISCUSSION

Experiments were carried out on the test combustor designed to burn low-medium heating value fuels (the test rig was described in chapter 4) in order to determine the combustion characteristics of such fuels and to study the influence of various operating and design parameters. These experiments consisted of varying parameters such as the air distribution among combustor zones, air flow rate, fuel flow rate, the fuel mixture ratio (and thereby, the heating value) and the vane angle of the swirler at the primary inlet, for selected conditions. This chapter outlines the effects of these parameters on the measured properties in the combustion space. Table 5.1a gives a summary of the operating conditions for the ten runs that are reported in this chapter for which detailed temperature and measurements were taken throughout the combustion space. At these conditions, run #1 is considered to be the base run for comparison purposes. Table 5.1b gives a summary of the conditions during ten more runs where measurements of temperature only were taken at the exit section. These runs were utilized in developing the global combustor efficiency formula described in chapter 6. Results of the present work are discussed and compared with results obtained by other investigators for both low-medium heating value gas and other gaseous fuels. Fig. 5.1 shows a schematic drawing of the combustor which indicates the various zones and measuring sections

The various results that were obtained during the experimental investigation are given in this chapter in graphic form. In order to simplify the comparison on the graphs, the same scale was used for the same plotted value (i.e. for temperature and for velocity components). This, however, resulted in some velocity profiles (especially the circumferential) to be absent from some figures and also, it was not possible to show the change of temperature or velocity at some sections. For this reason, the numerical values for the temperature and the three velocity components are given in

Annex 1, for easy reference to the numerical values.

In order to evaluate the performance of the combustor, an efficiency term was calculated and it was termed combustor efficiency to distinguish it from the conventionally used combustion efficiency. The combustor efficiency will give an evaluation of the performance of the combustor in delivering the energy in the fuel to its gases at the exit section taking into consideration the surface heat losses from the combustor. This is unlike the combustion efficiency which only indicates the completion of the combustion from the chemical point of view. The combustor efficiency is defined in Appendix C and an example of its calculation is also given. The combustor efficiency for the runs of this study are given in Table 5.1

5.1 Effect of Air Distribution Among Combustor Zones

In the case of combustors utilizing conventional fuels, the air addition is distributed almost equally among the various zones of the combustor. However, when burning fuels in the low-medium heating value range, some change in this distribution might be required for better performance. In the runs that are compared here, the air proportion introduced at the various zones of the combustor (mainly secondary and dilution) was varied while all other parameters were held constant.

Fig. 5.2 shows the results of radial measurements of temperatures at several cross sections along the combustor obtained upon changing the air distribution between the secondary and dilution zones of the combustor. These temperature profiles show clearly that changing the air distribution among combustor zones affects the temperature distribution inside the combustion space. The effect can be seen either in terms of the shape of temperature profiles or the magnitude of temperature at the various positions in the combustor. The shape of the temperature profile remains almost the same for the first six measuring sections of the combustor while a change could be noted for the last four sections, especially the section close to the combustor exit. The effect of air distribution among the combustor zones on the shape of the temperature profile can be illustrated better through a comparison of the "shape factor profiles" (or traverse numbers) defined below.

The common practice for combustor design is to calculate the traverse number as in Eq. (5.1). In this study it was decided to introduce a parameter which would help in quantitative analysis of the temperature profile at the exit section, namely a "shape factor" as defined in Eq. (5.2).

$$\text{traverse number} = (t_{\text{max}} - t_m) / (t_m - t_a) \quad (5.1)$$

$$\text{shape factor} = (t - t_m) / (t_m - t_a) \quad (5.2)$$

where, t_{max} = peak temperature at the combustor exit section

t_m = mean temperature at the combustor exit section

t_a = air inlet temperature to the combustor

t = temperature at any position at the combustor exit section

The shape factor gives a better evaluation of the uniformity of the temperature profile at the exit section. It also indicates the extent to which the first row of nozzle blades (in the gas turbine) is utilized (see Appendix D for more details regarding the shape factor). In order to avoid blade failure, the maximum value of the shape factor (or the traverse number) should occur at a location as far as possible from the highly stressed blade root. In most gas turbine designs, the temperature peak or traverse number should occur at the middle of the blade length.

Fig. 5.3 shows a plot of the shape factor profiles at the combustor exit resulting from variations in the distribution of air between the combustor secondary and dilution zones. Although the temperature profile of run #1 is not as uniform as that of run #2 or run #3, it is clear from Fig. 5.3 that the temperature distribution of run #1 is better than those of runs #2 & 3 from the viewpoint of the shape factor. The value of the traverse number (calculated at peak temperature) is 0.055 for run #1, while it is 0.126 for run #2 and 0.094 for run #3. In terms of blade utilization, run #1 shows a better approach since in this case the maximum negative value of the shape factor is less than that with run #2 or run #3. The shape factor approach provides a criterion for evaluating the temperature profiles at the exit section of a combustor. Since the blade strength is based on the maximum temperature (or the traverse number), a highly negative value for the shape

factor will imply that the blade is overdesigned for most of its length except near the location at which the traverse number value occurs. The author suggests that the maximum negative value of the shape factor should be within the same range as the traverse number and therefore it should not be higher than 0.2 [40].

The quantitative effect of the air distribution among the combustor zones on the temperature values can be explained through the use of a plot of the temperature near the combustor walls (upper and lower) and centerline. Fig. 5.4 represents such a plot. It can be seen that the effect is clear only in the first six measuring sections of the combustor. Varying the air distribution between the secondary and dilution zones affects the mixing rates as well as the reaction rates inside the combustor. Altering the reaction rate affects the extent of the reaction zone which results in change of temperature profiles in these zones. The mixing rate (between combustion products and air) has a direct effect on the uniformity of the temperature at the various sections of the combustor and hence, the variations shown in Figs. 5.2 and 5.4. Such a trend is enhanced in the case of low-medium heating value gas resulting in shorter flames compared to those with natural gas; hence, the demand for larger amounts of air at the secondary zone. The air distribution of run #3 resulted in a smooth temperature variation along the combustor at the three chosen positions. The maximum temperature of the gas near the upper liner wall approached 1300°C in run #1 and 1300°C in run #2 while in run #3, it was close to 1400°C . The gas temperature near the lower liner wall during run #1 was close to 1400°C , which was higher than the corresponding temperature for the other two runs. The gas temperature near the liner walls has a direct impact on the liner life and the film cooling technique required. From this point of view, it is clear that admitting larger amounts of air in the secondary zone of the combustor will prolong the life of its liner.

Figs. 5.5 through 5.7 summarize the results obtained for the radial profiles of the velocity components, namely, axial, radial and circumferential, upon changing the air proportion between the secondary and dilution zones. Fig. 5.8 depicts values of each of the three components along the combustor centerline as affected by the same parameter.

The extent of the recirculation zone indicated by the region of negative axial velocity in Fig. 5.5, does not change appreciably during the three test conditions. However, the shape of the recirculation eddy of run #3 is better defined than in runs #1 & 2. The recirculation eddies of runs #1 and 2 experience a double reversed region off the combustor centerline and close to the liner boundaries at the third measuring section, while run #3 shows only a central reversed region at the same section. This could be attributed to two main reasons. The first is the enlargement of the liner at the combustor head. This enlargement causes the annular air flow to separate from the axial flow coming from the combustor head [8]. However, due to the admission of air into the secondary zone, an annular flow region near the liner wall is formed which flows downstream resulting in the positive value of the axial velocity component near the boundaries. The second reason is the entrainment of air from the annular flow into the central primary and premixed flow at the centerline of the combustor. If the annular air is enough as required for entrainment by the core flow, the flow will proceed downstream as if flowing in a duct. However, if the entrainment is more than the air available, a reversed flow will be created near the boundaries [8]. Due to the large amount of air available at the secondary zone in the case of run #3, the reversed flow near the boundaries did not take place and only a central zone was created via the swirl imparted to the primary combustion air.

The peak in the axial velocity component, Fig. 5.5, near the liner wall at the dilution zone (the last three measuring sections before the combustor exit) experienced in run #1 and run #2 was eliminated by the air distribution of run #3. These peaks are believed to be caused as a result of the flow of air for film cooling of the liner and the presence of the 5-hole probe (for velocity measurements) in this region. However, when the flow in the dilution zone was reduced as in the case of run #3, the amount of axial air flow for film cooling became too small to disturb the 5-hole probe readings.

As for the axial velocity profile at the exit section, the air distribution of run #1 gave an almost uniform axial velocity across the section, unlike runs #2 and #3. The uniformity of the exit section velocity profile is dependent on the time given to the flow to stabilize itself close

heating value can be attributed to the lower flame temperature. This results in a gradual rise of temperature in the reaction zone and hence, in reduced demand for efficient mixing between the combustion products and the secondary or dilution air. Examining the profiles of the shape factor, Fig. 5.24,* which were calculated from the temperature profiles at the exit section reveals that the temperature distribution of run #1 is still the best, thus far. While run #7 had resulted in a slightly higher traverse number (0.062) than did run #1 (0.055), run #6 resulted in a much larger traverse number of 0.147. The maximum negative value of the shape factor experienced with run #6 was the highest of the three runs (0.225).

The temperature values near the upper wall, centerline and lower wall of the combustor liner along the combustor length are given in Fig. 5.25. The figure indicates that the maximum wall temperature is $\sim 400^{\circ}\text{C}$ lower in the case of run #6 compared to run #1, when the heating value is lowered. Fig. 5.25 also confirms the observation made from Fig. 5.23 that the temperature is lower everywhere upon reduction of the heating value; about 300°C difference between run #1 and run #6, while it is $\sim 100\text{--}150^{\circ}\text{C}$ between run #6 and run #7 through the primary and secondary zones.

The effect of the heating value on the profiles of the axial velocity plotted at various sections along the combustor was quite negligible for the three heating values tested, as can be seen in Fig. 5.26. The reason is mainly due to the constant flow rate of air and the small contribution of the fuel flow rate. The uniformity of temperature upon reducing the fuel heating value can be seen to also result in slight improvement in the uniformity of the axial velocity profiles, especially at the last four measuring sections.

The centerline distribution of the axial velocity along the combustor length, Fig. 5.27, shows the effect of the temperature level inside the combustor. The axial velocity is higher with the higher heating value due to the higher temperatures as implied by the continuity principle and the equation of state. This is especially clear at the exit section.

The radial velocity profile given in Fig. 5.28 indicates also lower values as the heating value is reduced. This is mainly due to the decrease in temperature upon lowering the heating value. Also, the centerline values of

magnitude and direction of the radial velocity at the various sections of the combustor. In run #3, a larger proportion of air was admitted at the secondary zone rather than at the dilution zone. This resulted in the radial velocity being counteracted by the buoyancy effect as the flow proceeded downstream. This appears as a positive value for the radial velocity in the upper half of the combustor. The radial component was present right up to the exit section. This undesirable situation can be avoided through redesign of the flow reduction area before the combustor exit section by making the area change more gradual over a larger length of the liner and the use of streamlined contour rather than the straight walls of the present design.

The plot of the radial velocity component at the combustor centerline along its length, Fig. 5.8, also shows a significant change in magnitude and direction, but the values for run #3 were close to zero along most of the combustor length.

The circumferential velocity component plotted in Fig. 5.7 (refer to Annex 1 for numerical values of the circumferential velocity especially where the profile does not appear on the figure) shows that larger values existed in the primary zone for runs #1 and #2, while the large values extend one section further downstream for run #3. The circumferential velocity component may arise from the component given by the swirler to the primary air, as well as due to the imperfection in the radial holes that admit the fuel-air mixture or the secondary and dilution air. It may also result if the axial holes for film cooling are not perfectly axial. For optimum performance, the circumferential velocity component should have a value as close to zero as possible when the flow leaves the combustor exit section. The plots in Figs. 5.7 and 5.8 both indicate a decay of the circumferential velocity as the flow proceeds downstream. It should be mentioned here that larger error is expected in measuring the circumferential velocity either due to its small value or very high pitch angle, particularly at the recirculation zone. However, the circumferential velocity could be considered acceptable in these three runs, provided that further improvement can be achieved through redesign of the flow reduction area near the combustor exit as mentioned above.

5.2 Effect of Air Flow Rate

The air that could be supplied by the blower for these tests was not sufficient to allow tests to be carried out at a flow rate higher than that used for the base run. Therefore, the only variation possible was reducing the air flow rate through the combustor. The air flow rate was reduced by a factor of 1.5 while keeping the other parameters constant. In order to keep the test conditions consistent in terms of air/fuel ratio, the fuel flow rate was also reduced by the same factor.

Fig. 5.9 shows the temperature profiles at the various cross sections along the combustor upon reducing the total air flow rate. The shape of the temperature profile is significantly different only for the first and exit sections. However, the magnitude of the temperature is less everywhere for lower mass flow rate (run #5) compared with run #1. Quantitative analysis of the difference of the temperature profiles at the exit section can be illustrated through reference to the shape factor plot of Fig. 5.10. The shape factor is less everywhere for the case with the higher flow rate (run #1). The traverse number for this run is 0.055 compared to 0.147 for run #5 having a lower air flow rate. Generally, the profile of run #1 indicates better blade utilization than that of run #5, since a maximum negative value of 0.235 for the shape factor is found for run #5 compared to 0.125 for run #1.

The lower mass flow rate results in reduced ability of penetration of the air admitted in the secondary and dilution zones; hence, reduced mixing occurs which leads also to incomplete combustion. This is clear from the lower level of temperature readings throughout the combustion space upon reduction of the flow rate, Fig. 5.9. Another fact to be stated here is that the chemical energy content of the fuel is reduced by the same factor of 1.5 upon reducing the mass flow rate. While the energy input and the flow rates were reduced by the same factor, in principle the heat losses from the combustor surface stay the same due to their dependence on the surface temperature as well as the surface area of the combustor. This is manifested in terms of loss in the combustor efficiency (Table 5.1). The plot of temperature near the upper wall, centerline and lower wall, Fig. 5.11, shows further that the temperature

is less everywhere with the reduction of flow rates.

Profiles of the three velocity components along the combustor are plotted in Figs. 5.12 through 5.14. The centerline value of each of these components along the combustor are plotted in Fig. 5.15. The effect of reducing the flow rate on the axial velocity as shown in Fig. 5.12, is a more uniform velocity throughout the combustor with the exception of the exit section. This may be interpreted based on two facts; the lower but more uniform temperature that prevails with the lower air flow rate implies more uniform flow. Also, the lower mass flow rate of run #5 through the secondary and dilution holes causes lower axial velocity, which in turn reduces the impact on the core flow. The size of the recirculation zone as indicated by the region of negative axial velocity in Fig. 5.12, however, did not change appreciably from run #1 to run #5. The illustration outlined in the previous section regarding the reversed flow zone near the liner walls is more clear with the axial velocity profiles of run #5 due to the lack of sufficient air for entrainment. The centerline value of the axial velocity along the combustor, Fig. 5.15, did not show a large qualitative variation; however, the magnitude of the axial velocity is less for run #5 than for run #1 due to the lower mass flow rate and lower temperature levels inside the combustor (assuming that pressure changes are negligible, considering that the inlet air pressure to the combustor was 160 mm water, from experimental measurements by the 5-hole probe, and the minimum average pressure at the exit section was above 80 mm of water, which means a pressure loss of less than 1%).

Fig. 5.13, which shows the radial velocity profiles at various sections along the combustor length for runs #1 and #5, illustrates again the effect of the reduction of mass flow rate upon this velocity component. The buoyancy effect is enhanced in the flow of run #5 due to the reduced radial admission of flow. The radial component at the combustor centerline, however, decays much faster and stays at a small value throughout the length of the combustor for run #5 than for run #1 except for the last four measuring sections where the radial velocity for run #1 was less than half that of run #5.

The circumferential velocity component profiles along the combustor, Fig. 5.14, indicate lower values of this component throughout the combustor for the lower air flow rate. Again, the reduction in the flow rate contributes in two ways to the reduction of this component. Firstly, the primary swirling portion of flow is reduced. Secondly, the amount of air that is admitted through the secondary and dilution holes is reduced and if there is any imperfection in these holes being radial, the possibility of a circumferential component being created this way is reduced. The circumferential component along the combustor centerline, Fig. 5.15, shows also faster decay with run #5 than with run #1, and also has a smaller magnitude for run #5 (about half that of run #1).

5.3 Effect of Fuel Flow Rate (Air/Fuel Ratio)

Due to the considerable influence of fuel on the temperature distribution throughout the combustor, it was decided during this test to increase the fuel flow rate by a factor of only 1.2 from its value for the base run. This resulted in temperatures comparable to those in the base run, while at the same time avoiding any excessive heating of the combustor liner.

Fig. 5.16 shows a comparison of temperature profiles at the various cross sections along the combustor corresponding to variation in fuel flow rate (lowering air/fuel ratio). The figure indicates that the temperature is higher only at the first and last four measuring sections of the combustor upon increasing the fuel flow rate. The higher temperature existing in the primary zone of the combustor resulted in an increase of heat loss from the combustor surface. This increase in heat loss caused the temperature to be lower than would be expected from the reduced air/fuel ratio. This also contributed to a decrease in the combustor efficiency, as indicated in Table 5.1 where the combustor efficiency was 74.4% for run #4 compared to 91.1% with the higher air/fuel ratio of run #1. A comparison of the shape factor profiles at the exit section, Fig. 5.17, shows again that the temperature profile of run #1 (lower fuel flow rate or higher air/fuel ratio) is superior to that of run #4 (lower air/fuel ratio). A traverse number of 0.055 was found for run #1; 0.09 was the corresponding value for run #4. Also better

blade utilization is expected if the temperature profile of run #1 (maximum negative shape factor of 0.125) is used rather than the temperature profile of run #4 (maximum negative shape factor of 0.21).

The longitudinal plots of temperature near the upper wall, centerline and near the lower wall of the liner are given in Fig. 5.18. The figure shows that with the exception of the first and last two measuring sections, the upper and lower walls of the combustor experienced lower temperatures with the lower air/fuel ratio. The centerline value, however, is about 150°C higher with the lower air/fuel ratio. Altering the flow rates at the combustor head changes the mixing rate between the primary air and air-fuel mixture introduced by the swirler-injector piece. This, in turn, changes the reaction rates and lowers the combustor and combustion efficiencies in the case of run #4.

Fig. 5.19 shows the profiles of the axial velocity component at various measuring sections along the combustor resulting from change in fuel flow rate. The width of the reversed axial velocity region in the second measuring section is larger for run #4 than for run #1 resulting in a bigger recirculation zone with the decrease in air/fuel ratio. This is as expected; increasing the amount of fuel in the mixture enhances the ability of the central flow to entrain a larger amount of air. However, due to the fixed air flow rate, this results only in a larger recirculation zone (the reversed flow near the boundaries is not affected). The profiles of run #4 are more uniform compared to those of run #1 and they appear to start to become fully developed earlier on in the dilution zone. The same observation can be made in Fig. 5.20 for the centerline value of the axial velocity along the combustor; the lower air/fuel ratio (run #4) results in more uniformity for the centerline value of the axial velocity than for the run with the higher air/fuel ratio (run #1).

Fig. 5.21 shows the profiles of the radial velocity component resulting from reduced air/fuel ratio (increased fuel flow rate). The figure shows the effect of changing the air/fuel ratio on the radial velocity component as a result of the change of temperature which, in turn, has a large effect on the buoyancy forces. The same magnitude of radial velocity resulted in both cases

without much noticeable difference. The longitudinal distribution of the centerline radial velocity, Fig. 5.20 shows that this value remains close to zero along the entire combustor length with the lower air/fuel ratio of run #4 than with the higher air/fuel ratio of run #1.

Fig. 5.22 gives a plot of the results obtained for the circumferential velocity component at the two fuel flow rates. It can be noted from the figure that lower values are experienced throughout the combustion space with the higher fuel flow rate. The circumferential velocity was also more uniform across the last four measuring sections with the higher fuel flow rate (run #4). Fig. 5.20 shows also that the centerline value of the circumferential velocity decays faster along the combustor length with the higher fuel flow rate and does not change direction as frequently as with the lower fuel flow rate (run #1). Both results concerning the radial and circumferential velocity components may be attributed to the more uniform temperature distribution. They can also be related to the fact that a larger amount of flow is coming from the swirler-injector head at a uniform velocity counteracting some of the downstream disturbances.

5.4 Effect of Fuel Heating Value

The change in heating value of a fuel is an indication of change in the fuel composition also. Varying the fuel composition would affect its flammability limits. This is one aspect of the effect of changing the heating value on the combustion characteristics of the fuel. Another aspect arises from the fact that reducing the heating value would require the combustor to be supplied with a larger amount of fuel in order to achieve the intended temperature rise. Increasing the fuel flow rate at the combustor head alters the mixing rate [8] inside the combustor, which in turn, affects the reaction rate and the performance of the combustor. The heating value of the fuel also affects the flame length; shorter flames were found to occur upon reduction of the heating value [101]. However, below a certain heating value, the flame starts to increase in length again due to the incomplete mixing of fuel and oxidant.

The air/fuel ratio should remain unchanged while investigating the effect

of varying the fuel heating value. However, reducing the heating value would result in a drop in the temperature levels inside the combustion space. This changes the kinetics and dynamics of the flow which makes comparisons invalid. For this reason, the fuel/air ratio was increased for lower heating values. This, of course, would be the practice in actual application.

During the course of testing, as the heating value was reduced to 6.14 MJ/m^3 ($\sim 165 \text{ Btu/scf}$), the combustion became very unstable. The temperature reading oscillated very rapidly and flame blowout occurred very often. That made it impossible to carry on with the run under these conditions. The air/fuel ratio and fuel flow rate were altered to try to achieve a flammable mixture and stable conditions without significantly changing the temperature levels, but these attempts were not successful. A value of 6.62 MJ/m^3 (178 Btu/scf) was found to be the minimum permissible to maintain stable combustion and to conduct the tests needed to collect the required data. This heating value also required that the premixed air to be reduced to 4%. Fuels of heating values lower than 6.14 MJ/m^3 were tested, but ignition was still very difficult and combustion could not be maintained even for those mixtures which achieved ignition. It is worth noting here that a "stable combustion" during these tests meant having combustion with the ignition source turned off and temperature oscillations at any measuring point in the combustion space being less than $\pm 5^\circ\text{C}$ at steady state.

The minimum heating value that could be utilized in the combustor as was found in this study is much higher than what other investigators had suggested. The reason mainly is due to premixing the fuel with part of the combustion air. The effective heating value, if air is considered as a diluent to the fuel, is 3.7 MJ/m^3 which falls within the range reported by others.

Fig. 5.23 shows the temperature profiles along the combustor for several selected sections at various heating values. Generally, the temperature level drops with the drop in heating value due to reduction in energy input. However, the temperature profiles throughout the combustor remain identical in shape, except at the combustor exit section. The slight improvement in temperature distribution noted at the various sections upon reduction of the

heating value can be attributed to the lower flame temperature. This results in a gradual rise of temperature in the reaction zone and hence, in reduced demand for efficient mixing between the combustion products and the secondary or dilution air. Examining the profiles of the shape factor, Fig. 5.24, which were calculated from the temperature profiles at the exit section reveals that the temperature distribution of run #1 is still the best, thus far. While run #7 had resulted in a slightly higher traverse number (0.062) than did run #1 (0.055), run #6 resulted in a much larger traverse number of 0.147. The maximum negative value of the shape factor experienced with run #6 was the highest of the three runs (0.225).

The temperature values near the upper wall, centerline and lower wall of the combustor liner along the combustor length are given in Fig. 5.25. The figure indicates that the maximum wall temperature is $\sim 400^{\circ}\text{C}$ lower in the case of run #6 compared to run #1, when the heating value is lowered. Fig. 5.25 also confirms the observation made from Fig. 5.23 that the temperature is lower everywhere upon reduction of the heating value; about 300°C difference between run #1 and run #6, while it is $\sim 100\text{--}150^{\circ}\text{C}$ between run #6 and run #7 through the primary and secondary zones.

The effect of the heating value on the profiles of the axial velocity plotted at various sections along the combustor was quite negligible for the three heating values tested, as can be seen in Fig. 5.26. The reason is mainly due to the constant flow rate of air and the small contribution of the fuel flow rate. The uniformity of temperature upon reducing the fuel heating value can be seen to also result in slight improvement in the uniformity of the axial velocity profiles, especially at the last four measuring sections.

The centerline distribution of the axial velocity along the combustor length, Fig. 5.27, shows the effect of the temperature level inside the combustor. The axial velocity is higher with the higher heating value due to the higher temperatures as implied by the continuity principle and the equation of state. This is especially clear at the exit section.

The radial velocity profile given in Fig. 5.28 indicates also lower values as the heating value is reduced. This is mainly due to the decrease in temperature upon lowering the heating value. Also, the centerline values of

the radial velocity along the combustor, Fig. 5.27, are closer to zero in the case of runs with lower heating values.

Fig. 5.29 shows the profiles of the circumferential velocity at selected sections along the combustor upon reducing the fuel heating value. While a value of ~ 3 m/s prevailed in the last five measuring sections of the combustor for run #1, the corresponding value for runs #6 and #7 are about 0-1.5 m/s. Even in the primary zone, the circumferential velocity levels are lower with reduced heating values. This again could be related to the temperature. Also, the contribution of the radial flow through the secondary and dilution zones is less with the lowered heating value due to less heat being radiated to the secondary and dilution air via the liner walls as a result of the drop in temperature levels. The effect is more clear along the centerline of the combustor as the circumferential velocity ranges from 0-1.0 m/s, Fig. 5.27.

The improvement of the velocity components upon reduction of the heating value could be related (as described above) to the decrease of the heat transfer to the air in the space between the liner and the combustor outer case, via the liner, due to the lower flame temperature. This causes the flow through the secondary and dilution holes or through the film cooling holes to have lower velocity, and hence less distortion in the velocity profiles or in their values. Also, the lower temperature results in better accuracy of the measurements of the 5-hole probe due to the decrease of the temperature gradient.

5.5 Effect of Primary Air Swirl

These tests were carried out by maintaining the total air flow rate constant and replacing the swirler-injector piece, Fig. 3.6, at each time to vary the amount of swirl imparted to the primary combustion air. Four swirler-injector pieces were made for this purpose with vane angles of 0° , 30° , 45° and 60° respectively (see Fig. 3.6 for details of swirler). The 60° vane angle swirler was mounted during the base run and the rest of the swirlers were then compared to its performance. During these tests flame flashback was encountered several times with 0° and 30° vane angle swirlers. However, the flame stabilized itself in the tube carrying the fuel-air mixture

to the injector piece which resulted in overheating of the surrounding parts of the combustor. The approaches that were followed to solve this problem were in principle attempts to render the fuel-air mixture, in the tube, non-flammable. The first approach was to reduce the fuel flow rate by a factor of 1.5. However, the problem persisted and it was decided to reduce the premixed air proportion to 4% (from 10%) and redistribute the extra air to the secondary and dilution zones, resulting in 43% in each (the other 10% goes to the primary zone at the swirler).

Generally, the swirlers with vane angle of 30° , 45° , and 60° performed reasonably well in terms of combustion stability. But severe instability and flame blowout were encountered with the 0° vane angle swirler. Some oscillations occurred with the 30° vane angle swirler which were noticed during the temperature measurement; flame flashback occurred a few times before the air flow distribution was readjusted to reduce the premixed air. The oscillations with the 30° vane angle swirler were, however, not severe enough to cause flame blowout or to require the ignition source to be kept on during the entire run. This is unlike the instability encountered with the 0° vane angle swirler which needed the ignition source to remain on for the entire run duration. This may be explained from the fact that a larger reversed flow zone exists at the first measuring section (primary zone) for the 30° vane angle swirler than for the 0° degree vane angle swirler which assisted in maintaining combustion stability.

Fig. 5.30 shows a plot of the temperature profiles at selected sections along the combustor for various values of primary air swirl. The effect of altering the swirl can be noticed mainly in the temperature profiles at the first two measuring sections as well as at the exit section. In particular, the effect was very noticeable for the first section of the 0° vane angle swirler compared to the other three swirlers. The centrifugal force created by swirl at the primary zone contributes greatly to the rate of mixing as well as to the entrainment characteristics of the core flow. The shape of the temperature profile of the run with zero swirl (run #8), at the first measuring section, is quite different from the conventional shape occurring in the presence of swirl. In the later case, the maximum temperature is shifted

to both sides off the combustor axis. Fig. 5.31 shows the shape factor profiles at the exit section for the four runs. Again, the temperature profile of run #1 with the highest swirl (corresponding to 60° vane angle swirler), resulted in the smallest value of the traverse number (0.055), compared with 0.084 for run #10 (45° vane angle swirler), 0.084 for run #8 (0° vane angle swirle) and 0.107 for run #9 (30° vane angle swirler). In terms of blade utilization, runs #1 and #8 are the same (maximum negative shape factor of 0.12), while run #9 approaches a maximum negative shape factor of 0.18 and 0.163 for run #10.

Fig. 5.32 shows plots of the temperature distribution along the combustor length near the upper wall, centerline and near the lower wall of the liner. The temperatures with the 0° , 30° , and 45° vane angle swirlers are about 200°C less than those with the 60° vane angle swirler. This may be attributed to the reduction in fuel flow rate of those runs (Nos.8 through 10).

The effect of primary air swirl on the axial velocity is illustrated in Fig. 5.33 which represents the axial velocity profiles at selected sections along the combustor, for the various swirlers of this study. The shape of the profiles remains unchanged throughout most of the sections; however, the width of the reversed zone in the first measuring section shows a decrease with the decrease of swirl. While the reversed portion of the axial velocity occupied 50% of the diameter with the 60° vane angle swirler (run #1), it occupied only 32% with the 0° vane angle swirl (run #8), 40% with the 30° vane angle swirler (run #9) and 44% with the 45° vane angle swirler (run #10). This is a direct result of widening the recirculation zone as the swirler angle increases. With the 0° vane angle swirler the axial velocity profile at the exit section showed less uniformity. This may be attributed to instability that is carried downstream to the combustor exit. However, this could be avoided in practice by improving the design of the region of flow area reduction at the combustor exit. At lower swirl, the axial velocity levels are somewhat lower, mainly because of the reduced temperature due to the drop in fuel flow rate. The centerline value of the axial velocity along the combustor, Fig. 5.34, changes only by less than 2 m/s for the runs with the swirlers of vane angle 0° , 30° and 45° as the flow moves downstream from the recirculation zone to the

reduced area before the combustor exit. In contrast, the change is quite large with the 60° vane angle swirler. Downstream from the primary zone, the axial velocity increases by about 15 m/s and gradually decreases within the secondary zone, with a gradual increase of about 10 m/s in the dilution zone before the exit section. There is no apparent reason for this behaviour, but, the reason given before in section (5.4) involving the combined effect of heat transfer and temperature could be invoked again.

It is clear from studying the axial velocity profiles for the various amounts of swirl imparted to the primary combustion air, that swirl is not the only cause for the recirculated flow. The central portion of the injector-swirler piece, Fig. 3.36, which is a solid plate of 38 mm diameter (1.5 in.) acts as a bluff body, thereby creating a reversed flow zone. The effect of swirl is then superimposed on the reversed flow created from the bluff body effect of the swirler-injector piece. Thus, although, the swirler vane angles were made zero for one swirler, a recirculated zone still existed.

The radial profiles of the radial velocity component along the combustor are given in Fig. 5.35 for the various swirlers. With the 0° vane angle swirler the value of the radial velocity is the smallest at the first section due to the axial entry of the primary air. However, as the flow proceeds downstream, the contribution of the radially admitted flow to the radial velocity becomes noticeable with all the swirlers. The profiles from the 0° , 30° and 45° vane angle swirlers do not vary greatly but the profile for the 60° vane angle swirler has a different shape. This can be related, as mentioned before, to the heat transfer and temperature distribution throughout the combustor. The same is true of the centerline distribution of the radial velocity along the combustor, Fig. 5.34. Slight variation (less than 2 m/s) is noticed along the combustor for the 0° , 30° and 45° vane angle swirlers, while a value of -6 m/s is reached with the 60° vane angle swirler which then rises to +3 m/s before the flow area reduction near the exit section.

The circumferential velocity component, Fig. 5.36, shows the lowest value at the first section with the 0° vane angle swirler. The circumferential velocity level of the 60° vane angle swirler is quite high throughout the combustion space in comparison with those of the 0° , 30° or 45° vane angle

swirlers. The small value of the circumferential velocity with the 0° vane angle swirler is directly related to the axial admission of the primary combustion air. The only source of such a component in the case of the 0° vane angle swirler is the possible imperfection in the radial injection holes, the axial vanes or in the secondary and dilution zone holes located downstream. The circumferential velocity decays to a satisfactorily low level as the flow proceeds downstream near the combustor exit where there are no holes for air admission. The centerline value of the circumferential velocity, Fig. 5.34, becomes less than 1.5 m/s, downstream from the recirculation zone for all four swirlers. A large value of up to 10 m/s was noticed at the primary zone for the 60° vane angle swirler. The reason that this component is quite high in this case may be due to the loss of sensitivity of the 5-hole probe in the highly recirculated zone obtained in this region with the 60° vane angle swirler.

5.6 Stability Loops

Four runs were carried out for the purpose of establishing the stability loops at various degrees of swirl for medium heating value gas in the test combustor. During these tests the heating value was maintained constant (by means of keeping the same proportions of the gaseous constituents of the fuel). Also, the proportions of the air distribution among the various zones of the combustor for all these runs were the same as those of the base run. The only parameter changed was the amount of swirl imparted to the primary combustion air. Fig. 5.37 shows the results of the stability loops obtained for various swirlers. The figure shows that at the various swirl numbers of the primary combustion air, the combustor has a satisfactory range of ignitable mixture ratios. The range, however, is somewhat narrow for the 60° vane angle swirler. The figure also shows clearly that the range of ratios of ignitable mixtures is very highly affected by the primary air swirl, with the range decreasing as the swirler vane angle to the combustor axis increases. Increase in swirl, on the other hand, resulted in a slight increase in the value of the rich mixture ratio (for the same air flow rate).

The main reason behind the variation in the range of ratios of ignitable

mixtures as the primary combustion air swirl varies is the effect of swirl on stability. Swirl is largely responsible for the creation of the recirculation eddy in the primary zone of the combustor (which in some cases may extend to some distance into the secondary zone). In this recirculation eddy, the flow velocity is reduced in order to match the flow velocity with the flame speed. Also, some of the heat generated from combustion is recirculated upstream to heat the fresh mixture, making it easier to maintain stable combustion. The stability of combustion of a certain mixture is enhanced by matching the flow velocity with the flame speed and by heating the fresh mixture of fuel and air with a part of the combustion products.

The plots given in Fig. 5.37 show clearly that the decrease of the primary air swirl will permit ignition of weak mixtures. This is due to the fact that longer flames result with the decrease in primary air swirl. This provides a better chance to burn a mixture that contains less amount of fuel by prolonging the period during which it can burn. Decreasing primary air swirl may thus be a useful approach if the heating value of the fuel is less than the limit stated earlier of 6.62 MJ/m^3 (178 Btu/scf) or if lean mixture exists due to load conditions.

During the course of the stability loop tests, it was found that the range of ratios of ignitable mixtures is very sensitive to the procedure followed. While all results plotted in Fig. 5.37 were obtained from tests that were carried out following the procedure outlined in Appendix B, some trials were made to avoid the detonation blast which occurred when the rich limit was reached. These trials were done by first allowing an ignitable mixture to flow through the combustor, and then reducing the air flow rate until flame blowout occurred. The results obtained following this procedure were different by a large margin (~50%) from those obtained earlier in Fig. 5.37 and they lacked repeatability as well.

These tests for the stability loops revealed one more important effect of swirl and hence, of the recirculation zone on combustion. The range of ratios of ignitable mixtures could have been wider (higher values of air/fuel ratio at the weak limit) if the air flow to the premixed zone was reduced or eliminated. Also, redistribution of the total air among the various zones

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could result in a wider ignitable range, particularly if a higher percentage of air was admitted downstream at the dilution zone.

It is important to point out here that due to the way in which the combustor was set up, it created some inherent error in the flow rate of air at very low flow rates. This resulted from the connection of the exhaust end of the combustor to the exhaust line of the laboratory. This exhaust line contained a fan to pump the exhaust gases outside the building. This resulted in a small amount of air flowing inside the combustor even when the blower was shut off. This amount was very difficult to measure due to its dependency on the number of engines that were operating in the laboratory at the time of testing. It is believed that this flow of air affected the rich limit results of the ratios of ignitable mixtures range, particularly due to the small air flow rate measured at this region.

5.7 Gas Analysis

Initially, gas analysis was intended to aid in developing chemical kinetics information through which the gas temperature distribution throughout the combustion space could be obtained and also to evaluate the combustion efficiency based on the amount of unburned fuel. However, upon starting to conduct the gas analysis, it was found that both carbon monoxide and hydrogen (which are the major species in the kinetics study) could not be detected or they vanished very quickly inside the combustor even at the first two measuring sections. At these sections, only a trace amount was found which was so small that it was not possible to distinguish its peak from the noise of the gas chromatogram at the base line at attenuation setting of 1. This was particularly true for hydrogen analysis. Due to these findings, plans to conduct complete gas analysis throughout the combustor (to include CO and H₂) were abandoned. Gas analysis for all detectable gases was carried out for the entire combustion space for run # 10 and the results were considered representative of the variation in gas composition throughout the combustion space. A complete gas analysis (including CO and H₂) was carried out for the first section alone, with various swirlers.

Fig. 5.38 shows the results of gas analysis for run #10 plotted at the

various sections along the combustor. The analysis shows only CO_2 , N_2 and O_2 concentration, since no H_2 was accurately detected. The results for CO are given in Fig. 5.39. Fig. 5.38 shows that the concentration profiles of CO_2 almost coincide with the temperature profiles (in terms of shape). This observation is enhanced by the fact that the completeness of combustion can be indicated by either profile. The figure also indicates the decline in the concentration of CO_2 as the flow proceeds downstream towards the combustor exit, while the concentrations of O_2 and N_2 increase and become more uniform. This, of course, is a direct result of mixing the secondary and the dilution air as the combustion products flow downstream to the combustor exit.

Fig. 5.39 gives a plot of CO concentration at the first measuring section in the combustor, resulting from various swirlers. The figure indicates clearly that the decay of CO is faster with increase of primary air swirl. Due to the enhanced mixing rates by the increase in swirl, the reactivity of the mixture becomes higher. The fastest consumption of CO was achieved with the 60° swirler (where no trace of CO was detected at the first measuring section).

Utilizing the gas analysis results at the first measuring section, together with the results on axial velocity component distribution in a heat balance equation, the average temperature at the first section was calculated (see Appendix C for details). It was found that the temperature at the section was only $\sim 10^\circ\text{C}$ different from the average value calculated from the measured temperatures, at the same section. In these calculations, the heat loss from that part of the combustor surface was approximated by measuring the surface temperature and considering only the radiation heat loss. The surface temperature was estimated by temperature indicating crayons (Tempilstiks) manufactured by Omega Engineering Incorporated.

If complete results of gas analysis (include CO & H_2) were available, it could be possible to obtain the flame length and boundary. It is usually considered in combustion analysis that the flame ends at points where the combustion is 99% complete [8]. Considering the air/fuel ratio used in this study and the air flow rates, the combined total concentration of both CO and H_2 would range from $\sim 10.0\%$ (by mass) at the combustor head to $\sim 1.0\%$ (by mass)

after the dilution air has been added if no combustion takes place. While these proportions are based on gravimetric analysis, the order of magnitude remains the same with volumetric analysis for CO but it will be one order of magnitude higher for H_2 . Therefore, in order to obtain the flame boundary it would be necessary to detect CO and H_2 if present in proportion close to 0.05% (by volume) in the primary zone or 0.005% in the dilution zone. This explains the difficulty in obtaining the gas analysis during this study, and this problem would be overcome only if a very sensitive and accurate gas analysis facility was available. With the existing gas analysis facility at its maximum resolution, CO can be only detected, if present, in proportions higher than 0.01% by volume and H_2 is higher than 0.05%. The error, however, even at these proportions is quite high due to the calibration of the gas chromatograph. Calibration of the chromatograph is done with attenuation setting in the neighbourhood of 128-64 by injecting 1 cc of standard sample. At this range of resolution the peaks for most gases analyzed will be within the page width. The analysis of CO & H_2 with the above mentioned proportions would require attenuation setting of 1. At this attenuation setting, the noise level of the chromatogram base line is quite high and almost of equal amplitude to the minimum possible percentage of CO and H_2 .

The above analysis implies that the absence of CO and H_2 at the second measuring sections, even with the given accuracy of the gas chromatograph, indicates that the combustion was 99% complete by the second section for all the runs. This also means that the combustion efficiency was 99% or better for all the runs that were reported in this study.

5.8 Optimum Operating Conditions and the Consequent Trade-offs

The experimental results presented in this chapter show that under a certain set of operating conditions, one or more aspects of the performance are improved but others are not. This section summarizes the effect of changing the independent parameters, dealt with in this study, on the performance of the combustor. This is followed by an outline of a selection of the optimum operating conditions for the test combustor and the trade-offs that are expected to be made.

5.8.1 Summary of the Results of the Study of Independent Parameters

5.8.1.1 Air Distribution Among the Combustor Zones

Maintaining the same total air flow rate to the combustor as for run #1 (which had equal proportion of air to the secondary and dilution zones) but admitting a larger proportion (53.3%) in the secondary zone than in the dilution zone (26.7%) resulted in lower liner temperature, more uniform profiles of the temperature and velocity components through the combustion space (except at the exit section). This is due to the larger amount of air that is mixed with the combustion products in the secondary zone where the mixing rates are higher (because of the high turbulence in the recirculation zone and in its neighbourhood). Also, the values of the radial and circumferential velocities were lower and decayed to values close to zero as the flow proceeded downstream to the combustor exit. Since proportionately less air is added in the dilution zone, the resulting disturbance to the flow is reduced as the flow approaches the combustor exit.

However, when the proportions of air admitted to the secondary and dilution zones were equal (40%), the combustor efficiency was higher (in comparison with the cases where the proportion of air in the secondary zone differs from that in the dilution zone). Also, the temperature profile and the axial velocity profile at the exit section were more uniform. The increased penetration ability of the dilution air in this case, causes better mixing of the combustion products and air in this region before the combustor exit section. Fig. 5.40 shows a bar diagram summary of the main results regarding air distribution among the combustor zones.

5.8.1.2 Total Air Flow Rate

If all other independent parameters are held constant, reducing the total air flow rate to the combustor (as in run #5) from that of the base run (#1) resulted in lower, more uniform velocity and temperature values throughout the combustion space and a lower temperature near the liner wall. The reason for the reduction in velocity values is due to the reduction in

flow rate. The drop in temperature is mainly because the ratio of heat loss from the combustor surface to the energy input from the fuel at these conditions is higher. This is due to the fact that the heat losses depend on the surface area and temperature of the combustor surface. The surface area is constant and the surface temperature did not change appreciably upon reducing flow rates. This implies that the heat losses practically remained constant while the input energy was reduced, hence the higher proportion of heat loss to the energy input in this case. Corresponding to the decrease of temperature, since the pressure is approximately the same, the density is increased. Consequently, for the same overall flow, the axial velocity is reduced. Also, with the reduced level of temperature, the flow is more homogeneous with the same degree of mixing.

On the other hand, the higher total air flow rate of the base run (compared to that of run #5) resulted in higher combustor efficiency and in more desirable temperature and axial velocity profiles at the exit section. The reason for the higher combustor efficiency is due to the lower proportion of heat loss from the combustor surface as explained above. Also, the higher flow rate (therefore higher velocity of the entering dilution flow) results in increased penetration of the dilution air, causing more uniform velocity and temperature distributions near the combustor exit. Fig. 5.41 gives a bar diagram illustration of the main results of the effects of changing the total air flow rate.

5.8.1.3 Fuel Flow Rate (A/F ratio)

With all parameters as for the base run (#1), except the fuel flow rate which was increased for run #4 (reducing the A/F ratio), lower values of the radial velocity component (almost near zero) were observed along the combustor axis. This is mainly due to the higher temperature levels obtained in this case which enhance the buoyancy effect creating a positive radial component in the upper half of the combustor that counteracts the negative component resulting from the air admission. This is shown in the radial velocity profiles, Fig. 5.21, as a negative value in the lower half of the combustor while a positive value exists in the upper half.

On the other hand, reducing the fuel flow rate as in the base run (compared to run #4) resulted in lower temperature levels throughout the combustor, higher combustor efficiency, more uniform profiles of temperature and axial velocity at the exit section and also a lower axial velocity throughout the combustion space (which means less pressure drop). The reason again is the lower heat loss from the combustor surface due to the reduced temperature levels. Also, the lower temperatures result in lower axial velocity (as understood from the principle of conservation of mass and the equation of state). The lower temperature level as noted earlier, produce more homogeneous flow for the same mixing, and hence better temperature and axial velocity profiles result at the exit section. Fig. 5.42 shows a bar diagram summarizing the main results regarding the effect of fuel flow rate.

5.8.1.4 Fuel Heating Value

Reducing the fuel heating value from that of the base run (as in runs 6 and 7) resulted in lower, more uniform temperature and velocity values throughout the combustion space and lower average axial velocity at all measuring sections (which implies lower pressure drop). The drop in temperature is a result of the reduced energy in the fuel and the drop in flame temperature. This, in turn, causes the velocities to be lower and hence the completeness of mixing is easier to accomplish due to the lower gradients of temperature and velocity.

Increasing the fuel heating value from 6.62 MJ/m^3 (run #7) to 7.78 MJ/m^3 (run #6) to 10.98 MJ/m^3 (run #1) resulted in increased combustor efficiency and stability and more uniformity of axial velocity and temperature profiles at the exit section. The improved combustor efficiency and stability are due to the increase in the flammable range as the heating value increases, as well as due to a greater ease of ignition (due to higher proportion of H_2). Stable combustion results in more homogenous rise in temperature as it avoids hot spots thus causing more uniform temperature and velocity profiles. Table 5.2 presents a summary of the results of the effect of the fuel heating value on the performance of the combustor.

5.8.1.5 Swirl

Reducing the vane angle of the swirler from 60° at the base run to 45° , 30° and 0° (while holding all other independent parameters constant) broadened the A/F range for which ignition is achievable but resulted in lower values of the circumferential velocity component throughout the combustor. This increase in range of ignitable mixture-ratios is due to the increase in flame axial length which allows a longer time for the fuel-air mixture to ignite and also increases the distance along which the flow velocity can be matched with the flame speed. The reason for the reduction in the circumferential velocity component as the swirler vane angle decreases is obvious since the circumferential component is proportional to the sine of the vane angle.

However, increasing the swirl, by increasing the swirler vane angle (0° , 30° , 45° and 60°), resulted in increased combustor efficiency, more stable combustion and also avoided blowout and flashback. The increase in combustor efficiency and stability is a direct result of the broadening of the recirculation zone (in the primary zone of the combustor) upon increasing the swirler vane angle. This wider recirculation zone acts as an improved flame holder delaying flashback and blowout. Table 5.3 presents a summary of the combustor performance upon changing the swirler vane angle.

5.8.2 Optimum Operating Conditions

Based on the above discussion of the effects of the various independent parameters the operating conditions for optimum performance of this combustor may be summarized as follows:

- a. The air distribution among combustor zones should be higher in the secondary zone ~ 53.3% vs. 26.7% in the dilution zone.
- b. The air mass flow rate should be such that the mass flow rate per unit area in the primary zone is $\geq 1.0 \text{ kg/s m}^2$. However, this value should not exceed 7.0 kg/s m^2 or excessive pressure losses

will take place.

- c. The fuel flow rate (or fuel/air ratio) should be the lowest possible (A/F ratio = 38.14 at heating value of 10.9 MJ/m^3).
- d. The fuel heating value should not be less than 6.62 MJ/m^3 if a portion (~4%) of the combustion air is to be premixed with the fuel.
- e. The primary air should be given a swirl equivalent to or greater than that obtained with a 30° vane angle swirler.

5.8.3 Consequent Trade-offs

The conditions selected above were based on their effect on the flow in the combustion space during the experimental study. However, some trade-offs have to be made because of the following disadvantages which may result from this selection.

- a. Deviating from the conditions of run #1 implies a decrease in the combustor efficiency and worsening of the shape factor profile.
- b. While swirl is needed for stable combustion it reduces the range of ignitable mixtures in the combustor.
- c. Utilizing the highest possible air flow rate might be in conflict with the load at some instant.
- d. The use of the minimum possible fuel/air ratio might be in conflict with the load demand.
- e. Limiting the heating value to be higher than 6.62 MJ/m^3 excludes several cheap fuels from being applied for electricity generation.

Although several disadvantages have been cited above, their effect is not very serious. A decrease of the combustor efficiency and deterioration of the shape factor profile are the major drawbacks upon deviating from the conditions of the base run (#1). It is believed that the redesign of the flow

reduction area near the combustor exit will improve the shape factor and temperature profile in this region. This redesign could be achieved by making the area reduction more gradual over a longer length of the liner and with a streamline wall contour rather than straight wall. Also a larger combustor will result in a better combustor efficiency than that obtained during experiments on the test combustor of this study. The lower surface area/volume of a larger combustor will result in a lower proportion of heat loss from the surface and hence better combustor efficiency.

5.9 Evaluation of Velocity Measurements

Having presented the velocity profiles in the axial, radial and circumferential directions at the various sections of the combustor during the experimental study, it may be noted that some discrepancies exist in the shape and sense of these profiles. This section will present an evaluation of the velocity measurements based on the conservation laws, since there was no other method of measurement against which to check the five-hole probe measurements.

5.9.1 Evaluation of the Axial Velocity Results

In order to evaluate the values obtained for the axial velocity, the continuity principle was applied at all the measuring sections for the ten runs that were presented in the previous sections. Appendix C gives a complete illustration of the approach and technique that was followed to calculate the mass flow rate at each section in addition to a summary of the results. The mass flow rate at each section was calculated from numerical integration of the axial velocity-density product over the area of the section. This mass flow rate was compared to the expected mass flow rate, from the input values, which should exist at the same section. The mass flow rate which should exist at any section was calculated based on the air proportion at the section, the open hole area and the assumption that the discharge coefficient is constant for all the holes of a particular zone in the combustor. Table C.3 in Appendix C gives a summary of the results

expressed as percentage difference in the mass flow rate for all ten runs. The last column and last row of Table C.3 contain the mean value of the percentage difference in one run through all sections and in any section from all the runs respectively. The agreement between the two approaches of calculating the mass flow rate at the combustor sections is quite close at the exit section where the maximum percentage difference was $\pm 2.5\%$. The difference, however, was higher in the combustion space.

It could be noted that while the difference seems high at some sections, the average of the ten runs for the ten sections is about 2.3%. The maximum average percentage difference for any run is about 4.6%. The maximum average percentage difference at any section from all the runs is about 11%, at section #4. The position of this section, however, might have contributed to this due to its location between the secondary and dilution zones and the possibility of air leaking from one zone into the other. The variation of these average percentage differences from one section to the other reveals that generally, for sections upstream of and including #3, the calculated mass flow rate from the measurements is less than what is expected from the amount of air admitted up to this section. The opposite trend is found for sections #4 to #9 to be generally true. On average, the axial velocity measurements obtained with the five-hole probe, therefore, agree very well with values deduced based on mass flow rate input and computed distribution.

5.9.2 Evaluation of the Radial Velocity Results

Referring to the plots of the radial velocity which were presented in the previous sections, it can be observed that there is a value for the radial velocity component that exists along the entire section even at the axis, with no change of sense for some profiles. At first glance, this might imply a discrepancy and the need for a source or sink flow at the axis to have such a profile. However, this profile can be reasonable if the velocity gradient in the other two directions is large enough to counteract the existence of a value for the radial component near the axis.

In order to check the validity (or otherwise) of the distribution obtained, the profiles which showed such discrepancy were examined in the

light of the continuity principle near the combustor axis. Indirectly this checked also the other two velocity components, namely the axial and circumferential. In order to apply the continuity principle, a semi-cylindrical control volume was considered to exist at the combustor axis between the section where the radial velocity profile shows a discrepancy to the next section along the combustor axis. The diameter of the control volume was taken to equal its axial length. This shape of the control volume allowed for the inclusion of the three velocity components as well as for the lack of symmetry around the combustor axis for the various radial profiles. The sections which showed discrepancy and required this check are #9 for run #1 and section numbers 1, 6, 7 and 8 for run #2 and section #9 for run #5. The approach was to calculate the rate of mass flowing into the control volume and the rate of mass leaving the control volume (through its surface). The calculation procedure and results are outlined in Appendix C. From these results, it is clear that the largest difference of the mass rate flowing into and out of a control volume is less than 40%. Considering that this is the worst case and the values throughout the combustor and for all the runs are within 5%, the radial velocity profiles, even with their given configuration, satisfy the continuity principle and therefore they are physically acceptable. It is worth noting here that the values of the radial component are dependent on the calculations of the pitch angle which depends on the pressure difference $P_4 - P_5$ from the five-hole probe. For very low velocities less than 1.0 m/s the accuracy in measuring $P_4 - P_5$ is lowered due to the limitations of the manometers' accuracy as discussed in Chapter 4.

5.9.3 Evaluation of the Circumferential Velocity Results

The presence of a swirler at the inlet of the primary air, implies that a circumferential component for the velocity vector should exist for the flow near the primary zone in the same direction of the swirler vane angle. This component should decay to a zero value near the combustor axis and as the flow proceeds downstream to the combustor exit, it should be near zero along the entire section of the combustor. However, the circumferential component as presented in the profiles given in this study does not conform to the above

mentioned behaviour and even at most of the sections show a negative value (opposite to the swirler direction). In order to check the validity of the circumferential velocity component profiles, the axial flux of angular momentum was calculated for all the runs at all the sections in an approach similar to that followed in obtaining the mass flow rate by integrating the required quantity over the area of the section. However, it should be mentioned that the shear stresses' and pressure differences' effect on the axial flux of angular momentum were not taken into consideration. The axial flux of angular momentum calculated from the integration was compared to the axial flux of angular momentum supplied at the combustor head via the swirler to the primary air. Appendix C outlines in more detail the procedure and results obtained for the axial flux of angular momentum at the various sections for the ten runs.

Comparing the results obtained for the axial flux of angular momentum from integrating the values (from measurements) at the various sections to the axial flux of angular momentum at the combustor inlet (Table C.6) shows that the two values do not agree for most of the cases. This implies that the circumferential velocity component may have been measured in the opposite sense to which it existed. However, it should be stated here that the inlet flux of angular momentum values are, in general, less than 8×10^{-5} Nm (compared to an axial momentum multiplied by combustor radius of 3.4×10^{-2} Nm). If during fabrication of the combustor the holes which admit the secondary and dilution air are 5% off the radial direction (due to misalignment of the drill for example), that would result in an axial flux of angular momentum of 1.2×10^{-5} Nm at the third measuring section and higher values downstream. The same could be said about the holes that admit the fuel-air mixture. If this misalignment is in the direction opposite to the direction given by the vanes of the swirler, this could result in a negative circumferential component. It should be noted that the values of the axial flux of angular momentum caused by a 5% error in fabrication of the combustor would be of the same order of magnitude as the inlet value of the axial flux angular momentum at the combustor inlet. This could be one of the main reasons for the large discrepancy in the circumferential velocity

measurements. The fact that pressure difference, particularly at the recirculation zone, and shear stress effects were not accounted for in calculating the axial flux of angular momentum, could add to the difference between the two values especially with large temperature gradients in the combustion zone. Also, add to this the lack of sensitivity of the 5-hole probe in the highly turbulent and recirculated flow regions. It should also be noted that the circumferential component is about an order of magnitude (or more) less than the axial velocity component so that using the two in a conservation equation to check the validity of the circumferential component may exaggerate the uncertainty of the circumferential component. This in most cases resulted in attempting to measure a pressure difference $P_4 - P_5$ of less than or in the neighbourhood of 0.001 in. of water. At this pressure difference, the manometer accuracy could reach 100% or more (as discussed in Chapter 4).

However, since there is no other approach or proof (as a measurement with another technique, for example) to justify the circumferential velocity profiles as they were obtained, they were given as dotted lines to reflect this uncertainty.

In order to evaluate the performance of the 5-hole probe in measuring the velocity vector in highly turbulent combustng flows (that involve a recirculated region), it could be said that while the response was quite satisfactory for the axial velocity component and reasonable for the radial component, through applications of the continuity principle, the circumferential velocity component could not be proven to be satisfactory (or otherwise). The variation in the reliability by which the 5-hole probe responds to the three components is mainly due to the variation in their magnitudes, since the uncertainty in reading a manometer is approximately the same whether the reading is 1 cm or 0.025 cm. The three components should in general have comparable magnitudes to each other and each should not be less than 10 m/s if overall reliability is to be expected by this method. Lower velocities will require high precision manometers which, if the pressure difference is below 0.0254 mm water (0.001 inch water), becomes impractical and cannot be measured accurately with ordinary inclined (or Hook's)

manometers. The accuracy of the five-hole probe for yaw angles beyond 40 degrees is degraded highly and the probe should not be used beyond this yaw angle. A five-hole probe requires calibration, which is normally done in flows that are uniform in velocity magnitude as well as at homogenous temperature and composition, suggests another possible source of error when it was used in the present study with its highly sheared flow. The five holes on the probe occupy a circle of about 5 mm diameter. Therefore, placing this probe in a flow that has a large thermal, dynamic and chemical composition gradients may lead to another source of error as each of these holes measures in fact a flow that is different from the others.

The above statements would imply that for this research, the values of the axial velocity could be taken as generally acceptable, the radial velocity are acceptable but with less certainty than the axial velocity. However, the circumferential velocity values should be treated with caution unless some other means is developed to prove them. Also, it is not recommended, as was found during this study, to use a five-hole probe of this size (9 mm diameter) in a flow that has large gradients of temperature, velocity and composition. In general, combustng and recirculated flows would be more accurately measured using optical methods which do not require insertion of any hardware in the flow. This is achieved now by the laser velocimeter and other laser anemometry. However, the cost of such a system puts it beyond consideration for this project. Therefore, the five-hole probe was used to derive as much information as possible from the measurements.

5.10 Comparison of Experimental Results with those of Other Investigators

Highlights of earlier results upon combustion of low-medium value gases were reviewed in Chapter 2. In this section comparison of the results obtained by earlier investigators and the present results are given. These comparisons will include the effect of parameters that were investigated in this study in the case of gaseous fuels other than low-medium heating value gases.

The results of temperature profiles agree in terms of their shape with all

the previously given temperature profiles in combustors. This is true not only in the case of experimental studies, but also in the case of theoretical analyses of the progress of the radial temperature profile as the flow proceeds downstream to the combustor exit. Swithenbank et al. [112] presented a study for obtaining the flow characteristics in a combustor by using a series of reactors to represent the flow in the various zones of the combustor (see Fig. 6.2 and Chapter 6 for more detail). In their study, the predictions from such an analysis agreed with the temperature traverses taken inside their combustor. These temperature traverses are reproduced from Ref. [112] in Fig. 5.43 and it is clear that they resemble to a great extent the traverses obtained in the present study.

Studies on combustion of natural gas by Shoffstall [101] and city gas by Lockwood et al. [68] showed that swirl has a great effect on the flame geometry and combustion characteristics. Lilley [64] found that the flame length depends on the amount of swirl imparted to the combustion air. The degree of swirl affects the size and shape of the recirculation eddy [65]. This recirculation eddy has a significant contribution to flame stabilization and completeness of combustion by providing a zone of reduced velocities where the flame speed can be matched with the flow speed [15,57]. All these findings are in full agreement with the results of this work. Schwartz [97-99] found that adding swirl to the combustion air resulted in more stable combustion and smoothness of operation, avoided detachment before blow off and resulted in a less noisy combustion. This is the experience in this study as outlined in section 5 of this chapter. Reducing the swirler vane angle to zero resulted in severe stability problems in the combustion space.

The fuel heating value was found to have considerable effect on the combustion of low-medium heating value fuels. The minimum possible limit found during this work, 6.62 MJ/m^3 (178 Btu/scf), was somewhat higher than what other investigators have reported. Pillsbury et al. [91] reported a minimum heating value of 4.03 MJ/m^3 (108 Btu/scf) while Battista and Farrell [5] found a value of 3.35 MJ/m^3 (90 Btu/scf) to be the minimum heating value. The reason may be due to the premixing of the fuel with some amount of air in this work (which dilutes the fuel and lowers its heating value). The

effective heating value, if the air was considered as a diluent to the fuel, is 3.7 MJ/m^3 rather than 6.62 MJ/m^3 . This value of 3.7 MJ/m^3 falls within the range of minimum heating value reported by other investigators. It could be also due to the fact that none of the previous investigators made measurements inside the combustion space to realize the oscillations that occurred. Another reason could be the small size of the test combustor in this work, compared to industrial combustors used by the other investigators. The small test combustor in the present work posed difficulties for maintaining combustion efficiency at very low heating values.

The air distribution among the various zones of the combustor reported in this work is rather different from what is conventionally used for industrial or even aircraft combustors. However, during the tests of a high temperature combustor designed to operate with low heating value gas, Koblish and Nucci [56], found that a larger amount of air was needed in the early stages of the combustor and less air was admitted in the dilution zone. This agrees with the findings of this study (mentioned in the first section of this chapter) that higher proportion of air in the secondary zone resulted in better performance of the combustor.

The somewhat narrow range of ignitable mixtures obtained inside the combustor burning low-medium heating value fuel was in agreement with the work carried out by Vogt [116].

The value of the traverse number found for this test combustor was within the acceptable range for industrial gas turbine combustors [40]. Although some runs resulted in a high traverse number, the condition of the run which led to this can be avoided and a lower traverse number can be achieved if the region of reduction in flow area before the combustor exit is aerodynamically improved by making the area change over a longer length of the liner and by making the wall contour streamlined instead of straight wall.

The combustor efficiency for the various runs is given in Table 5.1. This efficiency was calculated from experimental results of the temperature traverse at the exit section, the flow rates of fuel and air to the combustor, and using gas tables to calculate the standard enthalpy for the various components of the inlet mixture to the combustor or the products of combustion

at the exit section. The ratio of the total enthalpy calculated at the exit section to the enthalpy calculated at the inlet section gives the combustor efficiency (see Appendix C for more details and example of calculations). This was termed combustor efficiency rather than the combustion efficiency due to the consideration of the heat loss through the surface of the combustor from the enthalpy of the outflowing stream. This heat loss is proportional to the combustor surface area, and the difference between the fourth powers of the absolute ambient temperature and the surface temperature of the combustor for the radiation portion while the convective part of this heat loss is proportional simply to the temperature difference. Since the combustor area was fixed and the ambient temperature was practically constant during the various tests, the loss was mainly proportional to the combustor surface temperature raised to the fourth power. It is obvious also that the surface temperature of the combustor will depend on the temperature distribution inside the combustor and on the fuel flow rate. The gas analysis, as mentioned earlier, indicated that the combustion efficiency for this combustor is higher than or close to 99%. This combustion efficiency is considered satisfactory for a development combustor.

Having presented the results of the experimental work of this study, it is worthwhile to evaluate the contribution of this work in the light of its objectives and the earlier work summarized in Chapter 2. While all the previous work lacked access to the combustion space for measurements of various properties, the test combustor developed in this study facilitated this requirement quite successfully. The combustor allowed admission of a portion of the combustion air to be premixed with the fuel as well as the option of varying the air proportion among the combustor zones. Parameters that are more likely to be encountered in practical applications of combustors which burn this class of fuels (low-medium heating value), as they are produced from a gasification process or as a by-product from an industrial process, were analyzed in the present study. Changes in parameters such as fuel flow rate, fuel heating value and air flow rate may be encountered upon variation of load demand or efficiency of fuel delivery system. Swirl, which is an important parameter in combustor design was thoroughly analyzed to find

the optimum amount of swirl needed for fuels in the low-medium heating value range. The ability to perform various measurements inside the combustion space helped to explain in better detail the effect of the previously mentioned variables, on combustor performance. The various profiles presented in this chapter have suggested possible design modifications to improve the performance of the combustor. The limitation encountered with the heating value due to premixing part of the combustion air with the fuel made it obvious that this premixing should be avoided. Premixing should be applied only when the heating value is above the minimum mentioned in this study and when it is needed to reduce NO_x emissions. Higher percentage of air is recommended at the early stages of combustion (not premixed), particularly if the combustor is to operate at higher temperatures. In some cases, the combustor may be forced to operate at stoichiometric conditions where all the air should be admitted at the primary and secondary zones, with no dilution air. The present study showed clearly that swirl has a combined influence on the combustor performance. It will affect the flow regimes as well as the thermal patterns inside the combustion space. Swirl, then, should be tailored to the heating value of the particular application. This suggests that a variable swirl facility should be installed in combustion equipment that burn a fuel of low-medium heating value such that swirl can be adjusted according to the properties of the fuel or according to load variations. It would also be desirable, if the combustor (during its operation) would allow redistribution of the combustion air among its zones.

Table 5.1a Summary of Parameters Considered in This Study and the Corresponding Combustor Efficiency for the Various Runs Where Measurements Were Taken Throughout the Entire Combustion Space

Run #	Swirler vanes angle °	$\dot{m}_f$ kg/h	HV MJ/m ³	HV MJ/kg	Fuel Composition			$\dot{m}_a$ kg/h	Air Percentage at Various Zones %				A/F kg/kg	η
					CO ₂ %	H ₂ %	H ₂ %		Pnd	Pri	Sec	Dil		
1	60	1.995	10.89	15.13	45.6	9.2	45.2	76.08	10	10	40	40	38.14	91.1
2	60	1.995	10.89	15.13	45.6	9.2	45.2	76.08	10	10	26.7	53.3	38.14	80.9
3	60	1.995	10.89	15.13	45.6	9.2	45.2	76.08	10	10	53.3	26.7	38.14	78.5
4	60	2.394	10.89	15.13	45.6	9.2	45.2	76.08	10	10	40	40	31.78	74.4
5	60	1.336	10.89	15.13	45.6	9.2	45.2	50.72	10	10	40	40	38.14	85.6
6	60	3.73	7.68	7.43	32.2	36.2	31.6	76.08	10	10	40	40	20.4	84.8
7	60	4.079	6.14	5.16	25.7	49.0	25.3	76.08	4	10	43	43*	18.6	78.6
8	0	1.995	10.89	15.13	45.6	9.2	45.2	76.08	4	10	43	43	38.14	70.5
		1.336											56.95	
9	30	1.336	10.89	15.13	45.6	9.2	45.2	76.08	4	10	43	43	56.95	82.3
10	45	1.336	10.89	15.13	45.6	9.2	45.2	76.08	4	10	43	43	56.95	87.6

* This is the final air flow distribution
See Fig. 5.1 for the various combustion zones

Pnd - Premixed
 Pri - Primary
 Sec - Secondary
 Dil - Dilution

Table 5.1b Conditions During the Additional Ten Runs Where Measurements Were Only Made for Temperature at the Exit Section and the Corresponding Combustor Efficiency From Experiments and From the Combustor Efficiency Formula, Eq. (6.23), for Runs #16-20

Run	Swirler † vane angle	$\dot{m}_f$ kg/h	HV MJ/m ³	HV MJ/kg	Fuel Composition			$\dot{m}_a$ kg/h	Air Percentage at Various Zones %				A/F kg/kg	η	η_F^+
	0				CO ₂ %	CO ₂ %	H ₂ %		Pnd	Pri	Sec	Dil			
11	60	1.336	10.89	15.13	45.6	9.2	45.2	60.08	4	10	43	43	45.50	87.8	—
12	60	1.336	8.58	9.03	35.1	28.7	36.2	76.08	4	10	43	43	56.95	88.9	—
13	60	2.190	10.89	15.13	45.6	9.2	45.2	76.08	4	10	43	43	34.74	82.5	—
14	60	2.554	10.89	15.13	45.6	9.2	45.2	76.08	4	10	43	43	29.80	72.4	—
15	60	1.760	10.89	15.13	45.6	9.2	45.2	67.12	10	10	40	40	38.14	88.9	—
16	60	2.20	9.0	10.01	37.0	25.0	38.0	76.08	10	10	40	40	34.58	88.6	86.1
17	45	1.995	10.89	15.13	45.6	9.2	45.2	76.08	10	10	40	40	38.14	83.0	80.3
18	30	1.995	10.89	15.13	45.6	9.2	45.2	76.08	10	10	40	40	38.14	76.1	74.6
19	60	1.835	10.89	15.13	45.6	9.2	45.2	70.0	10	10	40	40	38.14	89.3	89.5
20	60	2.100	10.89	15.13	45.6	9.2	45.2	76.08	10	10	40	40	36.23	89.3	86.1

{ Pnd - Premixed
 Pri - Primary
 Sec - Secondary
 Dil - Dilution
 + Combustor efficiency calculated from Eq. (6.23).

Table 5.2 Summary of results regarding the effect of fuel heating value on the performance of the combustor

Heating value MJ/m ³	Combustion stability	Ignition	η %	Traverse number
10.9	best	easy	91.1	0.055
7.7	better	easy	84.8	0.062
6.6	good	possible	78.6	0.147
6.1 & lower	bad	difficult	----	----

Table 5.3 Summary of results regarding the effect of swirler vane angle (or swirl number) on the performance of the combustor

Swirler vane angle α	Swirl number	Combustion stability	η %	Traverse number
0	0	bad	70.5	0.084
30	1.01	good	82.3	0.107
45	1.75	better	87.6	0.084
60	3.04	best	91.1	0.055

S = measuring section location

Note: an air distribution of 10-10-40-40 means 10% of the total air introduced at the premixed zone, 10% at the primary zone, 40% at the secondary zone and 40% at the dilution zone

Fig. 5.1 Combustor zones and measurement sections of the test combustor

$HV=10.9 \text{ MJ/m}^3$

Swirler vane angle 60°

$\dot{m}_a=76.1 \text{ kg/h}$

$A/F=38.1 \text{ kg/kg}$

Fig. 5.2 Temperature profiles at selected cross sections along the combustor for three air distributions used in runs 1, 2 and 3

Fig. 5.3 Shape factor profile at exit section for three distributions of air among combustor zones used in runs 1, 2 and 3

a) TEMPERATURE NEAR UPPER WALL

b) TEMPERATURE AT CENTERLINE

c) TEMPERATURE NEAR LOWER WALL

Fig. 5.4 Temperature distribution near upper wall, centerline and lower wall of combustor along its axis for three proportions of air distribution among combustor zones used in runs 1, 2 and 3

$$HV = 10.9 \text{ MJ/m}^3$$

Swirler vane angle 60°

$$\dot{m}_a = 76.1 \text{ kg/h}$$

$$A/F = 38.1 \text{ kg/kg}$$

Fig. 5.5 Axial velocity profiles at selected cross sections along the combustor for three air distributions used in runs 1, 2 and 3

Fig. 5.6 Radial velocity profiles at selected cross sections along the combustor for three air distributions used in runs 1, 2 and 3

$$HV = 10.9 \text{ MJ/m}^3$$

$$\text{Swirler vane angle } 60^\circ$$

$$\dot{m}_a = 76.1 \text{ kg/h}$$

$$A/F = 38.1 \text{ kg/kg}$$

Fig. 5.7 Circumferential velocity profiles at selected cross sections along the combustor for three air distributions used in runs 1, 2 and 3

a) AXIAL VELOCITY

b) RADIAL VELOCITY

c) CIRCUMFERENTIAL VELOCITY

Fig. 5.8 Velocity components at combustor centerline along its axis for three proportions of air distribution among combustor zones used in runs 1, 2 and 3

Swirler vane angle 60°

Air Distribution 10-10-40-40%

HV=10.9 MJ/m³

Fig. 5.9 Temperature profiles at selected cross sections along the combustor for two air flow rates used in runs 1 and 5

Fig. 5.10 Shape factor profile at the exit section, for two air flow rates used in runs 1 and 5

a) TEMPERATURE NEAR UPPER WALL

b) TEMPERATURE AT CENTERLINE

c) TEMPERATURE NEAR LOWER WALL

Fig. 5.11 Temperature distribution near upper wall, centerline and lower wall of the combustor along its axis for two air flow rates used in runs 1 and 5

Swirler vane angle 60°

Air Distribution 10-10-40-40%

 $HV = 10.9 \text{ MJ/m}^3$

Fig. 5.12 Axial velocity profiles at selected cross sections along the combustor for two air flow rates used in runs 1 and 5

Swirler vane angle 60°

Air Distribution 10-10-40-40%

HV = 10.9 MJ/m^3

Fig. 5.13 Radial velocity profiles at selected cross sections along the combustor for two air flow rates used in runs 1 and 5

Swirler vane angle 60°

Air Distribution 10-10-40-40%

HV = 10.9 MJ/m^3

Fig. 5.14 Circumferential velocity profiles at selected cross sections along the combustor for two air flow rates used in runs 1 and 5

a) AXIAL VELOCITY

b) RADIAL VELOCITY

c) CIRCUMFERENTIAL VELOCITY

Fig. 5.15 Velocity components at combustor centerline along its axis for two air flow rates used in runs 1 and 5

Swirler vane angle 60°

Air Distribution 10-10-40-40%

$HV = 10.9 \text{ MJ/m}^3$

$\dot{m}_a = 76.1 \text{ kg/h}$

Fig. 5.16 Temperature profiles at selected cross sections along the combustor for two fuel flow rates used in runs 1 and 4

Fig. 5.17 Shape factor profile at the exit section for two fuel flow rates used in runs 1 and 4

a) TEMPERATURE NEAR UPPER WALL

b) TEMPERATURE AT CENTERLINE

c) TEMPERATURE NEAR LOWER WALL

Fig. 5.18 Temperature distribution near upper wall, centerline and lower wall of combustor along its axis for two fuel flow rates (A/F ratios) used in runs 1 and 4

Swirler vane angle 60°

Air Distribution 10-10-40-40%

HV = 10.9 MJ/m^3

$\dot{m}_a = 76.1 \text{ kg/h}$

Fig. 5.19 Axial velocity profiles at selected cross sections along the combustor for two fuel flow rates used in runs 1 and 4

a) AXIAL VELOCITY

b) RADIAL VELOCITY

c) CIRCUMFERENTIAL VELOCITY

Fig. 5.20 Velocity components at combustor centerline along its axis for two fuel flow rates used in runs 1 and 4

Swirler vane angle 60°

Air Distribution 10-10-40-40%

HV = 10.9 MJ/m^3

$\dot{m}_a = 76.1 \text{ kg/h}$

Fig. 5.21 Radial velocity profiles at selected cross sections along the combustor for two fuel flow rates used in runs 1 and 4

Swirler vane angle 60° Air Distribution 10-10-40-40%
 $HV = 10.9$ MJ/m³ $\dot{m}_a = 76.1$ kg/h

Fig. 5.22 Circumferential velocity profiles at selected cross sections along the combustor for two fuel flow rates used in runs 1 and 4

Fig. 5.23 Temperature profiles at selected cross sections along the combustor for three fuel heating values used in runs 1, 6 and 7

Fig. 5.24 Shape factor profile at the exit section for three fuel heating values used in runs 1, 6 and 7

a) TEMPERATURE NEAR UPPER WALL

b) TEMPERATURE AT CENTERLINE

c) TEMPERATURE NEAR LOWER WALL

Fig. 5.25 Temperature distribution near upper wall, centerline and lower wall of combustor along its axis for three heating values used in runs 1, 6 and 7

Fig. 5.26 Axial velocity profiles at selected cross sections along the combustor for three fuel heating values used in runs 1, 6 and 7

a) AXIAL VELOCITY

b) RADIAL VELOCITY

c) CIRCUMFERENTIAL VELOCITY

Fig. 5.27 Velocity components at combustor centerline along its axis for three heating values used in runs 1, 6 and 7

Fig. 5.28 Radial velocity profiles at selected cross sections along the combustor for three heating values used in runs 1, 6 and 7

Fig. 5.29 Circumferential velocity profiles at selected cross sections along the combustor for three fuel heating values used in runs 1, 6 and 7

Fig. 5.30 Temperature profiles at selected cross sections along the combustor for four swirlers used in runs 1, 8, 9 and 10

Fig. 5.31 Shape factor profile at exit section for four swirlers used in runs 1, 8, 9 and 10

a) TEMPERATURE NEAR UPPER WALL

b) TEMPERATURE AT CENTERLINE

c) TEMPERATURE NEAR LOWER WALL

Fig. 5.32 Temperature distribution near upper wall, centerline and lower wall of combustor along its axis for four swirlers used in runs 1, 8, 9 and 10

$$\dot{m}_a = 76.1 \text{ kg/h}$$

$$HV = 10.9 \text{ MJ/m}^3$$

Fig. 5.33 Axial velocity profiles at selected cross sections along the combustor for four swirlers used in runs 1, 8, 9 and 10

a) AXIAL VELOCITY

b) RADIAL VELOCITY

c) CIRCUMFERENTIAL VELOCITY

Fig. 5.34 Velocity components at combustor centerline along its axis for four swirlers used in runs 1, 8, 9 and 10

$$\dot{m}_a = 76.1 \text{ kg/h}$$

$$HV = 10.9 \text{ MJ/m}^3$$

Fig. 5.35 Radial velocity profiles at selected cross sections along the combustor for four swirlers used in runs 1, 8, 9 and 10

$$\dot{m}_a = 76.1 \text{ kg/h}$$

$$HV = 10.9 \text{ MJ/m}^3$$

Fig. 5.36 Circumferential velocity profiles at selected cross sections along the combustor for four swirlers used in runs 1, 8, 9 and 10

Fig. 5.37 Stability (blowout) loops for four swirlers with different vane angles used in runs 1, 8, 9 and 10

Swirler 45°

A/F=57.0 kg/kg

Air Dist. 4-10-43-43%

HV=10.9 MJ/m³

Fig. 5.38 Species concentration profiles for run #10 at selected cross sections along the combustor

Fig. 5.39 Carbon monoxide distribution at first section for four swirlers with different vane angles used in runs 1, 8, 9 and 10

Fig. 5.40 Effect of air distribution among combustor zones on the performance

Fig. 5.41 Effect of air mass flow rate on the combustor performance

Run #1	fuel flow rate		2.0 kg/h
	combustor efficiency		91.1%
	traverse number	0.055	
Run #4	fuel flow rate		2.4 kg/h
	combustor efficiency	74.4%	
	traverse number	0.09	

Fig. 5.42 Effect of fuel flow rate on combustor performance

Fig. 5.43 Gas-turbine can; temperature traverses
(temperature in $^{\circ}\text{C}$, Ref. 112) ,

CHAPTER SIX

COMBUSTOR MODELLING AND THEORETICAL ANALYSIS

The experimental approach for the development of a combustor and the investigation of its performance for a certain application was described in the previous chapters. It is obvious that this is an expensive and time consuming approach. Therefore, it would be useful to develop a theoretical technique through which the performance of the combustor can be predicted before its fabrication. A combined approach in which experimental results as well as theoretical models are used to derive empirical formulae is also possible. This is different from the purely theoretical approach since some experiments need to be conducted to determine the empirical constants. It is very useful in predicting the behaviour of existing equipment at operating conditions other than those of testing.

Intensive research in the area of combustion took place in the last two or three decades, in order to understand the process of combustion and the various phenomena that are associated with it. The result of this was the development of a wide variety of models and mathematical descriptions to simulate combustors and to describe the various processes associated with combustion. These models range from zero-dimensional (general or global performance models) to 3-dimensional models. These models and their study have now achieved quite an advanced stage, especially during the last decade.

An overview of the models known at present will be outlined in this chapter. Following this, a method of describing or predicting the performance of a combustor at operating conditions other than those used during the experiments of this study will be presented. In this overview, reference will be made to various review papers [59,65,66,81,83] as well as to some other related studies.

6.1 The Evaluation of Models

In a survey conducted by Odgers [81] regarding the approach used for combustor design, twenty-one companies and research institutes participated. Thirteen of these participants indicated that they use a convenient reaction rate equation (zero-dimensional or global model). The number of users employing other models ranged between one to three.

A model must satisfy the following requirements [81] in order to impress designers and find use in combustor applications:

- a) give accurate predictions
- b) use minimum calculation time
- c) It should be easy to understand, apply and interpret the results
- d) It should be economical for the purposes it will be used for.

If the models are too sophisticated and complicated, they would usually require so much time to obtain predictions with any reasonable accuracy that the cost involved would become prohibitive. Also, the understanding of such models will be difficult and in most cases a consultant would be needed. In spite of such efforts, there is no guarantee that the results can be directly applied to a real combustion system due to simplifying assumptions usually made in developing the model.

The approach attempted in this study was to develop an equation of global form to help design new combustors for use with gasification produced fuels. As a first step, an outline of the known models is given and the approach to be used in deriving a global equation is outlined.

6.2 Zero-Dimensional Models

These models are historically the first to be developed. Generally, they treat the entire reaction zone as a single unit. This unit is assumed to have the general, characteristic behaviour of a perfectly stirred reactor (PSR). In a PSR, velocities, temperature, heat flux, densities and composition are uniform. Models of this type were described by Bragg [13], Avery and Hart [4],

Longwell and Weiss [69] and Vulis [118]. These models were only able to give qualitative descriptions of combustor performance. Greenhough and Lefebvre [35] showed fair agreement between results from models and experiments conducted using a Rolls Royce engine's combustor. Swithenbank et al. [112] introduced a mixing correction by using partially stirred reactors and found that temperatures calculated according to this model agreed with experimental measurements. Starting with the model given originally by Longwell, Odgers and co-workers [19,21,58] carried out very extensive work using a spherical combustor in an attempt to relate the performance of real combustors to the ideal spherical combustor. The result of their work is given as a reaction rate equation in the form:

$$\frac{N}{VP^{2\phi}} = \frac{1.29 \times 10^{10} (n^* + 1) \{5(1 - Y_\epsilon)\}^\phi \{\phi - Y_\epsilon\}^\phi e^{-C/(T_i + \epsilon \Delta T)}}{0.082^{2\phi} Y_\epsilon \{5(n^* + 1) + \phi + Y_\epsilon\}^{2\phi} \{T_i + \epsilon \Delta T\}^{2\phi - 0.5}} \quad (6.1)$$

This equation was derived for propane gas as a fuel. Using the same equation in addition to data from real combustors, the efficiency of combustion chambers (for weak mixtures in the primary zone) was given by Odgers and Carrier [80] in the following form:

$$\eta = 0.911 \log \psi_{300} + 4.01 n - 1.097 + D' \quad (6.2)$$

where, N = air flow rate

V = combustor volume,

P = operating inlet pressure

n = reaction order

ϕ = equivalence ratio

n^* = ratio of inerts/oxygen in air (by volume)

$Y = \phi$ for $\phi \leq 1$, $Y = 1$ for $\phi \geq 1$

T_i = inlet temperature

ΔT = undissociated adiabatic temperature rise

C = fraction of activation energy = E/R

ϵ = fractional oxygen consumption

η = $\log \log 100/\eta$

η = combustion efficiency

ψ = fuel loading = $m_f / V' P^n$

V' = primary zone volume

m_f = mass of fuel

D' = a combustor constant

Eq. (6.2) is a very useful tool in combustor development. Predictions of performance at any condition is possible once D' is known. Therefore, an experiment is needed to obtain the value of D' in order to apply Eq. (6.2).

Results of the other model developed by Greenhough and Lefebvre [35], and applied to a Rolls Royce turbo-annular combustion chamber showed good agreement with experiment. This model is based on the effect of inlet temperature on the loading parameter. At any operating A/F ratio, the efficiency and stability for any combustor (aircraft) could be expressed by plotting a parameter θ against efficiency. This parameter is given as follows:

$$\theta = \frac{A D P^{1.75}}{m} e^{T/b} \quad (6.3)$$

where, A = combustor reference area

D = combustor reference diameter

m = combustion air flow rate

$b = 220 (\sqrt{2} \ln \phi / 1.03)$

6.3 One-Dimensional Models

These models are somewhat more advanced than the previous ones. The flow inside the combustor in this case is divided into various zones. The model utilizes the concept of turbulent mixing to facilitate the use of partially stirred reactors. These models are capable of predicting performance

variables such as blow off, stability limits, combustion efficiency, combustion intensities and overall pressure loss. The combustor is divided into zones each having a certain mass, volume and residence time which are calculated based on geometry, mass flow rate and the entrainment jet theory [112]. Models in this category vary widely in representing the primary zone. For the secondary zone, either a plug flow reactor or a well stirred reactor or a combination of both in series is used. Fig. 6.1 reproduced from a paper by Mellor [73] shows schematically a primary zone and the various modular approaches used. A simple stirred reactor (Fig. 6.1b), was used in the analysis of Pratt et al. [92]. A more sophisticated model using a macro-mixed partially stirred reactor (Fig. 6.1c), was given in the studies by Fletcher and Heywood [30] and by Heywood [42]. A pair of well stirred reactors (Fig. 6.1d), was used by Hottel et al. [44] to represent the primary zone. This representation resulted in a satisfactory qualitative agreement in the case of gas turbine combustors. This approach was used by Hammond and Mellor [38,39] to represent zones 2 and 3. Fig. 6.1e shows the well stirred reactor of Fig. 6.1d replaced by a plug flow reactor with vaporization of fuel allowed in the primary zone as well as in the secondary zone. Fig. 6.1f shows a schematic of a model that includes the effect of droplet evaporation to correspond to zones 1 and 2 in which a well stirred reactor (zone 3) is embedded. Fig. 6.2 reproduced from Ref. [112] shows the inter-connection of reactors proposed by Swithenbank et al. comprising a gas turbine combustor model. This model resulted in temperature predictions which were in accordance with experiment.

Due to the large dependence of NO_x emissions on temperature, one-dimensional models are quite popular [83]. Hung [45,46] calculated NO_x levels for aircraft engines using a one-dimensional model considering instantaneous mixing of the secondary air. The three-dimensional characteristic of the combustor is represented by a quasi-one-dimensional model. Kollrack and Aceto [57] studied recirculation effects in a combustor by solving the equations for chemical kinetics, thermodynamics and gas dynamics. They found that hot gas recirculation caused NO_x to form in an earlier stage, while cold gas recirculation could reduce NO_x by an order of magnitude.

6.4 Two-Dimensional Models

These models apply to cases where steady state flow with axial symmetry can be assumed. This is mainly applicable to a specific category of gas turbine combustors but is more likely to be used for furnaces. This approach was sparked by a group of workers at the Imperial College in London, England, headed by Spalding [33,87]. Over a period of almost two decades, various models were utilized by Spalding and co-workers in an attempt to solve the flow properties of various industrial combustion equipment. The models rely on solving the flow properties by solving the various conservation partial differential equations that describe the chemical kinetics, thermodynamics, heat transfer and fluid dynamics of the combustion equipment. The partial differential equations are transformed into finite difference equations prior to their solution. The technique of solution depends on the type of partial differential equations that describe the flow. Parabolic equations are generally encountered for flows without swirl or where there is no recirculation zone. Elliptic equations are expected for recirculated flows.

Most of the application of these models at the Imperial College was carried out on furnaces or free jet combustors. Experimental work was performed simultaneously to confirm the computation. In most of the cases reported, the experimental measurements were in agreement with predictions.

6.5 Three-Dimensional Models

These models which followed from the previous 2-D models became possible as computers' core capacity became larger and the computation speed increased. Three-dimensional models are more useful for combustor study than are 2-D models. While 2-D models may adequately describe some processes, the general dynamics of the flow demands a 3-D model. This modelling attempt started also at the Imperial College with attempts to apply the models to furnaces by Abou-Elail et al. [1]. This same approach was used later by several research installations. The latest and most complete study is that done by Mongia et al. [76,77] at AiResearch to develop a technique of combustor design and fabrication. The development was carried out in association with an

experimental test rig as well as with a computer code that contained a technique for the solution of the flow inside the combustor in all three dimensions. The design of the combustor in the test rig was made according to this model. Tests on the combustor were performed to determine the validity of the model. The model was modified and the combustor design was readjusted accordingly until close agreement between the model and the experimental results was reached. While a 3-D model was used to describe the combustor performance in terms of efficiency, fluid dynamics and turbulent mixing, only a 1-D or 2-D model was used for other processes. The annulus flow was described by a 1-D model. Liner cooling, transition and liner mixing, emission and fuel injection were all modelled in 2-D form. The flow field characteristics which were computed in 3-D included combustion efficiency, exhaust temperature quality and lean blowout. The final output of the study came in the form of a computer listing that accurately predicted the combustor performance.

6.6 Models for Various Processes in the Combustor

The application of 2-D or 3-D modelling includes the utilization of conservation principles besides other phenomena in order to determine the flow properties that lead to performance estimation. Turbulence is a major parameter in combustion studies. The turbulent eddy viscosity is used in various investigations [68,87,88,104] to describe the effect of turbulence and to include its effect in the computations. Two approaches are normally used to describe the eddy viscosity. The first is through an algebraic model which relates the eddy viscosity to the inlet kinetic energy, local density and combustion chamber dimensions. The second is called the two-equation model or is better known as $k-\epsilon$ model which requires the solution of two extra partial differential equations for the conservation of the turbulent kinetic energy and the rate of its dissipation [104].

The rate of creation of various species is determined through the reaction or chemical kinetics models. The approach to these models is similar to that used for turbulence. The algebraic model is known to depend on statistical information about the chemical reaction. This model is widely used under the

name "Arrhenius model". The second, called the "eddy-break-up" model involves the calculation of the mean square fluctuation component of fuel concentration which can be represented as a partial differential equation in the set to be solved. The model supposes that turbulent entrainment processes that occur at the outer flame boundary result in the formation of gas "parcels" having interleaved layers at different conditions. Fuel rich gas layers may be interspersed with air rich layers, and hot layers may mingle with cold ones [104]. Hence, the chemical kinetic limit is set by the reaction rate but a central point of the model is that there is a second limit, often much lower, set by the rate of dissipation of the fluctuations. This process involves the break-up of large eddies into small ones and although the molecular processes are involved in the final stages of dissipation, they are not controlling ones. The reaction rate is controlled mainly by fuel concentration dissipation rate [70]. The eddy-break-up model is given by relating reaction rate to the local density, fluctuation concentration and turbulent kinetic energy.

The solution of the flow in a combustor also requires a model to describe the liner mixing, another model for heat transfer (mainly that by radiation) from the combustor surface and a model to determine the liner cooling and its effectiveness. The only approach to the computation of flow in combustors that includes all these models is the one described by Mongia et al. [76,77]. Most of the models were 2-D in nature assuming axial symmetry. Some empirical global models exist for these processes as well [33,88].

6.7 Present Contribution

In this study, the first attempt to develop a global equation for the combustor was initiated during the design of the test combustor. It was thought that the size of the combustor could be determined through the use of some empirical formulae. Most of the empirical formulae that are known for the initial sizing of a combustor may be found in a review by Odgers [83]. However, upon applying these formulae for the conditions of the air and the fuel characteristics as well as the fuel flow rate and pressure drop across the combustor used in this work, a wide discrepancy in the combustor

dimensions was the result. An exception is the relationship given by Bragg [13] which gave a reasonable size that agreed with the present design.

The Odgers-Carrier parameter, Eq. (6.2), was used in an attempt to predict the combustor efficiency at conditions other than those used in the equation to calculate the constant D' . The direct application of Eq. (6.2) to calculate the combustor efficiency for run #14 utilizing the combustor efficiency of run #13, to calculate the combustor constant D' , resulted in over estimation of the combustor efficiency (see Appendix E for details of calculations). The reason for the inability of Eq. (6.2) to predict the combustor efficiency for the conditions of this combustor was believed to be due to the high value of the fuel loading factor and because of operating the combustor at atmospheric pressure. The constants of Eq. (6.2) were rearranged to increase the contribution of the fuel loading term being the only variable in the equation. The new form of the equation is given as:

$$\eta = 3.35 \log \psi_{300} + 4.36 n - 1.0 + D' \quad (6.4)$$

Applying Eq. (6.4) resulted in close estimation of the combustor efficiency of other runs such as run #9 (within 2%); however, the fuel composition and primary air swirl as well as total air must be kept the same as the values used to calculate the combustor constant D' .

The approach followed by Kretschmer and Odgers [58] as given by Eq. (6.1) was also examined. The conditions corresponding to run #1 were utilized to calculate the various parameters of Eq. (6.1). The equation was then solved for the oxygen consumption efficiency ϵ (which approximates the combustion efficiency [80]). The resulting value ϵ (or η) was 5.5% (see Appendix E for details of calculations). This value is more than an order of magnitude lower than the value based on an experimental measurements ($\eta=91.1\%$). Hence, this approach is considered invalid for the test combustor with the fuel used in the present study.

Several reasons may be cited to explain the inability of the above mentioned techniques or empirical formulae to size this combustor or predict

its performance. The first is that most of these correlations were based on data collected from aircraft engines which are entirely different from the present combustor application. Since the geometry in those tests did not change, the formulae developed did not contain any parameter that would take into account changes in geometry. Also, the fuel composition was kept the same and that resulted in consistent combustion characteristics of the fuel. The correlation used in the study by Kretschmer and Odgers [58] and by Kretschmer [59] was based on saturated hydrocarbon fuels and mainly on propane gas with which many combustors have been tested during the past 35 years.

The attempt to utilize the blowout data obtained from the experimental tests on this combustor to develop a global equation for this effect did not result in a proper form. The tests were carried out with various swirlers and hence the geometry was different from one run to another. Although the data showed small scatter in general, the weak limit for the zero swirler showed somewhat larger discrepancy compared to the other swirlers.

The objective was then redirected and instead of developing a global equation to predict the combustor performance at any condition, it was decided to develop a combustor efficiency equation. This equation was expected to predict the combustor efficiency at conditions other than those of testing, but for the same combustor. Three approaches were examined for this purpose and they are outlined in the following sections.

6.7.1 Inlet Temperature of Fuel

Low heating value gas may need to be introduced at an elevated temperature to bring it to ignition conditions or it may be fed directly from the gasifier into the combustor in which case it would already be at an elevated temperature. Since the relationship developed by Odgers and Carrier [80], Eq. (6.2), relies on one constant for the combustor to describe the efficiency at other conditions, it is needed to calculate the fuel loading factor at the base temperature (300 °K). The fuel loading factor in Eq. (6.2) could be corrected for inlet temperature through the use of the following form given by Kretschmer and Odgers [58].

$$F = \frac{\left(\frac{m_f}{V P^{2\phi}} \right)_{T_i}}{\left(\frac{m_f}{V P^{2\phi}} \right)_{300 \text{ K}}} = 10^{-3.05} Y^{-1.205} T_i^{1.232} Y^{-1.205} \quad (6.5)$$

where, $Y = \phi$ for weak mixtures

$= 1$ for rich mixtures

T_i = combustor inlet temperature $^{\circ}\text{K}$

Due to the lack of data available for low-medium heating value fuels at elevated temperatures, it was not possible to prove or disprove the validity of Eq. (6.5).

6.7.2 Air Inlet Pressure

The reaction rate equation, Eq. (6.1) given earlier, which was derived by Kretschmer [59] and by Kretschmer and Odgers [58], provides a means of calculating the combustion efficiency at various inlet pressures. Substituting the input conditions in Eq. (6.1), the only remaining unknown is the oxygen consumption efficiency which can be solved for. In their analysis, Kretschmer and Odgers assumed that the oxygen consumption efficiency is equal to the combustion efficiency. Although Kretschmer and Odgers reported to have obtained reasonable results from this approach, the author was not able to get realistic results from this approach at atmospheric pressures as explained earlier (see also Appendix E). The equation, however, was not attempted at high pressures due to the lack of experimental data at elevated pressures. It is believed that this equation, as stated by Kretschmer and Odgers, is applicable for saturated hydrocarbon fuels only.

6.7.3 Residence Time

Residence time in a combustor has a very pronounced effect on combustion efficiency, stability and the formation of various chemical species

[6]. The last effect is quite important for pollutant formation. It is well understood now that the prime factor in forming nitrogen oxides in combustion equipment is the residence time at the maximum temperature in the combustion space [79]. Work done on similarity or scaling factors in combustors by Stewart and Quigg [106] and by Woodward [121], and the analysis of the experimental work of this study indicate that the combustor efficiency is affected by several parameters. The major parameters are the Reynolds' number, total air flow rate, air/fuel ratio, fuel heating value, the amount of swirl imparted to the primary combustion air, the pressure inside the combustor and the reaction time. Through analysis of these parameters, they could be grouped together in a non-dimensional form and by applying a curve fitting technique to the experimental results, the manner in which the combustor efficiency depends on any parameter could be defined. The combustor efficiency could be expressed as a function of the various parameters in the following form:

$$\eta = \eta(\dot{m}_a, \dot{m}_f/\dot{m}_a, Re, \tau, S, HV) \quad (6.6)$$

where, η = combustor efficiency

$\eta()$ = indicates functional relationship

$\dot{m}_f$ = mass flow rate of fuel

$\dot{m}_a$ = mass flow rate of air

Re = Reynolds' number = $\rho \bar{V} d / \mu$ (calculated at combustor inlet)

$$\bar{V} = \dot{m}_a / \rho A_r$$

A_r = reference area of combustor

τ = ratio of reaction time to residence time

S = swirl number

$$= G_\phi / (G_x r_0)$$

G_ϕ = axial flux of angular momentum

G_x = axial thrust

r_0 = outer radius of swirler

HV = the fuel heating value

The swirl number S of an annular swirler with constant vane angle θ in a constant diameter duct is given as follows after Beer and Chigier [9] in terms of the vane angle and swirler dimensions.

$$S = \frac{2}{3} \frac{1 - (r_i/r_o)^3}{1 - (r_i/r_o)^2} \tan \theta \quad (6.7)$$

or S could be modified to be based on the combustor diameter rather than the duct diameter as suggested by Lilley [65] which will take the following form:

$$S = \frac{2}{3} \frac{r_c}{r_o} \frac{1 - (r_i/r_o)^3}{1 - (r_i/r_o)^2} \tan \theta \quad (6.8)$$

where, θ = vane angle
 r_c = radius of combustor
 r_i = inner radius of swirler
 r_o = outer radius of swirler

The ratio of reaction time to residence time τ was related to the operating parameters of the combustor by Woodward [121] as follows:

$$\tau = \frac{\Delta t_2}{\Delta t_1} \quad (6.9)$$

where, Δt_2 = reaction time
 Δt_1 = residence time

The average reaction rate could be expressed as follows:

$$\bar{u} = c \rho^n$$

where, c depends on inlet temperature and mixture condition

ρ = mixture density

n = effective overall reaction order

then Δt_2 could be expressed in the form

$$\Delta t_2 = \frac{\rho}{\bar{u}} = \frac{1}{C \rho^{n-1}} \quad (6.10)$$

The residence time is proportional to the combustor length and the average axial velocity of flow through the combustor,

$$\Delta t_1 = \frac{\ell}{u}$$

where, ℓ = combustor length
 u = average axial velocity

$$\therefore \tau = \frac{C' u}{\ell \rho^{n-1}} \quad (6.11)$$

Substituting for u from the mass flow rate in the combustor and its dimensions and for ρ from the equation of state,

$$\tau = \frac{C'' \dot{m}_a}{P^n D^3} \quad (6.12)$$

where D is the combustor reference diameter.

The expression given for the combustor efficiency could be given in the form

$$\eta = \eta (Re, \dot{m}_a, \dot{m}_f/\dot{m}_a, \dot{m}_a/P^n D^3, S, HV) \quad (6.13)$$

or

$$\eta = \eta (Re, \dot{m}_a, \dot{m}_f/\dot{m}_a, \beta, S, HV) \quad (6.14)$$

where $\beta = \dot{m}_a/P^n D^3$ (6.15)

This parameter β was introduced by Woodward [121]. Experimental tests on several combustors by Woodward [121] showed that Re has a very slight effect on η for $Re > 30,000$ (Re for the present work is in the range of 90,000) and η correlates very well with β . The correlation is given in the form

$$\eta = a_0 + a_1 \beta + a_2 \beta^2 + a_3 \beta^3 \quad (6.16)$$

where a_0, a_1, a_2, a_3 are constants. The approach that will be given here is to take equation (6.16) above and obtain the values of the a 's from the experimental results and include those effects that were not included in deriving equation (6.16) namely the fuel/air ratio, swirl and heating value. Through curve fitting techniques, the functional relationship of these parameters to the combustor efficiency is then determined. The main assumption here is that the functional relationship of η & β , η & F/A , η & S and η & HV are independent of each other.

The results of the calculation of the combustor efficiency based on the experimental values is given in Table 5.1. Due to the controversies over the reaction order ' n ' and also the lack of information available regarding this value for mixtures of CO , H_2 and CO_2 burning in air, it was decided to consider ' n ' to have a value of 2 which is in the range suggested by many investigators. Solving a set of linear equations for the values of a_0, a_1, a_2 , and a_3 from results in Table 5.1, the following values were obtained (it should be mentioned here that the units used for $\dot{m}_a$ is kg/h, for P in kPa and for D in m, which would result in reasonable values for β):

$$\left. \begin{aligned} a_0 &= 0.858 \\ a_1 &= 2.602 \times 10^{-2} \\ a_2 &= -1.978 \times 10^{-2} \\ a_3 &= 4.601 \times 10^{-3} \end{aligned} \right\} \quad (6.17)$$

The effect of fuel/air ratio on the combustor efficiency is expected to be hyperbolic. However, in the range of application of this combustor, the relationship could be approximated to a straight line. Fig. 6.3 shows the plot of η versus F/A ratio, while the other variables are kept constant. The plot tends to agree with the straight line relationship. The effect of F/A ratio could then be given as follows:

$$\eta = 1 - 40(F/A - F/A^*) \quad (6.18)$$

where the * superscript over any variable in this section indicates values at the base run (run #1) in Table 5.1.

Fig. 6.4 shows the variation of the combustor efficiency if all other variables are kept constant except for the fuel heating value. The fuel

heating value was normalized through the heating value of ~~CO~~ to render it non-dimensional. The choice of CO was due to its major mass proportion in the fuel and also because it has only one heating value unlike hydrogen which has two heating values (low and high heating values). Then the normalized heating value could be given as

$$\overline{HV} = HV/HV_{CO} \quad (6.19)$$

Curve fitting of the plotted results dictate the use of exponential function between η and $\overline{HV}$, it is given as follows:

$$\eta = \text{Exp } 0.200(\overline{HV} / \overline{HV}^* - 1) \quad (6.20)$$

A similar procedure was followed for the effect of swirl. As given earlier, a characteristic number was given to render the effect of swirl non-dimensional which was termed the swirl number. Fig. 6.5 shows the combustor efficiency versus swirl number with all the other variables kept constant. The shape of the curve suggested an exponential curve fitting of the form:

$$\eta = \text{Exp } 0.300(S/S^* - 1) \quad (6.21)$$

Combining all the above expressions in one global equation gives the combustor efficiency in the following form

$$\eta = \{ 1 - 40[(F/A) - (F/A)^*] \} (0.858 + 2.602 \times 10^{-2} \beta - 1.978 \times 10^{-2} \beta^2 + 4.601 \times 10^{-3} \beta^3) \\ \times \text{Exp } 0.200(\overline{HV} / \overline{HV}^* - 1) \text{Exp } 0.300(S/S^* - 1) \quad (6.22)$$

or after rearranging

$$\eta = \{ 1 - 40[(F/A) - (F/A)^*] \} (0.858 + 2.602 \times 10^{-2} \beta - 1.978 \times 10^{-2} \beta^2 + 4.601 \times 10^{-3} \beta^3) \\ \times \text{Exp}[0.2(\overline{HV} / \overline{HV}^* - 1) + 0.3(S/S^* - 1)] \quad (6.23)$$

The above equation for η relates the major parameters of the flow and the combustor in such a manner that would allow predicting the combustor efficiency at conditions which were not tested during the course of this study. This equation when applied to conditions that differ from those on which it was based, resulted in an accuracy of prediction of 4% (the last column of Table 5.1b gives the value of the combustor efficiency calculated from experimental measurements and the corresponding value calculated from the combustor efficiency formula, Eq. (6.23), for runs #16-20). This accuracy range is somewhat higher than what is demanded by gas turbine designers but this can be remedied through a wide range of tests with a wider variety of operating conditions to improve the constants included in the present form. Also, with a larger number of runs the assumption of the independence of the various parameters from each other or their effect on the efficiency can be examined.

Fig. 6.1 Schematics of primary zone and flow models thereof
 1=zone of droplet evaporation and/or combustion;
 2=zone with flow in downstream direction;
 3=recirculation zone;
 WSR=well stirred reactor;
 PSR=partially stirred reactor;
 ID=one dimensional(plug)flow
 TDF=turbulent diffusion flame;
 R=recirculation flow of combustion products

Fig. 6.2 Gas turbine van; proposed interconnection of reactors comprising the combustor model

Fig. 6.3 Combustor efficiency versus fuel/air ratio

Fig. 6.4 Combustor efficiency versus heating value of fuel

Fig. 6.5 Combustor efficiency versus swirl number

CHAPTER SEVEN

CONCLUSIONS, SUMMARY OF CONTRIBUTION AND RECOMMENDATIONS

The main conclusions drawn from this study are outlined in this chapter along with a summary of the contribution of this work and recommendations for further work.

7.1 Conclusions

Through measurements of temperature, flow rates, velocity vector and concentration of species, the effect and contribution of operating or design parameters on the combustor performance were determined. The parameters studied were all found to play a role in ensuring the achievement of the function required of the combustor of providing a high temperature gas stream with an acceptable shape factor profile at the exit section and uniform velocity under the constraints of a combustion efficiency close to 100% and pressure drop less than 5%.

Air flow rate should be carefully chosen as both temperature profiles and velocity profiles at the exit section depend on it. Effectiveness of cooling is also influenced by air flow rate and in order to avoid hot spots, the air flow rate should not drop below a certain limit depending on the combustor design and material. In general, the air flow rate should result in ≥ 1.0 kg/s m^2 mass flow rate per unit area in the primary zone. However, this value should not exceed 7.0 kg/s m^2 or the pressure losses will rise above the allowable limit.

The distribution of the total air flow rate among the combustor zones should be made in such a manner that will allow a smaller proportion (~7%), than usually utilized in the combustor industry (27% as compared with the usual 37% [112]), to be admitted at the dilution zone. Higher proportion of air is needed in the primary and secondary zones as the temperature at the combustor exit becomes higher. The consequence is more homogeneous flow at the combustor exit section.

The primary combustion air should be given a moderate swirl by making the inlet vane angle of the swirler in the vicinity of 30-45 degrees. That ensures a stable flame over a wide range of operation.

The heating value of the fuel for combustor application should not be less than 6.62 MJ/m^3 (if some portion, ~4%, of the combustion air is premixed with the fuel). If the heating value is less, stability problems will be encountered and countermeasures must be developed. Increase of air proportion admitted at the secondary zone would give the effect of cross flow which would enhance stability and widen the range of ratios of ignitable mixtures. Also, reduction of the premixed portion of air might help to expand the ignitability of fuels with heating value less than 6.62 MJ/m^3 to close to 3.7 MJ/m^3 .

The amount of swirl imparted to the primary combustion has a very significant effect on the range of mixture ratios that could be ignited in the combustor. The combustor must operate within this range or flame extinction will take place. Reducing the primary air swirl will increase the range of ratios of mixtures that could be ignited in the combustor.

The plots given in various figures presented in this study clearly show the absence of symmetry for the various properties of the flow inside the combustor. The flame (or flow in general) tends to rise toward the upper wall of the combustor tube. This buoyancy, the recirculation eddies created by the primary air swirl and the eddies formed due to the admission of air at different zones, all interact together destroying any symmetry which might otherwise exist. This should be taken into account when attempts are made to model the overall phenomena inside a combustor. The combustor must always be treated as a three-dimensional flow field with a very high degree of turbulence.

The study also reveals the fact that low-medium heating value combustion equipment can be made shorter than conventional combustor units due to the short distance the fuel needs to burn completely. This is of course true, only when sufficient mixing of fuel and air is achieved.

Based on the experimental results and some theoretical analysis, the dependency of the combustor efficiency on operating parameters (fuel/air ratio, pressure, heating value, swirl and air mass flow rate) was established

in the form of a global, empirical equation. This very simple equation includes all the effects that were not considered in any previous analysis such as, the combustor geometry (swirl), fuel composition (as it affects the heating value) and fuel-air ratio. This equation is capable of predicting the combustor efficiency at various conditions other than design or operating conditions. This is of great importance for combustor development in this area of low-medium heating value fuels due to its direct impact on reducing the number of experiments needed.

7.2 Summary of Contribution

The contribution of this work is summarized as follows:

1. A test combustor suitable for studying the combustion characteristics of fuels in the low-medium heating value range (as well as other fuels) has been developed.
2. Using the test combustor facility, detailed measurements of temperature, velocity vector and blowout conditions have been obtained throughout the combustor. These measured values have been used to develop an understanding of the interaction of variables which influence the performance of the combustor.
3. Through variation of the combustor operating parameters during the experiments conducted, the optimum combination of these parameters has been determined (as outlined in Sec. 5.8). The optimum combination consists of: air distribution among the combustor zones that would result in 53% in the secondary zone versus 27% in the dilution zone, air flow rate per unit area not less than 1 kg/s m^2 in the primary zone, air/fuel ratio about 38 (by mass), fuel heating value not less than 6.62 MJ/m^3 (with 4% of the combustion air premixed with the fuel), and the primary combustion air should be given a swirl equivalent to that

obtained from at least a 30 degree vane angle swirler in order to obtain stable combustion. Also the limitation of utilizing low heating value fuel has been demonstrated.

4. A global equation to calculate the combustor efficiency utilizing the operating conditions has been developed. This equation includes such parameters as Reynolds number, air mass flow rate and the ratio of reaction time to residence time which have been used in previous studies as well as parameters such as fuel/air ratio, heating value and swirl which are absent from previous work.

7.3 Recommendations for Further Work

7.3.1 Gas Analysis

A comprehensive gas analysis technique should be applied to samples drawn at various points inside the combustor. The gas analysis facility should be capable of detecting CO and H₂ (CH₄ if present) in quantities close to 0.001%. This includes pollutant species (such as NO_x and CO) which might exist in minute quantities measured in ppm. This will help to better define the kinetics involved in the combustion of such fuel.

Another extension of gas analysis should be in the direction of evaluating the impact on the environment of burning such gases. This could be done either by just measuring for the pollutant species at the exit section of the combustor, or even better, by measuring these species throughout the combustor in order to define their rate of creation or decay.

7.3.2 Gasifier

Since the source of these fuels would mainly come from gasification of coal, it is recommended to replace the simulated gas with the real gas produced from a gasifier. A fluidized bed gasifier already exists at the Energy Conversion Laboratory, but its throughput is too small for a test sized

combustor. A larger gasifier, which produces fuel flow rate in the range stated in this work or higher, is recommended to be built. This gasifier, if built for high pressure operation, will greatly enhance the research of low-medium heating value fuel.

7.3.3 Air Pressure

All the experimental testing done in this study was carried out with the air supply almost at atmospheric pressure. In real combustor application, however, air is usually at high pressures and so is the fuel. A series of runs are recommended to be performed at elevated air pressures to provide information at these conditions.

7.3.4 Fuel Kinetics

Very little data are published about the kinetics of a mixture of CO, H₂, CO₂ burning in air. Some investigation in this area with a wide range of the composition of the fuel mixture is recommended.

7.3.5 Numerical Approach

In this study, a formula was obtained from the experimental results to predict the efficiency over a wide range of operating conditions. Due to the statistical error inherent in this formula, it is recommended that a larger series of tests be carried out and the combustor efficiency be compared to the predictions from the formula to establish or extend its validity.

The numerical solution of the three-dimensional flow regime of this combustor is an ultimate goal to look for. It is expected that through such a study, along with enough knowledge developed in the area of kinetics and mixing, combustors can be built and developed for fuels in the low-medium heating value range at much lower cost than through purely experimental approaches.

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APPENDIX A

DESCRIPTION, CALIBRATION AND USE OF THE FIVE-HOLE PROBE

This appendix describes the 5-hole probe and how it is used with the 2-hole probe to obtain the velocity vector at desired positions in the combustion space. Also included in this appendix is a brief description of the calibration procedure of the 5-hole probe.

A.1 Description of the Five-Hole Probe

Fig. A.1 shows a schematic illustration of the shape of the 5-hole probe tip (measuring section). The probe is manufactured by United Sensor & Control Corporation from stainless steel and it is water cooled to withstand the high temperature in the combustion space. The five holes are located on a prism shaped section in such a manner as to allow measuring the yaw angle and pitch angle as well as static and total pressure. The centrally located pressure hole measures the total pressure P_1 while the two lateral pressure holes in the same plane measure P_2 and P_3 through which the yaw angle could be determined. The probe operates in the nulling mode by rotating the probe about its axis until the pressure difference $P_2 - P_3$ is zero. The yaw angle is, then, measured by a protractor whose centre is at the probe axis. The pitch angle is obtained through the pressure differential $P_4 - P_5$ with the holes for P_4 and P_5 located above and below the central total pressure hole P_1 . The layout of the 5-hole probe is given in Fig. A.2 together with the main dimensions and the location of the five holes. The probe has the advantage of having its measuring tip along its stem, which simplifies its insertion into the combustion space. The probe is fixed at any location by means of a manual traverse unit (also obtained from United Sensor & Control Corporation) which allows radial positioning and rotation of the probe about its axis. Fig. A.3 shows the layout of the manual traverse unit. The connections of the manometers to the five-hole probe while it is inserted in the combustion space is illustrated in Fig. A.4. The pressure differences P_1

- P_2 , $P_2 - P_3$ and $P_4 - P_5$ which are indicated in Fig A.4 are measured by inclined manometers except during very small values (< 0.005 inches of water) of $P_1 - P_2$ or $P_4 - P_5$ when a Hook's gauge is used. The pressure difference $P_1 - P_{atm}$ is measured by a U-tube manometer.

A.2 The Two-Hole Probe Traverse

The 5-hole probe cannot distinguish if the flow is travelling upstream or in the downstream direction. Immersion of the five-hole probe in a stream opposite to the direction of the velocity vector would result in a very serious error which cannot be detected during measurements. For this reason, a full traverse across every section along the combustor was made by the two-hole probe to identify the recirculation zone or the boundaries of the reversed flow. This traverse was then used as a guide to point the five-hole probe in the correct direction.

A.3 Calibration of the Five-Hole Probe

The five-hole probe used in the experimental work was calibrated at United Sensor & Control Corporation. This calibration was checked in a setup made of a nozzle fitted on a box (to stabilize the flow) with the help of a Pitot tube before using its data to obtain the pitch angle, the total and static pressures. Fig. A.5 represents the calibration results (in the nulling conditions, i.e. $P_2 = P_3$) for the 5-hole probe provided by United Sensor & Control Corporation. The calibration is given as pitch angle pressure coefficient and velocity pressure coefficient versus pitch angle in one plot while the other plot gives the total pressure coefficient versus pitch angle.

A.4 Procedure to Calculate the Three Velocity Components

This section gives an outline of the procedure followed to calculate the three velocity components from the measurements taken by the manometers and the yaw angle measured by the protractor. The calculations begin with

correction of P_2 , P_4 and P_5 for the error caused to the static pressure as a result of the reduction of flow area due to probe insertion in the combustion space. This correction is given [124] as follows.

$$\Delta P_s / (P_t - P_s) = 1.2 (a/A) \quad (A.1)$$

where ΔP_s = static pressure correction
 P_t = total pressure (P_1 could be taken to start with)
 P_s = static pressure to be corrected
 a = immersed area of the probe
 A = cross-sectional area of the passage

From the ratio $(P_4 - P_5)/(P_1 - P_2)$ and using the calibration curve of Fig. A.5a, the pitch angle could be determined. From the pitch angle, the quantity $(P_t - P_s)/(P_1 - P_2)$ is obtained from Fig. A.5a. The total pressure P_t is corrected for using Fig. A.5b to obtain $(P_1 - P_t)$ corresponding to the pitch angle. The static pressure calculated in Eq. (A.1) is to be corrected according to the value of P_t (instead of P_1), and the procedure is repeated until the difference through this iteration could be ignored. In the present case, $(P_1 - P_t)/(P_t - P_s)$ is zero for a wide range of the pitch angle, from -33° to 40° , hence the correction or iteration procedure was not needed in this pitch angle range.

The dynamic pressure difference calculated from $(P_t - P_s)/(P_1 - P_2)$ is used to compute the magnitude of the velocity vector from the following relationship:

$$U = \sqrt{2(P_t - P_s)/\rho} \quad (A.2)$$

where ρ = the gas density calculated from the gas composition and temperature where the velocity measurement is made.

The three velocity components are calculated as follows (see Fig A.6 for the angles and components of the velocity vector).

$$V_z = U \cos\theta \cos\phi$$

$$V_r = U \cos\theta$$

(A.3)

$$V_\theta = U \cos\theta \sin\phi$$

These velocity components are plotted along the combustor sections for the runs that were performed during this study.

Fig. A.1 Detail of five-hole probe tip

Fig. A.2 Layout of the five-hole probe, main dimensions, pressures and angles

Fig. A.3 Manual traverse unit ,used to mount the various probes in this study

Fig. A.4 Connections of the manometers while the five-hole probe is placed in the combustion space

Fig. A.5a Pitch angle pressure coefficient and velocity pressure coefficient versus pitch angle

Fig. A.5b Total pressure coefficient versus pitch angle

Fig. A.6 Components and angles of velocity vector

APPENDIX B

PROCEDURES AND TECHNIQUES

This appendix outlines the procedure followed to measure the various quantities needed for the experimental work. It will also outline the techniques used to carry on the experiments or to change the parameter that needed to vary from one test to another.

It should be mentioned here that the average time needed to make the necessary measurements for the temperature and velocity was approximately 25 hours for one run. The set of gas bottles (3 of each of CO , H_2 and CO_2) could last only about 6 hours. This meant that the run had to be stopped at least four times to change the gas bottles. In order to establish a consistent procedure for continuation after a set of gas bottles was consumed, it was decided to move back to one previous section rather than the section that was under consideration and proceed with the type of measurements that were being performed. This, established some kind of repeatability due to the fact that the difference upon redoing any section was less than .3% for velocity measurements and less than 5 degree C for temperature measurements.

B.1 Temperature

The aspirated thermocouple probe is mounted in a manual traverse unit obtained from United Sensors and Control Corp. (Fig A.3). The unit is fixed by a clamp at the desired section with the probe tip at the far side of the combustor liner. The scale is fixed in position by a thumb screw. After each reading, the probe is moved gradually outward a distance of 5 mm using the hand adjusting nut at the top of the scale. At every measuring position, the aspiration pump is turned on and the flow rate through the pump is adjusted to match the average axial velocity of flow in the combustor (as calculated from the mass flow rate and an approximate average temperature at the section where the measurements are being taken). The temperature is read from a digital readout calibrated to the thermocouple wire used. The reading is taken when

oscillations of the reading came to $\pm 2^{\circ}\text{C}$ which required about 30 minutes for the first reading if the test is just starting from cold conditions. The time needed for consecutive reading is much less and usually in the range of about 3-5 minutes. After a traverse along the section, the probe is moved to the second hole at a circumferential position 120° from the previous position at the same axial section. When the traverse is done for the three holes at the section, the procedure is repeated at the next section until temperature is measured at all sections along the combustor.

B.2 Velocity Vector

The 5-hole probe is mounted in the manual traverse unit (Fig A.3) and the unit is clamped such that the traverse unit protractor reading is zero when the axis of the probe hole number 1 is parallel to the combustor axis and facing the velocity vector (as determined by the 2-hole probe). Alignment of the probe is done with the help of a set square and a fixed platform parallel to the combustor axis. Again, the probe is set inside the combustor with its tip at the opposite liner wall. The probe is slowly rotated about its axis until $P_2 - P_3$ is zero (with the help of the manometer) and stays so for a while. Then the other readings are recorded namely $P_1 - P_{\text{atm}}$, $P_1 - P_2$, $P_4 - P_5$ (as shown in Fig A.4) as well as the yaw angle (Fig A.2) which is obtained with the help of the protractor surrounding the probe and the rotation vernier. The probe is then moved outward a distance of 5 mm with the aid of the traverse scale and the linear vernier (Fig A.3). The probe may be kept in the same position or rotated 180° depending on the flow direction at the measuring position as determined from the 2-hole probe traverse. When steady conditions are achieved, the readings mentioned above are recorded at the new position. This procedure is repeated until a full radial traverse is completed. The probe is then moved to the hole which is located 120° in the circumferential direction from the one which was just finished. After completing measurements at one section, the whole procedure is repeated until measurements have been made at all sections along the combustor.

The procedure for calculating the velocity vector from the readings of the 5-hole probe are illustrated in Appendix A together with the use of the 2-hole probe and calibration steps of the 5-hole probe.

B.3 Gas Analysis

Simultaneously, while measuring the gas temperature at any position and after recording the temperature reading, the gas is allowed to flow through the sampling tube with the aspiration pump still on, for a period sufficient to flush the previous gas from the sampling tube and fill it with the gas at the measuring section. One cc of gas from the gas sampling tube is injected (via a syringe) into the gas chromatograph and the analysis is obtained. The plot is marked for the sample position and run number. After completing measurements and analysis for one section, the results are analyzed and repeated if considered unsatisfactory or if verification is desired. This is mainly decided according to the consistency of the resulting analysis. Calculations of the proportions of the various gases in the sample gases were done by a comparison of the chromatograms obtained for the particular sample and from the standard gas sample. Hydrogen, if present, will be expected only in the primary zone of the combustor. Therefore analysis for the first four sections of the combustor was carried out on two chromatographs. One chromatograph was for hydrogen analysis using nitrogen as a carrier gas; while the other is for CO , CO_2 , N_2 , O_2 analysis using helium as carrier gas.

It should be pointed out here that the plots given in Chapter 5 for the radial distribution of the temperature and the three velocity components at the various sections of the combustor were averaged as the mean value of the three readings (or in the case of the velocity from the calculated velocity components at the three positions of the 5-hole probe across the section diameter). The five-hole probe, however, has a length at its tip (Fig. A.1) which prevented any measurement closer than 19 mm from the opposite liner wall. This means that the averaging of the velocity components was done from one reading for the upper four measuring positions at any section while it was done from two readings for the lower four measuring sections.

B.4 Swirl

The amount of swirl imparted to the primary combustion air was varied by installing various swirlers. The swirlers have a vane angle of 0° , 30° , 45° , 60° (Fig 3.4). The degree of swirl is measured by a dimensionless parameter called the swirl number, which is defined by [15] as the ratio of the angular momentum to the axial thrust times the nozzle radius. Throughout this work, the expression "various swirl" or "various swirlers" will be used to indicate the use of a swirler (or swirlers) of different vane angle (or angles).

B.5 Stability Loops of the Combustor

The range of mixture ratios that could be ignited in the combustor were obtained by adjusting the fuel flow rate to about 5% of the normal value. The test starts with fuel flowing while the air supply is completely shut off. The ignition source is turned on then the air is slowly allowed to pass through the combustor. A thermocouple is placed inside the combustor to detect the start of combustion when initially not burning or extinction of the flame in the opposite case. When sufficient air is mixing with the fuel flowing through the combustor, the rich mixture ratio would be reached (which could be identified by a sudden rise in the temperature indicated by the thermocouple). The air flow rate at this instant is recorded. The ignition source is then turned off and the air flow is allowed to continue to increase until flame extinction occurs. This would indicate that the lean ignitable ratio is reached. Upon obtaining both ignitable ratios at a certain fuel flow rate, the whole system is turned off. The flow rate of fuel is increased by another 5% and the above procedure is repeated until the full fuel flow rate stability loop is obtained.

B.6 The Pattern Factor

The pattern factor (or as sometimes called temperature factor and traverse number) is defined as the peak gas temperature minus mean gas temperature divided by mean temperature rise [40]. The pattern factor is calculated from the above definition at the exit section for various runs.

APPENDIX C

PROCEDURE USED IN DATA REDUCTION

This appendix outlines the procedure followed in the various calculations of this work. These calculations include using experimental results to evaluate the mass flow rate at the various sections of the combustor, the axial flux of angular momentum and the application of the continuity principle at some selected sections of the combustor in order to justify (or otherwise) the radial velocity profiles obtained at these sections. The definition of the combustor efficiency as used in this research is also given with the steps that were followed in its calculation. This appendix includes also the procedure of estimating the average temperature at the first section of the combustor in order to compare it with the average value from the measurements.

C.1 Mass Flow Rate

C.1.1 Mass Flow Rate From Axial Velocity and Temperature Measurements

The mass flow rate at any section of the combustor is calculated by integrating the product of the density and the axial velocity over the area of the section, which could be expressed as follows:

$$\dot{m}_s = \int_A \rho V_z dA$$

where $\dot{m}_s$ = mass flow rate at the section being examined

$\int_A$ = integration over the area of the section A

ρ = density

V_z = axial velocity

The integration was performed numerically using the method of trapezoids by averaging the values of V_z between two adjacent points. The area increment is calculated from the mean radius between the relevant two measuring points

and the distance increment between these two measuring points. The density of the gas was calculated at the mean temperature following the calculation of the gas constant from the expected composition (based on fuel and air flow rates) at each section and assuming atmospheric pressure throughout the combustion space. A complete listing of the computer program used to perform these calculations is given at the end of this appendix. Table C.1 contains the values obtained from these calculations at the various sections for the ten runs.

C.1.2 Mass Flow Rate From Fuel and Air Flow Rates and the Open Hole Area

The mass flow rate at a given section depends on the mass flow rate of fuel and the amount of air admitted at this section together with the open hole area to admit the secondary and dilution air or the air for film cooling. This in addition to the air flow rate that exists upstream of the section under consideration. From the dimensions of the open holes, the total area at one set of air admission holes (one section normal to the combustor axis that contains 8 holes of 1.27 cm diameter) is 10.13 cm^2 and the open area for film cooling in one ring is 1.9 cm^2 . Therefore, the mass flow rate at the various sections could be estimated as follows (according to the geometry of the combustor):

$$\dot{m}_1 = \dot{m}_f + \dot{m}_{pre} + \dot{m}_{pri}$$

$$\dot{m}_2 = \dot{m}_1 + \dot{m}_{sec} \left(\frac{A_c}{2A_c + A_f} \right)$$

$$\dot{m}_3 = \dot{m}_2 + \dot{m}_{sec} \left(\frac{A_c + A_f}{2A_c + A_f} \right)$$

$$\dot{m}_5 = \dot{m}_4 = \dot{m}_3 \text{ (since there is no open area in this region)}$$

$$\dot{m}_6 = \dot{m}_5 + \dot{m}_{dil} \left(\frac{A_c + A_f}{2A_c + 3A_f} \right)$$

$$\dot{m}_7 = \dot{m}_6 + \dot{m}_{dil} \left(\frac{A_c + A_f}{2A_c + 3A_f} \right)$$

$$\dot{m}_8 = \dot{m}_7 + \dot{m}_{dil} \left(\frac{A_c + A_f}{2A_c + 3A_f} \right)$$

$$\dot{m}_{10} = \dot{m}_9 = \dot{m}_8$$

where $\dot{m}$ = mass flow rate

A_c = liner open hole area at the section of interest
for the admission of secondary or dilution air

A_f = open hole area for film cooling

subscripts 1-10 refer to the measuring section number

f refers to fuel

pre refers to premixed air

pri refers to primary air

sec refers to secondary air

dil refers to dilution air

Table C.2 contains the values of mass flow rate of the various sections for all the runs calculated following this procedure. Table C.3 gives the values for the percentage difference between the calculated mass flow rate from the measurements of axial velocity and temperature and that from the open hole area. The percentage difference at any section was calculated from the following expression:

$$\Delta \dot{m} \% = ((\dot{m}_f + \dot{m}_h) / \dot{m}_h) \times 100$$

where $\Delta \dot{m} \%$ = percentage difference in mass flow rate

$\dot{m}_f$ = mass flow rate from integration of the
area of the section

$\dot{m}_h$ = mass flow rate calculated from the input
mass flow rates and the open hole area

The last column and last row of Table C.3 contains the average of the percentage difference for any run through all sections and any section from all the runs respectively.

C.2 Application of the Continuity Principle to a Control Volume at the Combustor Axis

The sections that were found to show a discrepancy in the radial velocity profile are #9 in run #1 and sections 1, 6, 7, and 8 in run #2 and section #9 in run #5. Application of the continuity principle was performed on a control volume adjacent to the axis of the combustor. The control volume was selected to be a half-cylinder which extended in the axial direction between the section under consideration and the next section (which determined the length of the half-cylinder) and of a diameter equal to the length. The reason for this choice was to allow all three velocity components to be considered and to permit the variation in the radial direction due to the lack of symmetry of flow in the combustor. Fig. C.1 shows a schematic sketch of the control volume. Axial flow resulting from the axial velocity component will only take place through the two semicircular ends of the control volume. Radial flow is only possible through the curved surface of the cylindrical control volume (ABFDCEA). Circumferential flow is possible only through the flat surface of the control volume (ABCD). The radial velocity component was obtained from the measurements as given in the various plots at points A, B, C and D. They were averaged assuming linear variation and the resulting value was considered to flow through the curved surface of the cylinder to be used for the calculations of the mass flow rate.

The axial mass flow rate was calculated over the area following the same procedure as described in section C.1 with the exception that the cross section is half a circle and extends only to the diameter of the control volume. The circumferential velocity was averaged over AO_1 , O_1C , BO_2 , and O_2D . The average over CO_1 & O_2D was considered for flow through the area CDO_2O_1 and the average over AO_1 & BO_2 was considered for flow through ABO_2O_1 . The calculations performed for section #9 of run #1 are given below as an example of the calculations:

the length between sections #8 and 9 $\Delta Z = 0.025 \text{ m}$

the diameter of the control volume = 0.025 m

The radial velocity component at corner A of the C.V. = 1.9 m/s

The radial velocity component at corner B of the C.V. = 1.1 m/s

The radial velocity component at corner C of the C.V. = -1.0 m/s

The radial velocity component at corner D of the C.V. = 1.4 m/s

$$v_{r,average} = \frac{1}{4}(v_{r,A} + v_{r,B} + v_{r,C} + v_{r,D}) = 0.85 \text{ m/s}$$

mass flow rate due to axial velocity entering C.V. through section #8

$$= \int_A \rho V_z dA = 5.3 \text{ kg/h}$$

mass flow rate due to axial velocity leaving C.V. through section #9

$$= \int_A \rho V_z dA = 4.8 \text{ kg/h}$$

mass flow rate leaving C.V. due to radial velocity component

$$= v_{r,average} \pi d \Delta z \rho \times 3600/2$$

$$= 1.6 \text{ kg/h}$$

$$v_{\theta,AO_1} = -0.2 \text{ m/s}$$

$$v_{\theta,O_1C} = 0.0 \text{ m/s}$$

$$v_{\theta,BO_2} = -0.1 \text{ m/s}$$

$$v_{\theta,O_2D} = -0.15 \text{ m/s}$$

average circumferential velocity on surface $ABO_2O_1 = -0.15 \text{ m/s}$

average circumferential velocity on surface $O_1O_2DC = -0.075 \text{ m/s}$

According to the placement of the swirler vanes v_{θ} is positive if it will cause the tangent to the C.V. base at C to rotate clockwise. A positive circumferential component on the upper half of the C.V. (area O_1O_2DC) means mass flowing into the C.V. while a positive circumferential component on the lower half of the C.V. (area ABO_2O_1) means mass flowing out of the C.V. The negative circumferential component in either case means the reverse will occur.

mass flowing into the C.V. due to circumferential velocity

$$\dot{m} = v_{\theta,ABO_2O_1} \pi d \Delta z \rho \times 3600/2 = -0.045 \text{ kg/h}$$

mass flowing out of the C.V. due to circumferential velocity

$$= V_{\theta,0_1} \rho_2 DC^x d \times \Delta z \times \rho \times 3600/2 = -0.09 \text{ kg/h}$$

the net mass flowing into the C.V. = $5.3 - 0.045 = 5.255 \text{ kg/h}$

the net mass flowing out of the C.V. = $4.8 + 1.6 - 0.09 = 6.31 \text{ kg/h}$

The percentage difference $\Delta \% = (\text{mass}_{\text{in}} - \text{mass}_{\text{out}}) / \text{mass}_{\text{in}} \times 100 = -20\%$

The following are the values of the mass in and out of the C.V. and the percentage difference as calculated for the sections demanded verification:

Run #	Sec #	Mass _{in} kg/h	Mass _{out} kg/h	$\Delta \%$
1	9	5.255	6.31	-20.0
2	1	5.03	4.78	5
2	6	23.8	19.8	17
2	7	8.0	5.7	28
2	8	6.2	4.4	29
5	9	3.8	5.23	-38

C.3 Calculations of the Axial Flux of Angular Momentum

C.3.1 Axial Flux of Angular Momentum at the Various Sections From Integrating Experimental Measurements Over Area of Section

The axial flux of, angular momentum (a quantity which is conserved in swirling flows) may be calculated at any section of the combustor from the relationship

$$G_{\phi} = \int_A \rho V_z V_{\theta} r dA$$

The procedure that was followed to calculate G_{ϕ} is similar to that for calculating the mass flow rate at any section as described in section C.1. The value for V_{θ} was averaged between two adjacent readings at the same

section similar to averaging V_z as described in section C.1. Table C.4 summarizes the results obtained for G_ϕ through the various sections for all the ten runs considered in the experimental study.

C.3.2 Axial Flux of Angular Momentum at the Inlet to the Combustor Due to the Presence of a Swirler

The axial flux of angular momentum of the flow at the combustor inlet was calculated from an expression similar to the one above with the area being the open area of the swirler. This was expressed by Beer and Chigier [9] in the following form:

$$G_\phi = \int_{r_i}^{r_o} \rho v_z v_\theta 2\pi r^2 dr$$

where r_i = inner radius of swirler

r_o = outer radius of swirler

If the axial velocity v_z is assumed to be uniform over the range $r_i - r_o$, the circumferential velocity v_θ will also be uniform as $v_\theta = v_z \tan \theta$ for straight vanes of angle θ to the combustor axis.

$$\begin{aligned} \text{Therefore } G_\phi &= \int_{r_i}^{r_o} \rho v_z^2 2\pi r^2 dr \tan \theta \\ &= 2\pi \rho v_z^2 \tan \theta \int_{r_i}^{r_o} r^2 dr \\ &= 2/3\pi \rho v_z^2 \tan \theta (r_o^3 - r_i^3) \end{aligned}$$

Substituting for v_z from the mass flow rate through the swirler and the swirler area,

$$G_\phi = \frac{1}{6\pi \rho} (\dot{m}_{pri})^2 \frac{r_o^3 - r_i^3}{(r_o^2 - r_i^2)^2} \tan \theta$$

where $\dot{m}_{pri}$ = mass flow rate through swirler

Then the values of G_ϕ for the ten runs were calculated from the above equation according to the swirler vane angle and the mass flow rate in the primary zone ($r_o = 0.025$ m, $r_i = 0.019$ m). The density was calculated

assuming that the flow achieved a temperature of 300°C at the swirler exit section. Table C.5 presents the values of the axial flux of angular momentum calculated for the conditions of the ten runs. Table C.6 gives the difference in the axial flux of angular momentum from Tables C.4 and C.5.

C.4 Combustor Efficiency

The combustor efficiency was introduced in this work as a means of evaluating the performance of the combustor other than the conventionally used combustion efficiency. The combustion efficiency is defined as the heat released in combustion divided by the energy available in fuel. This gives an indication of the completion of the combustion from the chemical point of view. On the other hand the combustor efficiency, as used in this study, is defined as the energy content in the outflowing stream divided by the energy content in the entering stream. This will give an evaluation of how efficient the combustor is in delivering at its exit section the energy (in the form of sensible energy) that it received at its inlet (or inlets, much of which was chemical potential energy). In an equation form, the combustor efficiency is then defined as follows for a steady flow system:

$$\text{combustor efficiency} = \frac{|\Sigma h_{\text{out}}|}{|\Sigma h_{\text{in}}|}$$

where Σh_{out} = total enthalpy content in the gas flowing at the combustor exit

$$= \sum_i n_i \hat{h}_i$$

i = particular specie at the exit section

n_i = number of moles of specie i

$\hat{h}_i$ = standard enthalpy of specie i at the temperature of the gases flowing at the exit section

similarly Σh_{in} = total enthalpy content of all constituents flowing into the combustor

$$= \sum_i n_i \hat{h}_i$$

i = a particular constituent flowing into
the combustor

n_i = number of moles of constituent i

$\hat{h}_i$ = enthalpy per mole of i determined at
the inlet temperature

The number of moles at the combustor inlet were calculated from the input mass flow rates and the composition of the fuel. The inlet temperature was the average room temperature in the lab where the combustor was operating which was approximately 27°C . The out-flowing number of moles were calculated from the assumption of complete combustion and based on the mass flow rate of both the air and the fuel. The temperature of the outflowing gas was considered to be the mean value of temperature measurements at the exit section. The molal enthalpies were obtained from gas tables given by Reynolds [C.1] assuming linear variation with temperature for interpolation purposes. The steps followed to calculate the combustor efficiency for run #1 are given below as an example of procedure of calculations.

At inlet

$$\dot{m}_{\text{air}} = 76.08 \text{ kg/h}$$

$$\dot{m}_f = 1.995 \text{ kg/h}$$

fuel composition 45.6% CO , 9.2% CO_2 & 45.2% H_2 (by volume)

Due to the use of imperial units in the gas tables, the calculations were performed in these units

$$n_{\text{CO}} = 0.113 \text{ lb mole/h}$$

$$n_{\text{H}_2} = 0.22 \text{ lb mole/h}$$

$$n_{\text{CO}_2} = 0.023 \text{ lb mole/h}$$

$$n_{\text{air}} = 5.724 \text{ lb mole/h}$$

At inlet temperature of 300°K or 540°R , the enthalpies were obtained from the gas tables as follows:

$$\begin{aligned}
 \hat{h}_{CO} &= -47491.9 \text{ Btu/lb mole} \\
 \hat{h} &= 20.7 \text{ Btu/lb mole} \\
 \hat{h}_{H_2} &= -169150.4 \text{ Btu/lb mole} \\
 \hat{h}_{CO_2} &= 13.1 \text{ Btu/lb mole} \\
 \hat{h}_{air} &= -9191.6 \text{ Btu/lb mole} \\
 \therefore \Sigma \hat{h}_{in} &= -9191.6 \text{ Btu/lb mole}
 \end{aligned}$$

At the exit section assuming complete combustion of the fuel the various molal contents of all gases are given as follows:

$$\begin{aligned}
 n_{CO_2} &= 0.136 \text{ lb mole/h} \\
 n_{H_2O} &= 0.112 \text{ lb mole/h} \\
 n_{O_2} &= 4.594 \text{ lb mole/h} \\
 n_{N_2} &= 1.11 \text{ lb mole/h}
 \end{aligned}$$

The mean temperature at the exit section from the measurements is 365°C or 1148.9°R , the enthalpies at 1148°R were obtained from the gas tables as follows:

$$\begin{aligned}
 \hat{h}_{CO_2} &= -162846.9 \text{ Btu/lb mole} \\
 \hat{h}_{H_2O} &= -98864.7 \text{ Btu/lb mole} \\
 \hat{h}_{O_2} &= 4504.5 \text{ Btu/lb mole} \\
 \hat{h}_{N_2} &= 4320.0 \text{ Btu/lb mole} \\
 \therefore \Sigma \hat{h}_{out} &= -8373.9 \text{ Btu/lb mole} \\
 \text{the combustor efficiency } \eta &= |-8378.9| / |-9191.6| = 91.1\%
 \end{aligned}$$

Table 5.1 gives the values of η for the ten runs that were reported in this study as well as the other runs that were used to develop the combustor efficiency formula.

C.5 Calculation of the Average Temperature at the First Section of the Combustor

The reaction between the fuel and air at this section could be represented as follows (after assuming complete combustion of hydrogen):

where m_1, m_2, m_3, m_4, m_5 represent the mass flow rates of CO, CO₂, H₂, O₂ and N₂ respectively in the reactants and A, B, C, D and E represent the mass flow rate of CO₂, CO, H₂O, O₂ and N₂ of products. M represents the molecular weight of the gas corresponding to its subscript. The above equation could be written after simplification as follows:

From balance of C

$$\alpha_1 + \alpha_2 = \beta_1 + \beta_2 \quad (\text{C.1})$$

From balance of H

$$\alpha_3 = \beta_3 \quad (\text{C.2})$$

From balance of O

$$\frac{1}{2}\alpha_1 + \alpha_2 + \alpha_4 = \beta_1 + \frac{1}{2}\beta_2 + \frac{1}{2}\beta_3 + \beta_4 \quad (\text{C.3})$$

From balance of N

$$\alpha_5 = \beta_5 \quad (\text{C.4})$$

If the concentration of CO in the first section is known from gas analysis at this section the following equation can be added to Eqs. C.1 to C.4,

$$\delta = \beta_2 / (\beta_1 + \beta_2 + \beta_4 + \beta_5) \quad (\text{C.5})$$

where δ = dry analysis concentration of CO in the first section

Solving the above system of linear equations in β_1 to β_5 , their values can be given as follows:

$$\beta_1 = \alpha_2 (1 - \delta) - \delta \beta_4 - \delta \beta_5$$

$$\beta_2 = \delta (\beta_1 + \beta_4 + \beta_5) / (1 - \delta)$$

$$\beta_3 = \alpha_3$$

$$\beta_4 = \frac{\alpha_4 + \delta(1 - \delta)\beta_5}{(2 - \delta)(1 - \delta)}$$

$$\beta_5 = \alpha_5$$

where $\mu_1 = \frac{1}{2} \alpha_1 + \alpha_2 + \alpha_4 - \frac{1}{2} \alpha_3$

$$\mu_2 = \alpha_1 + \alpha_2$$

$$\rho = 2 \mu_1 (1 - \delta) - \mu_2 (1 - \delta)(2 - \delta)$$

The average temperature at the first section was calculated from applying the principle of energy conservation on the products and reactants of the chemical equation given earlier as follows:

$$\Sigma h_{in} = \Sigma h_{1st, sec} + q_r \quad (C.6)$$

where Σh_{in} = total enthalpy at the inlet to the combustor

$\Sigma h_{1st section}$ = total enthalpy at the first section
calculated as follows

$$\Sigma h_{1st section} = \beta_1 \hat{h}_{CO_2} + \beta_2 \hat{h}_{CO} + \beta_3 \hat{h}_{H_2O} + \beta_4 \hat{h}_{O_2} + \beta_5 \hat{h}_{N_2}$$

q_r = heat lost by radiation from the combustor
surface up to the first section

The total heat input was calculated similar to the approach used in calculating the efficiency, $\Sigma h_{1st section}$ was calculated by trial and error assuming a temperature and obtaining the corresponding enthalpies from the gas tables after solving for β_1 to β_5 from mass flow rate and gas analysis at the first section. q_r was calculated from the surface temperature of the combustor at the first section as estimated by temperature crayons (Tempilstiks) manufactured by Omega Engineering, knowing the surface area. The following is the procedure of calculating the average temperature at the first section of run #1 given as example of calculations.

At inlet

$$\begin{aligned} \alpha_1 = n_{CO} &= 0.113 \text{ lb mole/h} \\ \alpha_2 = n_{CO_2} &= 0.023 \text{ lb mole/h} \\ \alpha_3 = n_{H_2} &= 0.22 \text{ lb mole/h} \\ (\alpha_4 + \alpha_5) = n_{air} &= 1.14 \text{ lb mole/h} \end{aligned}$$

The enthalpies at 540°R are

$$\begin{aligned}\hat{h}_{CO} &= -47491.9 \text{ Btu/lb mole} \\ \hat{h}_{CO_2} &= -169150.4 \text{ Btu/lb mole} \\ \hat{h}_{H_2O} &= 20.7 \text{ Btu/lb mole} \\ \hat{h}_{air} &= 13.1 \text{ Btu/lb mole} \\ \Sigma \hat{h}_{in} &= -9237.6 \text{ Btu/h}\end{aligned}$$

Radiation heat loss

Radiation heat loss from the combustor surface were calculated from the relation

$$q_r = \sigma \epsilon A_s (T_s^4 - T_a^4)$$

where σ = Stefan-Boltzmann constant

$$= 0.56697 \times 10^{-8} \text{ W/m}^2 \text{K}^4$$

ϵ = emissivity

A_s = surface area

T_s = combustor surface temperature °K

T_a = ambient temperature = 300°K

The emissivity = 0.85 (for stainless steel),

A_s = area of combustor surface up to 1st section & flange area

$$= \pi DL + \pi (D^2 - d^2)$$

where D = combustor diameter = 0.127 m

L = combustor length up to first section

$$= 0.0424 \text{ m}$$

d = inner diameter of flange = 0.05 m

$$A_s = 0.0276 \text{ m}^2$$

T_s = 400°C (as estimated with temperature crayons)

$$q_r = 9.4 \times 10^4 \text{ J/h}$$

$$= 89.5 \text{ Btu/h}$$

the total energy content at the first section should be

$$\Sigma \hat{h}_{1st \text{ sec}} = \Sigma \hat{h}_{in} + q_r \text{ (taking into consideration the sign convention for heat leaving a system)}$$

$$= -9148.1 \text{ Btu/h}$$

at the first section

$$\beta_1 = 0.136 \text{ lb mole/h}$$

$$\beta_2 = 0 \text{ lb mole/h}$$

$$\beta_3 = 0.112 \text{ lb mole/h}$$

$$\beta_4 = 0.1636 \text{ lb mole/h}$$

$$\beta_5 = 1.039 \text{ lb mole/h}$$

at $t = 2600^\circ\text{R}$ $\Sigma h_{1\text{st sec}} = -10,045.5 \text{ Btu/h}$ (after obtaining the enthalpies from gas tables)

at $t = 2700^\circ\text{R}$ $\Sigma h_{1\text{st sec}} = -8725.1 \text{ Btu/h}$

or the average temperature should be approximately 2670°R (by interpolation) or 1210°C

The measured average temperature at the first section = 1219°C . Table C.7 gives the values of the average temperature as calculated from the temperature measurements at the first section compared to the average value calculated from energy balance using Eq. C.6.

C.1 Reynolds, W.C., Thermodynamics, McGraw Hill Book Company, 1965.

Table C.1 Mass Flow Rate at the various sections for all the runs as obtained from integrating axial velocity-density product over the area of the section using velocities obtained from the five-hole probe and densities from measured temperatures and assumed constant pressure.

Mass Flow Rate kg/hr										
Sec. #	1	2	3	4	5	6	7	8	9	10
Run #										
1	15.5	32.4	45.4	56.3	47.1	62.8	88.6	77.7	84.4	80.1
2	16.7	24.1	33.2	40.7	45.1	56.8	70.9	73.4	86.1	79.8
3	18.0	31.1	57.6	70.4	68.1	68.6	85.0	81.8	76.1	77.9
4	16.1	27.4	48.5	51.9	49.0	64.0	79.0	81.8	90.1	76.5
5	11.2	19.7	26.7	45.6	28.6	46.7	50.7	50.4	59.8	51.1
6	19.5	31.9	46.4	50.0	50.4	64.4	81.2	79.1	81.0	79.3
7	15.3	25.3	51.0	44.0	48.9	67.9	80.6	82.5	79.9	78.9
8	14.9	23.0	37.6	46.8	45.6	63.6	78.0	91.9	90.4	76.2
9	12.9	23.3	39.9	49.2	41.1	62.7	78.4	88.5	85.8	75.9
10	11.5	20.7	41.7	45.2	48.0	62.5	82.7	92.5	86.6	78.2

Table C.2 Mass Flow Rate at the various sections of the combustor
for all the runs as calculated based on the input mass
flow rates and the open hole area.

Mass Flow Rate kg/hr										
Sec. #	1	2	3	4	5	6	7	8	9	10
Run #										
1	17.2	31.1	47.6	47.6	47.6	61.7	75.8	78.1	78.1	78.1
2	17.2	26.5	37.5	37.5	37.5	56.3	75.1	78.1	78.1	78.1
3	17.2	35.7	57.8	57.8	57.8	67.2	76.6	78.1	78.1	78.1
4	17.6	31.5	48.0	48.0	48.0	62.1	76.2	78.5	78.5	78.5
5	11.5	20.8	31.8	31.8	31.8	41.2	50.6	52.1	52.1	52.1
6	18.9	32.9	49.4	49.4	49.4	63.5	77.6	79.8	79.8	79.8
7	14.7	29.7	47.4	47.4	47.4	62.6	77.8	80.2	80.2	80.2
8	12.0	26.9	44.7	44.7	44.7	59.9	75.0	77.4	77.4	77.4
9	12.0	26.9	44.7	44.7	44.7	59.9	75.0	77.4	77.4	77.4
10	12.0	26.9	44.7	44.7	44.7	59.9	75.0	77.4	77.4	77.4

Table C.3 Percentage difference in mass flow rate between values obtained from integration over section area and values obtained from inlet flow rates and open hole area with the second flow rate as the base value for all sections and runs given in Tables C.1 and C.2.

$\Delta m\% = [(\dot{m}_i - \dot{m}_h) / \dot{m}_h] \times 100$												Ave*
Sec. #	1	2	3	4	5	6	7	8	9	10		
Run #												
1	-10.3	4.04	-4.65	18.2	-1.2	1.7	16.8	-0.5	8.1	2.5	3.5	
2	-2.9	-8.9	-11.40	8.4	20.2	0.8	-5.5	-6.0	10.3	2.2	0.7	
3	4.6	-12.9	-0.4	21.8	17.9	2.1	11.0	4.8	-2.5	-0.2	4.6	
4	-8.6	-13.1	1.0	8.0	2.0	3.0	3.6	4.2	15.0	-2.5	1.3	
5	-2.7	-5.2	-15.6	43.5	10.0	13.5	0.2	0.0	14.8	-2.0	3.7	
6	2.9	-2.9	-6.1	1.3	2.0	1.4	4.6	-0.9	1.5	-0.7	0.3	
7	3.7	-14.9	7.6	-7.1	3.1	8.4	3.7	3.0	-0.3	-1.5	0.6	
8	24.5	-14.6	-15.9	4.8	2.0	6.3	3.9	18.7	16.8	-1.8	4.5	
9	7.74	-13.7	-12.1	10.1	-8.1	4.8	4.5	14.4	10.8	-2.0	1.65	
10	-4.1	-23.1	-6.7	1.2	7.4	4.5	10.2	19.5	11.8	0.8	2.13	
Average	1.5	-10.5	-6.4	11.0	3.5	4.7	5.3	5.7	8.6	-0.5	2.3	

* The average given in this table was calculated as the arithmetic average which means that it was the total value of the error in the column or the row (taking the error sign into consideration) divided by the number of sections or runs respectively.

Table C.4 Axial flux of angular momentum calculated from integration over the area for the various sections and for all the ten runs.

Axial flux of angular momentum $\times 10^4$ Nm											
Run # \ Sec. #	1	2	3	4	5	6	7	8	9	10	
1	-1.57	1.15	1.92	0.0217	-1.74	0.157	-5.5	-2.78	-2.04	-1.41	
2	-1.17	-1.06	-1.78	-0.44	-1.11	0.18	1.02	2.19	3.29	0.27	
3	-0.98	-0.89	-0.42	-1.50	-1.73	3.20	-0.56	5.03	2.44	1.96	
4	-6.00	-0.68	-7.02	0.51	1.11	0.19	3.05	2.59	1.83	6.20	
5	-0.341	0.66	0.114	-0.968	-0.56	-0.288	0.71	.153	-.011	-0.54	
6	0.720	0.546	0.613	-0.223	-0.91	-0.412	-.79	-1.75	3.23	1.41	
7	-0.833	-.277	-.503	-1.28	-1.99	-0.89	0.189	0.74	1.64	1.19	
8	0.193	1.31	0.9	-0.82	-1.2	0.43	0.27	1.74	1.92	2.23	
9	0.05	0.01	-0.2	-3.7	-0.665	0.9	1.82	.664	1.41	2.41	
10	0.22	0.23	0.2	-0.4	-0.348	-0.07	3.81	3.37	0.85	1.96	

Table C.5 Axial flux of angular momentum at combustor inlet
for the ten runs ($G_{\phi i}$)

Run #	1	2	3	4	5	6	7	8	9	10
$G_{\phi i} \times 10^4$ Nm	0.078	0.0788	0.0788	0.0788	0.035	0.0788	0.0788	0	0.0263	0.0455

Table C.6 Difference in axial flux of angular momentum from calculations at the various sections and the inlet value to the combustor.

$(G_{\phi_f} - G_{\phi_i}) \times 10^4 \text{ Nm}$											
Sec. #	1	2	3	4	5	6	7	8	9	10	
Run #											
1	-1.65	1.07	1.84	-0.06	-1.82	0.078	5.58	-2.86	-2.12	1.49	
2	-1.25	-1.74	-1.85	-0.52	-1.19	-0.26	0.94	2.11	3.21	0.19	
3	-1.06	-0.97	-0.50	-1.58	-1.81	3.12	-0.64	4.95	2.36	1.88	
4	-6.08	-0.76	-7.10	0.43	1.03	0.11	2.97	2.51	1.75	6.12	
5	-0.38	0.63	0.08	-1.00	-0.60	-0.32	0.68	0.12	-0.05	-0.58	
6	0.64	0.47	0.53	-0.31	-0.99	-0.49	-0.87	-1.83	3.15	1.33	
7	-0.91	-0.36	-0.58	-1.36	-1.07	-0.97	0.11	0.66	1.56	1.11	
8	0.19	1.31	0.90	-0.82	-1.20	0.43	0.27	1.74	1.92	2.23	
9	0.02	-0.04	-0.22	-3.73	-0.69	0.87	1.79	0.64	1.38	2.38	
10	0.17	0.18	0.15	-0.44	-0.39	-0.11	3.76	3.30	0.80	1.91	

Table C.7 Results of average temperature calculated from energy balance at the first section, Eq. (C.6), and from the temperature measurements at the first section for the ten runs (temperature is given in $^{\circ}\text{C}$).

method of obtaining $t_{\text{1st section}}$	Run #									
	1	2	3	4	5	6	7	8	9	10
from energy balance Eq. (C.6)	1210	1210	1210	1206	1200	755	851	1117	1117	1076
average of measurements at first section	1219	1218	1200	1198	1190	747	841	1107	1110	1070

Fig. C.1 Control volume at the combustor axis

APPENDIX D

SHAPE FACTOR AND TRAVERSE NUMBER

These non-dimensional parameters, namely the shape factor and the traverse number, are used to describe the quality of the temperature profile at the exit section of a combustor. This temperature profile is very important and usually is developed to optimize the lives of the high pressure turbine stator vanes and the rotor blades. Since the vanes are fixed relative to the combustor, they have to be designed to accept the highest temperature that will be experienced at combustor exit, whereas the blade only feels the circumferentially averaged temperature at any radius. The design of the vanes therefore, represents the most severe temperature in an engine. Then it is essential to be able to define a non-dimensional parameter (factor) to represent the measured temperature profile. The traverse number is defined to represent the effect of the peak temperature in non-dimensional form while the shape factor represents the temperature quality at any position at the combustor exit section. The mathematical expressions for these parameters were given before in Chapter 5, however, for the sake of clarity they are repeated here.

$$\text{Traverse Number} = (T_{\max} - T_m) / (T_m - T_a) \quad (D.1)$$

$$\text{Shape Factor} = (T - T_m) / (T_m - T_a) \quad (D.2)$$

The design of the blades of the stator is affected by the traverse number (together with the mean temperature)

and hence the degree of sophistication of the blade design is dictated by this factor. If the shape factor experiences a large negative area (or a large negative-value) it implies that the blade strength is not fully utilized. Thus, the extent of blade utilization can be identified by the extent to which the shape factor becomes negative.

This non-dimensional parameter changes the conventional approach of evaluating the temperature profile of a combustor at its exit section. While

a smooth temperature profile is desired, it is also essential that the shape factor profile calculated from such a profile does not exhibit high negative values. This could be described by the following examples. If the temperature distribution at the exit section could be represented by a half sine wave, the resulting shape factor will result in high negative values at the extremities of the profile. The resulting shape factor profile could be approximated by a cosine wave shifted $1/4$ of a cycle, Fig. D.1. In spite of the temperature profile in Fig. D.1 being smooth and the traverse number occurring at the center of the blade, high negative values of the shape factor exist at the extremities.

A better temperature profile is given in Fig D.2, a half parabola with the maximum temperature at one end of the nozzle blade. The resulting shape factor profile is represented also on the same figure. This distribution results in a traverse number at one end of the blade but a smaller negative area in the shape factor profile. If the temperature profile is represented by a horizontal straight line. The shape factor in this case will not experience any negative value. While this is quite suitable for the nozzle guide vanes, it will not suit the rotor blades. Since the stresses are high near the rotor blade root, due to the manner of mounting, the blades in this case will have to be overdesigned.

The above description recommends a temperature profile which while resulting in a traverse number occurring near (but not at) the tip of the blade, also results in a shape factor with a maximum negative value less than 0.2, preferably at the blade root.

Fig. D.1 Sinusoidal temperature profile
and the corresponding shape
factor

Fig. D.2 Half parabolic temperature
distribution and the
corresponding shape factor

APPENDIX E

APPLICATION OF THE "CONVENIENT REACTION RATE EQUATION" AND THE ODGERS-CARRIER PARAMETER TO THE TEST COMBUSTOR

This appendix gives an outline of the procedure followed to apply the "convenient reaction rate equation", Eq. (6.1), developed by Odgers and co-workers [58,80] and the Odgers-Carrier parameter, Eq. (6.2), [80]. The convenient reaction rate equation is given here again, for easy reference, as follows:

$$\frac{N}{VP^2 \phi} = \frac{1.29 \times 10^{10} (n^* + 1) \{5(1 - Y_E)\}^\phi \{ \phi - Y_E \}^\phi e^{-C/(T_i + \epsilon \Delta T)}}{0.082^{2\phi} Y_E \{5(n^* + 1) + \phi + Y_E\}^{2\phi} \{T_i + \epsilon \Delta T\}^{2\phi - 0.5}}$$

(6.1).
repeat

Eq. (6.1) was derived assuming that propane fuel is used. Hence, the equation needs to be modified for the fuel of the present study, namely, low-medium heating value gas. The chemical equation for burning 1 mole of low-medium heating value fuel with the same composition as that used during the base run, run #1, (Table 5.1) is given as follows:

Where ϕ = equivalence ratio: $(\text{oxygen/fuel})_{\text{sto}} / (\text{oxygen/fuel})_{\text{actual}}$

n^* = ratio of inerts/oxygen by volume in atmospheric air

ϵ = fractional oxygen consumption efficiency

$Y = \phi$ for $\phi \leq 1$, $Y = 1$ for $\phi > 1$

Substituting from the mass balance chemical equation above in Eq. (6.1) results in the following form.

$$\frac{N}{VP^{2\phi}} = \frac{1.29 \times 10^{10} (n^* + 1) \{ 0.404(1 - Y_{\epsilon}) \}^{\phi} \{ \phi - Y_{\epsilon} \}^{\phi} e^{-C/(T_i + \epsilon \Delta T)}}{0.0.82^{2\phi} Y_{\epsilon} \{ 0.404(n^* + 1) + \phi \}^{2\phi} \{ T_i + \epsilon \Delta T \}^{2\phi - 0.5}}$$

(E.1)

The following values are either given by the conditions corresponding to run #1 (Table 5.1) or calculated from these conditions.

N = mole flow rate into the reaction zone of the combustor

= 0.165 mole/s (based on air and fuel flow rates in the primary zone)

T_i = initial air temperature at inlet to reaction zone (300°K)

ΔT = undissociated adiabatic temperature rise in the reaction zone (1745°C)

ϕ = 0.412

n^* = 3.86

$C = E/\bar{R}$

E = activation energy (20,000 cal/mole) based on the

work by Minkoff and Tipper [E1] on mixtures of CO and H₂

$\bar{R}$ = universal gas constant = 1.986 kcal/kgmole⁰ K

V' = volume of reaction zone (primary zone) = 0.3475 l

Substituting the above values in Eq. (E.1) and solving (by trial and error) for ϵ the resulting value is 0.055. This corresponds to a combustion efficiency η of 0.055 which is unrealistic in comparison with the real experimental measurements. It was found that Eq. (E.1) is very sensitive to the exponential term especially the value of C . When the value of $C = 28,000$ was substituted in the equation (which was the value corresponding to weak mixtures of propane fuel in air as used by Kretschmer and Odgers [58]), the resulting value for ϵ and therefore for the combustion efficiency η was 0.53 which is one order of magnitude higher than the value obtained with the value of C corresponding to medium heating value fuel. This indicates that this equation is applicable only to saturated hydrocarbon fuels (which have values of C less than 20,000).

The Odgers-Carrier parameter, Eq. (6.2), is given as follows:

$$\tau = 0.911 \log \psi_{300} + 4.01n - 1.097 + D' \quad (E.2)$$

where $\tau = \log \log 100/\eta$

η = combustion efficiency

ψ = fuel loading factor at 300°K = $\dot{m}_f / V'P$

V' = volume of primary zone (ft³)

$\dot{m}_f$ = fuel mass flow rate (lb/sec)

n = reaction order

P = operating inlet pressure (atm)

D' = combustor constant

From Table 5.1 b, the following values are extracted for runs #13 and #14.

Run #13 $\dot{m}_f = 2.19 \text{ kg/h}$

$\eta = 82.5\%$

Run #14 $\dot{m}_f = 2.55 \text{ kg/h}$

$\eta = 72.4\%$

From the combustor design, $V' = 0.006136 \text{ ft}$

Since the combustor inlet pressure is nearly atmospheric, the loading factor could be calculated as follows:

For run #13
$$\psi = \frac{2.19 \times 2.2046}{3600 \times 0.006136} = 0.2186$$

Assuming $n = 2$ and substituting in Eq. (E.2), results in

$$\log \log 100/82.5 = 0.911 \log 0.2186 + 4.01 \times 2.0 - 1.097 + D'$$

Therefore $D' = -7.386$, and Eq. (E.2) takes the form

$$\eta = 0.911 \log \psi_{300} + 4.01n - 1.097 - 7.386 \quad (\text{E.3})$$

For run #14
$$\psi = \frac{2.55 \times 2.2046}{3600 \times 0.006136} = 0.2545$$

Substituting in Eq. (E.3)

$$\log \log 100/\eta = 0.911 \log 0.2545 + 4.01 \times n - 1.097 - 7.386 = -1.0044$$

This results in a value for η of 0.796 which is higher than the value calculated from temperature measurements at the exit section for run #14. However, if Eq. (6.4) is used which is a modified arrangement of Eq. (E.2) but the fuel loading factor term was increased, the combustor constant D' could be calculated as follows:

$$\log \log 100/82.5 = 3.35 \log 0.2186 + 4.36 \times 2.0 - 1 + D'$$

$D' = -6.572$, and Eq. (6.4) takes the form

$$\log \log 100/\eta = 3.35 \log \psi_{300} + 4.36x\eta - 1 - 6.572 \quad (\text{E.4})$$

Substituting for the fuel loading factor of run #14 results in $\eta = 72\%$. It should be noted here that the Odgers-Carrier parameter is limited to be used only for the same total air flow rate and combustor geometry and assuming that the fuel composition (or heating value) is the same.

El. Minkoff, G.J. and C.F.H. Tipper, Chemistry of Combustion Reactions, Butterworths, London 1962.

ANNEX 1 of
Ph.D. Thesis

on

INVESTIGATION OF COMBUSTION CHARACTERISTICS OF LOW-MEDIUM
HEATING VALUE GASES IN AN ATMOSPHERIC PRESSURE COMBUSTOR

by

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1984

FOREWORD

This annex contains a listing of the numerical values of temperature and the three components of velocity together with exit section shape factor for the ten guns which were reported in Chapter 5 of the thesis. Also, it contains the temperature values at the exit section for the extra ten runs that were used to develop the combustor efficiency formula and to check the formula. This is followed by a listing of the computer program that was used to carry out the calculations required in Appendix C.

GLOSSARY

TEMP = temperature

VR = radial velocity component

* VT = circumferential velocity component

VZ = axial velocity component

RUN # 1

1

SEC. #1

TEMP	VZ	VT	VR
1407.0000	0.0000	0.0000	0.0000
1420.0000	11.4000	-1.4000	0.4000
1264.0000	10.0000	-1.4000	0.3500
1144.0000	8.0000	0.5000	0.3000
1124.0000	5.0000	3.1000	0.2000
1130.0000	-5.0000	3.2000	-0.6000
1141.0000	-5.5000	3.0000	0.3500
1132.0000	-3.6000	3.6000	0.4000
1132.0000	-3.6000	2.6000	1.6000
1134.0000	-2.4000	3.8000	1.7000
1158.0000	-2.1000	-3.2000	1.7500
1175.0000	-3.1000	-1.5000	1.8000
1221.0000	5.5000	-1.0000	1.9000
1304.0000	6.5000	-2.0000	1.7500
1338.0000	9.0000	-2.0000	1.3000
1279.0000	0.0000	0.0000	0.0000

SEC. #2

TEMP	VZ	VT	VR
878.0000	0.0000	0.0000	0.0000
1110.0000	14.0000	-0.5000	0.5000
1260.0000	13.0000	-0.5000	4.1000
1210.0000	12.0000	0.4000	6.1000
1120.0000	10.6000	3.0000	5.2000
1063.0000	7.6000	10.5000	-5.2000
940.0000	2.6000	10.5000	-5.2000
910.0000	-1.1000	10.5000	-5.1000
920.0000	-1.4000	10.5000	-4.5000
1000.0000	-0.9000	11.0000	2.1000
1070.0000	-0.5000	6.0000	4.0000
1144.0000	4.5000	4.5000	5.5000
1196.0000	6.5000	2.2000	5.1000
1205.0000	7.1000	0.5000	5.2000
1004.0000	12.1000	-1.0000	0.5000
820.0000	0.0000	0.0000	0.0000

SEC. #3

TEMP	VZ	VT	VR
638.0000	0.0000	0.0000	0.0000
790.0000	6.2000	0.3000	1.6000
762.0000	6.0000	0.4000	3.5000
740.0000	5.8000	0.7000	4.0000
680.0000	4.8000	0.2000	3.8000
570.0000	-2.3000	1.0000	1.5000
465.0000	6.0000	1.5000	-2.0000
407.0000	18.0000	1.7000	-2.2000
390.0000	22.1000	2.2000	-2.0000
413.0000	25.1000	2.5000	-0.9000
456.0000	25.0000	0.7000	1.8000

551.0000	18.1000	2.7000	4.6000
676.0000	-2.5000	0.7000	5.1000
767.0000	-2.8000	0.4000	5.2000
806.0000	6.0000	0.4000	4.0000
818.0000	7.8000	0.4000	3.5000
810.0000	0.0000	0.0000	0.0000

SEC. #4

TEMP	VZ	VT	VR
597.0000	0.0000	0.0000	0.0000
695.0000	2.8000	-0.1600	-1.1000
698.0000	2.9000	-0.2200	-1.2000
675.0000	3.0000	-0.5000	-1.7000
625.0000	9.5000	-0.5000	-3.7000
560.0000	10.2000	0.0000	-4.1000
502.0000	15.2000	0.1800	-4.1000
478.0000	18.4000	0.3400	-4.2000
477.0000	27.1000	0.2800	-3.9000
508.0000	26.5000	0.2400	-3.1000
551.0000	18.1000	0.1400	-3.0000
525.0000	15.0000	0.0800	-3.0000
648.0000	10.5000	0.1000	-4.0000
702.0000	9.2000	0.2400	-1.5000
753.0000	7.9000	0.1400	-1.1000
782.0000	2.7000	0.0000	-0.5000
780.0000	0.0000	0.0000	0.0000

SEC. #5

TEMP	VZ	VT	VR
558.0000	0.0000	0.0000	0.0000
686.0000	4.0000	-0.5000	-1.5000
727.0000	4.2000	-0.5000	-1.6000
723.0000	4.1000	-0.7000	-1.7000
705.0000	5.6000	-0.8000	-2.1000
661.0000	8.5000	-0.7000	-4.1000
608.0000	10.9000	-0.6500	-5.5000
571.0000	11.7000	-0.4000	-5.6000
563.0000	14.6000	-0.5000	-6.1000
579.0000	18.7000	-0.5500	-6.0000
623.0000	14.6000	-0.5000	-5.1000
683.0000	12.1000	-0.5500	-4.7000
742.0000	10.0000	-0.6000	-3.5000
782.0000	8.2000	-0.7000	-3.1000
800.0000	8.2000	-0.7500	-3.1000
783.0000	6.0000	-0.7500	-1.0000
818.0000	0.0000	0.0000	0.0000

SEC. #6

TEMP	VZ	VT	VR
375.0000	0.0000	0.0000	0.0000
365.0000	4.1000	-0.1500	0.6000
380.0000	5.5000	-0.1500	-0.3000
460.0000	5.0000	-0.1500	-0.3000
564.0000	5.5000	-0.1000	-1.7000

606.0000	7.2000	0.4500	-2.2000
611.0000	8.1000	0.6000	-3.2000
612.0000	10.5000	0.8200	-3.3000
611.0000	11.7000	0.7000	-2.8000
612.0000	12.8000	0.5000	-2.8000
603.0000	14.1000	-0.2000	-2.8000
580.0000	11.9000	-0.1500	-2.8000
540.0000	9.1000	0.2000	-2.8000
494.0000	7.4000	0.2500	-3.1000
434.0000	6.8000	-0.2700	-0.5000
388.0000	6.1000	0.2000	-0.5000
388.0000	5.0000	0.1500	1.5000
442.0000	0.0000	0.0000	0.0000

SEC. #7

TEMP	VZ	VT	VR
344.0000	0.0000	0.0000	0.0000
353.0000	4.0000	0.4000	0.9000
370.0000	6.0000	-2.0000	1.2000
462.0000	3.0000	-1.2000	5.1000
479.0000	2.3000	-0.8500	-1.2000
492.0000	15.3000	-0.6000	-1.1000
505.0000	10.2000	-0.4000	0.0000
508.0000	10.2000	-0.4000	0.5000
506.0000	10.4000	-0.4000	1.1000
500.0000	10.4000	-0.4000	0.0000
487.0000	10.7000	-0.5000	-3.1000
465.0000	10.5000	-0.5000	-1.7000
442.0000	10.5000	-1.2000	-1.1000
408.0000	10.0000	-3.0000	4.0000
365.0000	10.1000	-2.4000	1.1000
392.0000	15.6000	-0.6000	0.8000
330.0000	6.4000	0.0000	0.5000
339.0000	5.6000	0.0000	0.0000
348.0000	0.0000	0.0000	0.0000

SEC. #8

TEMP	VZ	VT	VR
351.0000	0.0000	0.0000	0.0000
408.0000	2.0000	-0.2000	0.5000
408.0000	3.0000	-0.3000	1.5000
375.0000	10.1000	-0.5000	1.9000
416.0000	8.0000	-0.6500	4.6000
474.0000	7.0000	-0.6500	5.1000
491.0000	7.0000	-0.5500	3.7000
487.0000	8.5000	-0.3500	1.9000
474.0000	9.0000	-0.1500	1.6000
469.0000	10.5000	0.0000	1.3000
466.0000	12.0000	0.1500	-1.4000
455.0000	9.0000	0.2000	-1.0000
430.0000	8.1000	-0.1500	-0.4000
388.0000	7.6000	-0.2000	-0.9000
362.0000	6.0000	-0.3000	-1.7000
340.0000	7.0000	-0.4000	-2.4000

358.0000 4.0000 -0.9000 -2.3000
 368.0000 3.3000 -0.7500 -2.2000
 389.0000 2.3000 -0.3000 -2.7000
 392.0000 0.0000 0.0000 0.0000

SEC. #9

TEMP VZ VT VR

349.0000 0.0000 0.0000 0.0000
 384.0000 1.1000 -0.3000 0.6000
 385.0000 2.5000 -1.2000 1.2000
 392.0000 3.5000 -0.2500 1.4000
 390.0000 15.1000 -0.1000 1.8500
 412.0000 14.7000 -0.1000 1.9000
 423.0000 13.5000 -0.1000 1.5000
 422.0000 12.4000 -0.1000 1.1000
 413.0000 10.4000 -0.1000 1.1000
 401.0000 10.4000 -0.1000 1.5000
 398.0000 10.4000 -0.1000 1.3500
 392.0000 9.4000 -0.1000 1.4000
 370.0000 8.4000 -0.2500 1.9500
 352.0000 8.4000 -0.5000 1.9000
 342.0000 7.8000 -0.8000 1.2000
 334.0000 3.3000 -0.3000 1.3000
 326.0000 2.7000 -1.5000 1.5500
 342.0000 2.0000 -0.2000 1.1500
 370.0000 1.2000 0.0000 1.8500
 380.0000 0.0000 0.0000 0.0000

SEC. #10

TEMP VZ VT VR

373.0000 0.0000 0.0000 0.0000
 384.0000 24.0000 -0.9000 3.6000
 378.0000 24.3000 -0.2000 2.6000
 364.0000 24.6000 0.5000 1.5000
 360.0000 24.2000 0.6000 0.5000
 359.0000 24.4000 0.7000 0.0000
 370.0000 24.2000 0.8000 -0.5000
 365.0000 24.0000 0.5000 -0.5000
 362.0000 23.8000 -0.2000 2.0000
 380.0000 23.6000 -1.7000 3.0000
 323.0000 0.0000 0.0000 0.0000

RUN # 2

5

SEC. #1

TEMP	VZ	VT	VR
1300.0000	0.0000	0.0000	0.0000
1426.0000	9.2000	-0.6000	-3.7000
1322.0000	16.1000	-0.8000	-3.5000
1200.0000	-5.0000	-0.5000	-3.5000
1140.0000	-7.1000	0.5000	-3.4000
1145.0000	-8.1000	0.3000	-1.8000
1140.0000	-8.5000	0.2500	-2.1000
1135.0000	-7.6000	0.3500	-2.2000
1140.0000	-6.6000	0.4800	-2.6000
1147.0000	-6.2000	1.7000	-2.8000
1136.0000	4.4000	1.9000	-2.8000
1175.0000	6.1000	0.0000	-2.0000
1277.0000	15.1000	-1.0000	-2.0000
1320.0000	10.4000	-1.1000	-3.6000
1340.0000	8.6000	-0.6000	-4.0000
1292.0000	0.0000	0.0000	0.0000

SEC. #2

TEMP	VZ	VT	VR
996.0000	0.0000	0.0000	0.0000
1190.0000	6.2000	-1.2000	-3.4000
1280.0000	8.0000	-2.2000	-3.6000
1245.0000	9.1000	-2.0000	-3.8000
1185.0000	7.1000	2.0000	-3.8000
1109.0000	4.1000	4.0000	-1.5000
1040.0000	-3.4000	3.5000	-0.9000
995.0000	-2.7000	0.8000	0.0000
1020.0000	-2.5000	0.0000	0.0000
1045.0000	1.5000	-1.7000	0.0000
1090.0000	6.6000	-1.7000	0.0000
1145.0000	7.5000	-0.6000	-1.6000
1183.0000	7.1000	-0.4000	-2.4000
1208.0000	8.8000	-1.5000	-2.9000
1246.0000	10.1000	-1.0000	-2.7000
1240.0000	0.0000	0.0000	0.0000

SEC. #3

TEMP	VZ	VT	VR
820.0000	0.0000	0.0000	0.0000
894.0000	6.5000	0.4000	-3.1000
845.0000	8.1000	-0.4000	-4.0000
765.0000	-2.0000	-0.7000	-4.0000
680.0000	-4.5000	-0.6000	-3.8000
565.0000	-3.1000	-0.3000	-1.1000
465.0000	9.1000	0.3000	1.8000
415.0000	10.4000	2.0000	2.2000
418.0000	12.5000	0.6000	4.5000
477.0000	14.5000	-0.4000	5.1000
597.0000	15.5000	-0.2000	5.2000

739.0000	15.5000	-1.2000	5.0000
836.0000	9.1000	-2.0000	1.0000
873.0000	-2.7000	-2.2000	-1.5000
900.0000	6.2000	-2.4000	-2.1000
876.0000	8.6000	-1.6000	-2.5000
730.0000	0.0000	0.0000	0.0000

SEC. #4

TEMP	VZ	VT	VR
814.0000	0.0000	0.0000	0.0000
910.0000	1.2000	0.3000	-3.1000
856.0000	1.3000	0.3000	-3.1000
758.0000	1.2000	0.4000	-3.2000
658.0000	1.3000	0.6000	-5.1000
570.0000	7.8000	1.2000	-5.1000
520.0000	10.1000	1.4000	-4.0000
505.0000	11.2000	1.4000	-2.7000
528.0000	12.4000	1.0000	-2.0000
570.0000	14.1000	0.4000	1.8000
615.0000	13.2000	-0.2000	4.4000
705.0000	12.1000	-0.4000	6.7000
733.0000	11.4000	-0.4000	2.5000
742.0000	9.6000	-0.6000	2.1000
833.0000	9.2000	-0.6000	2.1000
890.0000	7.1000	-0.5000	2.1000
903.0000	0.0000	0.0000	0.0000

SEC. #5

TEMP	VZ	VT	VR
667.0000	0.0000	0.0000	0.0000
878.0000	5.1000	0.4000	-2.8000
840.0000	5.2000	0.4000	-3.2000
777.0000	5.3000	0.4000	-4.1000
687.0000	5.2000	0.4500	-4.5000
613.0000	4.9000	0.7000	-5.1000
555.0000	7.6000	1.1000	-4.5000
536.0000	7.8000	1.4000	-4.0000
539.0000	8.9000	1.4000	-3.1000
560.0000	14.6000	1.1000	-2.4000
605.0000	13.1000	1.0000	-2.4000
670.0000	10.9000	-1.4000	-2.6000
742.0000	10.7000	-1.6000	-2.8000
800.0000	10.2000	-1.0000	-3.2000
838.0000	9.7000	-1.0000	-3.8000
839.0000	5.1000	-1.0000	-4.4000
900.0000	0.0000	0.0000	0.0000

SEC. #6

TEMP	VZ	VT	VR
449.0000	0.0000	0.0000	0.0000
466.0000	2.2000	1.0000	-1.1000
428.0000	3.9000	1.0000	-0.5000
483.0000	4.9000	1.0000	-0.6000
615.0000	5.5000	0.8000	-1.1000

639.0000	6.5000	0.6000	-2.4000
624.0000	9.6000	0.4000	-2.6000
614.0000	12.1000	0.2000	-2.4000
610.0000	11.2000	0.0000	-1.2000
615.0000	10.1000	-0.1000	-0.8000
622.0000	9.9000	-0.2000	-0.2000
614.0000	9.1000	-0.4000	-0.2000
586.0000	8.5000	-0.5000	-1.2000
504.0000	7.3000	-0.6000	-2.0000
437.0000	7.2000	-0.6000	-2.2000
428.0000	7.1000	-0.7000	-2.0000
490.0000	6.5000	-0.5000	-1.0000
565.0000	0.0000	0.0000	0.0000

SEC. #7

TEMP	VZ	VT	VR
265.0000	0.0000	0.0000	0.0000
268.0000	4.8000	0.1000	-0.3000
360.0000	5.2000	0.1500	-0.4000
515.0000	6.1000	0.2000	-1.0000
545.0000	7.2000	0.3000	-1.2000
550.0000	7.5000	0.6000	-1.4000
555.0000	7.4000	0.7500	-1.4000
553.0000	8.6000	0.8000	-1.8000
550.0000	8.5000	0.8000	-2.0000
544.0000	8.5000	0.5000	-2.4000
539.0000	8.5000	0.2000	-2.2000
526.0000	8.5000	0.2000	-2.0000
503.0000	8.6000	0.1000	-1.4000
470.0000	7.4000	0.0000	-1.2000
420.0000	6.5000	0.0000	-1.0000
340.0000	7.2000	0.0000	-0.6000
305.0000	6.2000	0.3000	-0.4000
300.0000	4.0000	0.2000	-0.2000
300.0000	0.0000	0.0000	0.0000

SEC. #8

TEMP	VZ	VT	VR
360.0000	0.0000	0.0000	0.0000
399.0000	1.8000	0.1000	-1.0000
401.0000	2.2000	0.1500	-1.4000
425.0000	5.7000	0.2000	-1.4000
476.0000	13.1000	0.3500	-1.1000
527.0000	9.6000	0.5500	-1.0000
543.0000	9.1000	0.7000	-1.0000
547.0000	9.6000	0.7500	-1.6000
536.0000	9.0000	0.8000	-2.0000
526.0000	9.9000	0.7500	-2.0000
515.0000	9.5000	0.6500	-2.2000
491.0000	9.0000	0.4500	-2.0000
462.0000	8.5000	0.4000	-2.0000
429.0000	7.2000	0.3500	-2.0000
387.0000	5.5000	0.3000	-2.0000
357.0000	5.5000	0.4500	-2.2000

336.0000	4.1000	0.6500	-2.6000
334.0000	3.5000	0.4000	-2.2000
367.0000	1.5000	0.2000	-1.0000
386.0000	0.0000	0.0000	0.0000

8

SEC. #9

TEMP	VZ	VT	VR
350.0000	0.0000	0.0000	0.0000
404.0000	2.0000	0.2000	0.0000
423.0000	3.5000	0.6500	0.0000
441.0000	5.6000	0.8500	-0.2000
466.0000	6.8000	0.9000	-0.4000
495.0000	8.9000	0.6000	-0.5000
513.0000	10.2000	0.5000	-0.8000
510.0000	10.2000	0.5500	-1.0000
502.0000	10.0000	0.6000	-2.2000
494.0000	10.7000	0.6500	-2.2000
474.0000	10.5000	0.6000	-2.4000
454.0000	10.8000	0.4500	-2.0000
428.0000	10.6000	0.3500	-2.2000
400.0000	10.6000	0.2000	-2.2000
373.0000	8.6000	0.2000	-2.0000
361.0000	7.5000	0.4000	-2.2000
351.0000	6.5000	0.5500	-1.4000
348.0000	5.5000	0.5000	-1.2000
391.0000	2.2000	0.3000	-1.0000
384.0000	0.0000	0.0000	0.0000

SEC. #10

TEMP	VZ	VT	VR
376.0000	0.0000	0.0000	0.0000
406.0000	21.9000	0.2000	0.6000
422.0000	24.4000	0.3000	0.2000
419.0000	29.9000	0.4000	-0.2000
409.0000	29.4000	0.4000	-0.8000
392.0000	29.1000	0.3500	-1.7000
387.0000	27.1000	0.2000	-2.0000
365.0000	24.2000	-0.1500	-1.2000
339.0000	23.7000	-0.1000	2.2000
320.0000	21.0000	-0.1000	0.5000
311.0000	0.0000	0.0000	0.0000

RUN # 3

9

SEC. #1

TEMP	VZ	VT	VR
1239.0000	0.0000	0.0000	0.0000
1336.0000	6.5000	-1.2000	-1.4000
1330.0000	10.7000	-0.6000	-1.4000
1273.0000	14.2000	-0.4000	1.4000
1139.0000	10.7000	-0.4000	3.7000
1068.0000	-11.1000	-2.5000	3.9000
1053.0000	-11.1000	-2.5000	3.7000
1055.0000	-11.2000	-2.5000	1.6000
1055.0000	-11.1000	-2.0000	0.6000
1084.0000	-11.1000	-1.9000	0.0000
1108.0000	-11.5000	-1.4000	0.0000
1141.0000	6.1000	-0.6000	-0.6000
1177.0000	14.0000	-0.2000	-4.0000
1350.0000	9.5000	-1.2000	2.0000
1388.0000	7.0000	-1.4000	4.6000
1390.0000	0.0000	0.0000	0.0000

SEC. #2

TEMP	VZ	VT	VR
1088.0000	0.0000	0.0000	0.0000
1108.0000	10.1000	-0.3500	1.6000
1075.0000	13.7000	-0.6000	3.6000
1025.0000	15.1000	-0.6000	4.2000
930.0000	9.6000	-0.5500	3.5000
802.0000	-2.2000	-0.4000	-1.0000
758.0000	-10.1000	0.2000	-1.6000
768.0000	-11.2000	0.2000	4.0000
785.0000	-10.2000	0.2000	6.0000
816.0000	-8.0000	0.0000	4.6000
840.0000	-8.5000	-0.6000	4.0000
893.0000	4.5000	-0.7000	-1.0000
998.0000	12.5000	-0.7000	-2.0000
1014.0000	13.2000	0.9000	0.8000
916.0000	12.0000	-0.9500	2.0000
900.0000	0.0000	0.0000	0.0000

SEC. #3

TEMP	VZ	VT	VR
665.0000	0.0000	0.0000	0.0000
685.0000	12.1000	0.1000	3.0000
658.0000	16.2000	0.1000	3.0000
610.0000	11.2000	0.1000	3.0000
550.0000	-2.0000	0.1000	2.0000
465.0000	-7.7000	0.1000	1.0000
416.0000	-10.1000	0.1000	0.0000
400.0000	-11.0000	0.1000	-1.6000
421.0000	-10.5000	0.1000	-3.6000
466.0000	25.5000	0.0500	-3.1000
530.0000	24.1000	0.0000	-3.1000

590.0000	22.1000	-0.0500	-4.0000
640.0000	20.1000	-0.1000	-5.0000
690.0000	8.2000	-0.4000	-4.2000
762.0000	6.8000	-0.5000	-3.6000
760.0000	5.6000	-0.6000	-1.8000
743.0000	0.0000	0.0000	0.0000

SEC. #4

TEMP	VZ	VT	VR
537.0000	0.0000	0.0000	0.0000
690.0000	1.1000	1.1000	-1.0000
680.0000	1.2000	1.2000	-1.4000
640.0000	1.5000	1.2000	-3.0000
580.0000	3.5000	1.1000	-6.0000
522.0000	6.6000	0.9000	-8.0000
481.0000	15.1000	0.8500	-8.2000
447.0000	19.2000	0.8000	-8.0000
437.0000	22.1000	0.6000	-5.6000
445.0000	25.0000	0.3000	-3.6000
465.0000	24.2000	-0.3000	8.0000
495.0000	22.6000	-0.6000	10.0000
519.0000	18.1000	-0.6000	10.0000
553.0000	15.1000	-0.7000	-7.0000
638.0000	12.1000	-0.8000	-7.0000
683.0000	5.5000	-1.2000	-2.0000
706.0000	0.0000	0.0000	0.0000

SEC. #5

TEMP	VZ	VT	VR
584.0000	0.0000	0.0000	0.0000
649.0000	1.1000	0.1500	-1.0000
625.0000	1.2000	0.1700	-1.8000
595.0000	1.2000	0.2000	-2.0000
555.0000	1.5000	0.4000	-4.0000
516.0000	5.1000	0.6000	-5.0000
486.0000	8.1000	0.7500	-6.5000
468.0000	10.5000	0.6000	-4.0000
461.0000	15.7000	0.4000	-1.6000
471.0000	20.1000	0.3000	0.0000
481.0000	22.6000	0.0000	3.8000
494.0000	25.5000	-0.4000	3.9000
506.0000	22.5000	-0.4000	4.0000
524.0000	15.4000	-0.6000	3.4000
615.0000	12.1000	-0.8000	3.2000
635.0000	6.1000	-1.0000	2.4000
636.0000	0.0000	0.0000	0.0000

SEC. #6

TEMP	VZ	VT	VR
379.0000	0.0000	0.0000	0.0000
436.0000	1.2000	1.2000	-0.7000
446.0000	1.3000	1.2000	-2.0000
464.0000	2.1000	1.2000	-3.0000
488.0000	4.2000	1.2000	-4.0000

503.0000	7.7000	1.2000	-5.0000
505.0000	9.7000	1.2000	-5.0000
500.0000	10.1000	1.2000	-4.0000
493.0000	11.5000	1.2000	-2.0000
484.0000	13.1000	1.0000	-0.8000
481.0000	14.1000	0.8500	0.8000
481.0000	15.1000	0.8000	3.0000
482.0000	13.2000	0.7000	5.0000
475.0000	12.1000	0.7000	6.5000
466.0000	11.2000	0.5000	7.0000
474.0000	10.1000	0.5000	7.0000
483.0000	6.1000	0.5000	4.0000
483.0000	0.0000	0.0000	0.0000

SEC. #7

TEMP	VZ	VT	VR
300.0000	0.0000	0.0000	0.0000
314.0000	5.1000	0.0000	-0.8000
348.0000	5.2000	0.0000	-1.0000
420.0000	5.3000	0.0000	-1.5000
464.0000	5.5000	0.0000	-2.0000
462.0000	6.1000	0.0000	-4.0000
467.0000	9.5000	0.1000	-4.6000
472.0000	10.6000	0.1500	-4.0000
475.0000	12.1000	-0.2000	0.0000
477.0000	12.5000	-0.2000	4.0000
479.0000	12.5000	-0.2200	3.0000
477.0000	14.1000	-0.2000	1.8000
473.0000	14.2000	-0.2000	2.0000
463.0000	11.5000	-0.2000	2.2000
446.0000	11.1000	-0.2000	1.8000
423.0000	9.8000	-0.1800	-0.8000
446.0000	6.7000	-0.1500	-2.0000
434.0000	5.5000	-0.1000	-2.0000
403.0000	0.0000	0.0000	0.0000

SEC. #8

TEMP	VZ	VT	VR
342.0000	0.0000	0.0000	0.0000
377.0000	2.5000	0.7000	-0.2000
395.0000	3.7000	1.3000	-0.4000
406.0000	4.0000	1.2000	-1.0000
432.0000	5.1000	0.8000	-1.8000
457.0000	6.2000	0.5000	-3.4000
470.0000	7.5000	0.2500	-3.8000
479.0000	7.8000	0.2000	-4.0000
488.0000	8.8000	0.1000	-3.7000
490.0000	9.4000	0.0000	-3.6000
480.0000	13.1000	0.0000	-3.5000
480.0000	12.2000	0.0000	-3.2000
480.0000	10.1000	0.0000	-1.8000
478.0000	9.1000	0.2000	-1.0000
465.0000	8.9000	0.4000	0.0000
450.0000	7.8000	0.8000	0.0000

463.0000	6.7000	1.0000	0.8000
452.0000	6.6000	1.2000	1.0000
430.0000	5.8000	1.0000	2.0000
420.0000	0.0000	0.0000	0.0000

SEC. #9

TEMP	VZ	VT	VR
353.0000	0.0000	0.0000	0.0000
375.0000	2.2000	0.2000	-0.7000
385.0000	3.5000	0.3500	-1.0000
400.0000	3.7000	0.4000	-1.8000
415.0000	4.1000	0.4500	-2.2000
428.0000	5.6000	0.4500	-2.4000
444.0000	6.8000	0.3500	-3.0000
458.0000	8.6000	0.3000	-3.0000
464.0000	9.2000	0.3000	-3.0000
470.0000	12.0000	0.3000	-2.4000
476.0000	11.5000	0.3500	-2.4000
476.0000	10.7000	0.3500	-2.0000
474.0000	9.8000	0.4000	-1.8000
473.0000	8.3000	0.5000	-1.0000
465.0000	7.0000	0.6000	-0.8000
457.0000	6.8000	0.6500	-0.4000
455.0000	6.4000	0.5500	-0.6000
452.0000	6.1000	0.4500	-1.0000
446.0000	5.6000	0.2000	-1.0000
440.0000	0.0000	0.0000	0.0000

SEC. #10

TEMP	VZ	VT	VR
340.0000	0.0000	0.0000	0.0000
350.0000	18.1000	0.2000	2.8000
371.0000	19.6000	0.2000	2.6000
396.0000	23.8000	0.2500	2.0000
410.0000	23.9000	0.3000	0.0000
414.0000	24.4000	0.3000	-3.0000
412.0000	27.5000	0.5000	-4.0000
403.0000	27.8000	1.0000	-2.0000
391.0000	27.7000	1.2000	-2.0000
351.0000	25.8000	1.0000	1.0000
340.0000	0.0000	0.0000	0.0000

RUN # 4

13

SEC. #1

TEMP	VZ	VT	VR
1295.0000	0.0000	0.0000	0.0000
1300.0000	8.7000	-5.0000	4.0000
1274.0000	11.5000	-2.1000	2.0000
1146.0000	-4.5000	-0.7000	-1.8000
1092.0000	-7.6000	0.0000	-1.8000
1072.0000	-8.1000	0.0000	-1.8000
1088.0000	-8.0000	0.0000	-0.4000
1100.0000	-7.6000	0.0000	3.2000
1141.0000	-6.2000	0.0000	4.0000
1157.0000	-4.5000	0.0000	1.0000
1166.0000	3.5000	-0.5000	-1.0000
1162.0000	12.0000	-0.7000	-3.0000
1262.0000	13.5000	-5.5000	-8.0000
1313.0000	11.2000	-3.5000	-5.0000
1332.0000	7.6000	-4.1000	-2.0000
1329.0000	0.0000	0.0000	0.0000

SEC. #2

TEMP	VZ	VT	VR
1048.0000	0.0000	0.0000	0.0000
1080.0000	5.1000	-1.2500	3.0000
1166.0000	10.6000	-0.5000	3.0000
1021.0000	13.5000	0.5000	3.0000
975.0000	18.5000	0.5000	3.0000
870.0000	-3.5000	-1.0000	-2.0000
788.0000	-7.0000	-1.0000	-2.0000
794.0000	-6.1000	0.0000	-2.5000
838.0000	-7.5000	-1.0000	-2.1000
870.0000	-5.5000	-1.0000	-2.0000
880.0000	-3.5000	0.0000	-1.0000
886.0000	6.2000	1.5000	-2.0000
913.0000	8.4000	0.0000	-2.5000
1001.0000	10.1000	-0.5000	-3.0000
968.0000	8.7000	-1.5000	3.0000
970.0000	0.0000	0.0000	0.0000

SEC. #3

TEMP	VZ	VT	VR
700.0000	0.0000	0.0000	0.0000
710.0000	13.0000	-1.5000	2.0000
670.0000	15.5000	-2.0000	2.0000
602.0000	-6.5000	0.5000	1.5000
526.0000	-9.6000	2.0000	1.5000
419.0000	-12.5000	1.7000	3.0000
350.0000	-4.1000	1.5000	0.0000
331.0000	28.0000	-0.5000	-1.5000
361.0000	30.1000	-3.5000	-5.0000
414.0000	29.2000	-3.5000	-2.0000
505.0000	29.1000	-2.5000	-3.5000

578.0000	25.2000	-3.5000	-2.0000
632.0000	16.1000	-3.2000	0.0000
686.0000	-6.1000	2.1000	0.0000
717.0000	3.2000	0.0000	0.0000
711.0000	7.5000	-0.5000	-2.5000
690.0000	0.0000	0.0000	0.0000

SEC. #4

TEMP	VZ	VT	VR
87.0000	0.0000	0.0000	0.0000
700.0000	1.5000	0.2000	-1.5000
648.0000	3.5000	0.3000	-2.5000
581.0000	6.5000	0.4000	-3.0000
513.0000	8.1000	0.6000	-5.0000
455.0000	9.6000	0.6000	-4.5000
416.0000	12.2000	0.7000	-3.5000
402.0000	13.1000	0.5000	0.0000
402.0000	13.2000	0.1000	3.0000
430.0000	20.1000	-0.1000	5.5000
453.0000	21.5000	-0.2000	8.5000
476.0000	15.5000	-0.1000	9.5000
500.0000	6.1000	0.0000	8.5000
577.0000	5.1000	0.0000	7.0000
633.0000	4.6000	0.0000	5.0000
639.0000	1.1000	0.0000	3.0000
600.0000	0.0000	0.0000	0.0000

SEC. #5

TEMP	VZ	VT	VR
567.0000	0.0000	0.0000	0.0000
611.0000	1.5000	0.3000	-1.0000
584.0000	2.8000	0.4000	-1.5000
542.0000	4.1000	0.5000	-3.0000
490.0000	6.9000	0.6000	-3.0000
465.0000	8.8000	0.7000	-3.5000
435.0000	10.1000	0.6000	-2.5000
425.0000	11.5000	0.5000	-2.0000
422.0000	12.2000	0.4000	0.8000
437.0000	14.2000	0.4000	3.0000
445.0000	12.8000	0.3000	5.5000
449.0000	10.8000	0.3000	6.2000
450.0000	9.9000	0.3000	7.4000
453.0000	8.8000	0.3000	6.0000
570.0000	4.1000	0.5000	5.8000
543.0000	2.1000	0.3000	4.0000
514.0000	0.0000	0.0000	0.0000

SEC. #6

TEMP	VZ	VT	VR
365.0000	0.0000	0.0000	0.0000
419.0000	2.0000	0.0000	-1.0000
430.0000	4.1000	0.0000	-1.2000
451.0000	5.5000	0.3000	-1.5000
475.0000	6.1000	0.6000	-2.0000

485.0000	8.5000	0.7000	-2.0000
483.0000	9.1000	0.6000	-2.0000
476.0000	9.8000	0.5000	-1.5000
467.0000	11.2000	0.4000	-1.0000
457.0000	12.2000	0.3000	0.0000
456.0000	13.1000	0.2000	1.5000
447.0000	12.0000	-0.2000	2.0000
425.0000	10.8000	-0.4000	2.0000
402.0000	9.0000	-0.2000	2.0000
400.0000	6.2000	-0.2000	2.0000
403.0000	5.6000	-0.1000	2.0000
378.0000	3.1000	0.0000	2.0000
340.0000	0.0000	0.0000	0.0000

SEC. #7

TEMP	VZ	VT	VR
305.0000	0.0000	0.0000	0.0000
317.0000	2.7000	0.1000	-1.5000
395.0000	4.8000	0.2000	-4.0000
457.0000	5.1000	0.2000	-4.0000
472.0000	8.2000	0.4000	-3.0000
474.0000	10.2000	0.6000	-2.5000
476.0000	12.1000	0.6000	-2.5000
473.0000	12.2000	0.6000	-1.5000
470.0000	12.3000	0.6000	-1.5000
466.0000	11.4000	0.6000	0.0000
450.0000	11.3000	0.6000	2.0000
438.0000	11.2000	0.5000	1.5000
413.0000	11.1000	0.5000	1.0000
365.0000	11.0000	0.5000	0.5000
350.0000	9.1000	0.6000	1.0000
357.0000	6.5000	1.0000	6.0000
330.0000	5.4000	1.0000	7.0000
300.0000	2.3000	0.8000	8.0000
310.0000	0.0000	0.0000	0.0000

SEC. #8

TEMP	VZ	VT	VR
313.0000	0.0000	0.0000	0.0000
347.0000	1.1000	0.3000	-2.0000
358.0000	2.1000	0.4000	-2.0000
377.0000	3.2000	0.6000	-2.0000
406.0000	3.8000	0.6000	-2.0000
433.0000	5.1000	0.5000	-2.5000
458.0000	5.2000	0.4000	-2.0000
468.0000	7.5000	0.4000	-2.0000
473.0000	8.6000	0.4000	-1.5000
474.0000	11.2000	0.4000	-1.5000
472.0000	13.4000	0.4000	-1.0000
469.0000	14.2000	0.4000	0.0000
459.0000	13.5000	0.4000	0.0000
449.0000	12.6000	0.3500	1.0000
433.0000	11.1000	0.3000	2.0000
407.0000	8.1000	0.4000	2.5000

395.0000	7.8000	0.4000	2.5000
382.0000	5.2000	0.4000	3.0000
355.0000	4.4000	0.4000	2.0000
333.0000	0.0000	0.0000	0.0000

SEC. #9

TEMP	VZ	VT	VR
392.0000	0.0000	0.0000	0.0000
410.0000	1.6000	0.4000	-1.5000
424.0000	2.1000	0.6000	-1.5000
439.0000	3.1000	0.6000	-2.0000
462.0000	4.6000	0.5000	-2.0000
482.0000	6.5000	0.5000	-1.8000
494.0000	8.2000	0.5000	-1.5000
500.0000	9.2000	0.4000	-1.0000
515.0000	9.5000	0.3500	-1.0000
520.0000	10.6000	0.2500	-0.8000
513.0000	11.1000	0.2000	0.0000
509.0000	11.2000	0.2000	0.5000
504.0000	11.4000	0.1500	0.5000
493.0000	15.1000	0.0000	0.8000
474.0000	15.3000	0.0000	1.5000
460.0000	13.1000	0.0000	2.5000
441.0000	10.1000	0.2000	3.0000
412.0000	7.8000	0.4000	3.5000
400.0000	2.2000	0.3500	2.5000
396.0000	0.0000	0.0000	0.0000

SEC. #10

TEMP	VZ	VT	VR
430.0000	0.0000	0.0000	0.0000
457.0000	20.2000	1.9000	6.0000
472.0000	26.1000	2.0000	5.0000
475.0000	27.1000	2.0000	2.5000
483.0000	30.5000	2.0000	1.0000
484.0000	30.1000	2.1000	0.2000
477.0000	30.8000	2.0000	0.2000
454.0000	30.4000	2.0000	0.2000
435.0000	27.1000	2.0000	0.2000
377.0000	22.5000	1.9500	6.0000
355.0000	0.0000	0.0000	0.0000

RUN # 5

SEC. #1			
TEMP	VZ	VT	VR
1097.0000	0.0000	0.0000	0.0000
1132.0000	2.5000	-0.2000	2.5000
1224.0000	4.3000	-0.7000	4.5000
1283.0000	3.3000	-0.9000	0.7000
1242.0000	-1.1000	-0.5000	0.3000
1247.0000	-2.1000	-0.1500	1.0000
1259.0000	-2.5000	-0.1000	1.1000
1259.0000	-2.4000	0.0000	1.1000
1231.0000	-2.1000	0.2000	0.0000
1201.0000	-0.9000	0.8000	0.0000
1172.0000	3.1000	1.0000	0.0000
1134.0000	5.5000	0.4000	3.0000
1120.0000	6.5000	-1.2000	5.0000
1086.0000	6.0000	-0.5000	1.0000
1155.0000	4.0000	0.0000	1.0000
1170.0000	0.0000	0.0000	0.0000

SEC. #2			
TEMP	VZ	VT	VR
822.0000	0.0000	0.0000	0.0000
920.0000	6.6000	0.1000	0.7000
950.0000	7.5000	0.1000	1.5000
920.0000	4.2000	0.1000	2.0000
822.0000	-1.2000	0.0000	0.0000
730.0000	-3.5000	-0.1000	-2.5000
685.0000	-4.1000	-0.9000	-1.0000
690.0000	-5.1000	-0.1000	0.0000
711.0000	-5.6000	0.0000	1.5000
760.0000	-2.5000	0.0000	3.0000
815.0000	3.1000	0.0000	0.0000
860.0000	5.1000	0.1000	0.0000
945.0000	6.6000	0.2000	-1.5000
982.0000	10.2000	1.0000	-1.3000
1019.0000	8.2000	0.7500	-1.0000
1045.0000	0.0000	0.0000	0.0000

SEC. #3			
TEMP	VZ	VT	VR
626.0000	0.0000	0.0000	0.0000
662.0000	-2.1000	0.0000	0.0000
630.0000	-6.1000	0.0000	0.0000
570.0000	-5.6000	0.0000	0.0000
490.0000	-1.0000	0.1000	0.0000
400.0000	8.5000	0.6500	-0.7000
348.0000	10.8000	0.7500	-5.0000
328.0000	22.2000	0.6000	0.0000
335.0000	20.1000	0.3000	5.0000
393.0000	19.2000	0.0500	8.0000
472.0000	18.8000	-0.0500	7.0000

564.0000	17.4000	-0.2000	3.5000
665.0000	12.2000	-0.2000	2.5000
735.0000	5.5000	0.0000	1.5000
760.0000	-2.1000	0.0000	0.5000
770.0000	-2.1000	0.0000	0.0000
755.0000	0.0000	0.0000	0.0000

SEC. #4

TEMP	VZ	VT	VR
413.0000	0.0000	0.0000	0.0000
620.0000	1.0000	0.0000	-1.5000
608.0000	1.5000	0.1500	-1.5000
562.0000	2.0000	-0.2000	-1.0000
475.0000	3.6000	-0.2500	0.0000
433.0000	5.8000	-0.1000	1.0000
404.0000	9.1000	-0.1000	1.5000
386.0000	10.2000	-0.1000	1.3000
387.0000	11.1000	-0.2000	0.0000
404.0000	12.1000	-0.6000	0.0000
448.0000	17.5000	-1.4000	0.5000
481.0000	16.2000	-0.7000	1.2000
532.0000	10.1000	-0.7000	1.4000
593.0000	6.4000	-0.3000	1.7000
689.0000	5.8000	-0.2000	2.0000
705.0000	3.5000	-0.2000	1.8000
688.0000	2.5000	0.0000	0.0000

SEC. #5

TEMP	VZ	VT	VR
415.0000	0.0000	0.0000	0.0000
619.0000	2.6000	-0.1000	-1.0000
607.0000	2.6000	-0.1500	-1.5000
561.0000	2.6000	-0.1500	-1.7000
517.0000	2.6000	-0.1500	-1.8000
480.0000	3.5000	-0.1000	-2.0000
450.0000	5.6000	-0.2000	-1.5000
436.0000	8.5000	-0.2500	0.0000
433.0000	9.6000	-0.5000	1.0000
443.0000	10.1000	-0.6000	1.7000
468.0000	8.6000	-0.6000	3.7000
497.0000	7.6000	-0.6000	5.0000
532.0000	5.7000	-0.6000	5.5000
570.0000	3.2000	-0.6000	5.5000
651.0000	2.5000	-0.5000	4.2000
665.0000	1.6000	-0.3000	3.5000
644.0000	0.0000	0.0000	0.0000

SEC. #6

TEMP	VZ	VT	VR
377.0000	0.0000	0.0000	0.0000
440.0000	1.2000	0.0000	1.0000
418.0000	2.1000	-0.1000	1.5000
517.0000	3.8000	-0.2000	1.7000
462.0000	3.5000	-0.3000	1.5000

500.0000	5.2000	0.4000	1.3000
503.0000	6.1000	0.4000	0.0000
500.0000	7.6000	0.3000	0.0000
496.0000	7.6000	0.2000	0.0000
494.0000	7.6000	0.1000	1.0000
488.0000	8.1000	0.0000	2.5000
485.0000	8.6000	0.0000	2.9000
470.0000	8.6000	0.0000	3.2000
427.0000	8.6000	-0.0500	3.5000
420.0000	6.6000	-0.1000	2.2000
399.0000	6.1000	-0.2000	2.0000
390.0000	1.1000	-0.5000	1.5000
405.0000	0.0000	0.0000	0.0000

SEC. #7

TEMP	VZ	VT	VR
280.0000	0.0000	0.0000	0.0000
275.0000	1.9000	0.1500	0.5000
317.0000	2.9000	0.4000	1.0000
430.0000	3.9000	0.2000	1.5000
474.0000	4.5000	0.1500	2.5000
479.0000	6.7000	0.1000	2.0000
485.0000	9.6000	0.0000	0.5000
490.0000	9.7000	0.1000	0.8000
494.0000	9.8000	0.1500	1.0000
496.0000	9.9000	0.1500	1.5000
456.0000	8.9000	0.1500	2.4000
450.0000	8.8000	0.1500	3.0000
430.0000	6.5000	0.2000	3.5000
407.0000	5.4000	0.1500	3.5000
378.0000	5.0000	0.1500	-1.0000
365.0000	3.5000	0.2000	-3.5000
314.0000	2.8000	0.5000	-4.5000
269.0000	1.9000	0.2000	-4.0000
269.0000	0.0000	0.0000	0.0000

SEC. #8

TEMP	VZ	VT	VR
335.0000	0.0000	0.0000	0.0000
385.0000	1.7000	0.0000	1.5000
397.0000	2.3000	0.0000	2.0000
384.0000	2.3000	0.0000	2.7000
383.0000	4.4000	0.0000	3.0000
415.0000	7.5000	-0.1000	3.0000
450.0000	7.6000	-0.1000	2.0000
466.0000	9.7000	-0.1000	1.0000
468.0000	9.5000	-0.1000	1.0000
465.0000	9.4000	-0.0500	1.0000
458.0000	9.3000	-0.0500	1.0000
452.0000	9.2000	-0.0100	0.5000
438.0000	7.2000	0.0000	0.0000
413.0000	6.1000	0.1000	-0.5000
380.0000	4.0000	0.2000	-1.0000
352.0000	2.8000	0.3000	-1.1000

348.0000	1.1000	0.4000	-1.0000
343.0000	0.8000	0.3000	0.0000
342.0000	0.5000	0.2000	1.0000
342.0000	0.0000	0.0000	0.0000

SEC. #9

TEMP.	VZ	VT	VR
328.0000	0.0000	0.0000	0.0000
315.0000	1.0000	0.2000	0.4000
352.0000	1.5000	0.3000	0.5000
359.0000	1.6000	0.2500	2.0000
365.0000	2.5000	0.2000	2.0000
385.0000	4.5000	0.1500	2.2000
403.0000	6.8000	0.1000	2.0000
417.0000	8.1000	0.1000	2.0000
420.0000	8.5000	0.1500	2.2000
418.0000	8.6000	0.1500	2.0000
416.0000	8.8000	0.1500	1.0000
408.0000	8.8000	0.1500	0.5000
390.0000	8.8000	0.1000	1.0000
370.0000	8.4000	0.0000	0.5000
349.0000	8.3000	0.0000	0.5000
335.0000	5.1000	-0.1000	0.7000
342.0000	4.9000	-0.2000	1.2000
336.0000	2.5000	-0.3000	1.5000
338.0000	1.0000	-0.3000	2.0000
330.0000	0.0000	0.0000	0.0000

SEC. #10

TEMP	VZ	VT	VR
367.0000	0.0000	0.0000	0.0000
392.0000	12.9000	0.1000	3.0000
414.0000	14.1000	0.2000	4.5000
425.0000	17.2000	0.2500	4.7000
419.0000	18.6000	0.5000	4.0000
406.0000	18.6000	0.4000	2.5000
386.0000	18.1000	-0.1000	1.0000
361.0000	17.8000	-1.0000	0.8000
334.0000	17.1000	-1.2000	0.7000
310.0000	13.1000	-0.2000	6.0000
291.0000	0.0000	0.0000	0.0000

RUN # 6

SEC. #1			
TEMP	VZ	VT	VR
842.0000	0.0000	0.0000	0.0000
1042.0000	5.0000	0.0000	-2.1000
893.0000	7.5000	0.3000	-2.5000
730.0000	2.0000	1.0000	-2.6000
671.0000	-5.6000	1.0000	-1.6000
645.0000	-5.6000	-0.1500	0.0000
635.0000	-4.1000	-0.1500	1.5000
634.0000	-3.5000	0.0000	2.0000
645.0000	-3.1000	0.0000	2.0000
660.0000	-3.1000	-0.1500	2.5000
671.0000	7.1000	0.0000	2.5000
686.0000	8.0000	1.0000	0.0000
733.0000	10.0000	0.9500	-2.0000
810.0000	8.0000	0.5000	-2.5000
848.0000	5.0000	0.2000	-2.0000
866.0000	0.0000	0.0000	0.0000

SEC. #2			
TEMP	VZ	VT	VR
705.0000	0.0000	0.0000	0.0000
865.0000	8.6000	0.0000	2.0000
910.0000	9.6000	0.2000	2.0000
889.0000	10.6000	0.7500	2.0000
820.0000	9.6000	0.8000	1.5000
685.0000	8.8000	0.0000	0.0000
600.0000	-2.1000	0.7000	0.0000
590.0000	-3.0000	0.7000	3.5000
615.0000	-5.5000	0.3000	4.5000
660.0000	-4.5000	0.6000	2.0000
702.0000	-3.5000	0.3500	-1.7000
787.0000	-2.5000	0.2000	-2.0000
858.0000	9.5000	0.1500	-2.0000
910.0000	12.0000	0.1500	-2.0000
963.0000	10.0000	0.1500	-1.5000
980.0000	0.0000	0.0000	0.0000

SEC. #3			
TEMP	VZ	VT	VR
479.0000	0.0000	0.0000	0.0000
570.0000	9.0000	0.1500	0.0000
562.0000	14.0000	0.2000	0.0000
506.0000	10.5000	0.2500	0.0000
447.0000	-6.1000	0.4000	0.0000
368.0000	-5.6000	1.5000	1.5000
311.0000	10.2000	1.0000	3.5000
285.0000	24.2000	1.4000	0.0000
284.0000	24.5000	1.7000	2.5000
313.0000	23.1000	1.5000	5.0000
372.0000	-8.0000	1.3000	6.0000

440.0000	-8.1000	1.0000	10.0000
522.0000	-6.2000	2.1000	-2.0000
600.0000	7.5000	0.5000	-1.0000
637.0000	12.0000	0.3000	0.0000
670.0000	7.5000	0.2000	0.0000
650.0000	0.0000	0.0000	0.0000

SEC. #4

TEMP	VZ	VT	VR
498.0000	0.0000	0.0000	0.0000
571.0000	2.7000	0.2500	-1.0000
552.0000	2.8000	0.8000	-2.0000
501.0000	2.9000	1.5000	-3.0000
450.0000	3.1000	1.7000	-3.5000
395.0000	3.5000	2.0000	-4.5000
358.0000	4.1000	1.7000	-4.0000
336.0000	6.5000	1.1000	-2.2000
331.0000	8.1000	0.8000	0.0000
348.0000	10.5000	0.0000	1.5000
381.0000	10.7000	-0.2000	4.0000
423.0000	11.1000	-0.3000	6.0000
475.0000	10.8000	-0.3500	6.5000
560.0000	9.8000	-0.6000	6.5000
601.0000	8.5000	-0.7000	6.5000
639.0000	7.5000	-0.5000	2.0000
645.0000	0.0000	0.0000	0.0000

SEC. #5

TEMP	VZ	VT	VR
445.0000	0.0000	0.0000	0.0000
545.0000	2.5000	-0.0500	-1.0000
535.0000	2.5000	-0.2000	-1.5000
505.0000	2.5000	-0.2000	-2.0000
462.0000	3.5000	0.1500	-2.5000
425.0000	7.1000	-0.1000	-3.0000
400.0000	8.1000	-0.0500	-3.0000
382.0000	9.6000	-0.0500	-1.5000
380.0000	9.8000	-0.0500	0.0000
382.0000	10.2000	-0.0500	1.0000
396.0000	10.1000	-0.1000	4.5000
420.0000	9.8000	-0.2000	5.0000
457.0000	9.2000	-0.3000	5.5000
472.0000	8.4000	-0.5000	6.0000
540.0000	7.5000	-0.5000	6.0000
570.0000	6.5000	-0.4000	4.5000
575.0000	0.0000	0.0000	0.0000

SEC. #6

TEMP	VZ	VT	VR
359.0000	0.0000	0.0000	0.0000
411.0000	1.6000	0.3000	-1.0000
417.0000	1.6000	0.3500	-1.0000
423.0000	1.7000	0.3000	-1.0000
454.0000	2.0000	0.2000	-1.0000

491.0000	4.1000	0.1000	-1.0000
503.0000	7.0000	0.0000	-1.5000
504.0000	7.8000	-0.1000	-1.5000
499.0000	8.5000	-0.1000	-1.0000
501.0000	9.1000	-0.1500	0.0000
500.0000	9.6000	-0.1500	1.0000
502.0000	9.6000	-0.1500	2.0000
492.0000	10.1000	-0.2000	3.0000
460.0000	9.4000	-0.2000	2.5000
453.0000	9.3000	-0.2000	1.5000
426.0000	8.2000	-0.2000	1.0000
420.0000	6.2000	-0.2000	0.0000
442.0000	0.0000	0.0000	0.0000

SEC. #7

TEMP	VZ	UT	VR
290.0000	0.0000	0.0000	0.0000
296.0000	6.5000	0.0000	1.0000
313.0000	6.0000	-0.1000	1.0000
394.0000	6.0000	-0.1500	1.0000
438.0000	6.1000	-0.1500	1.0000
441.0000	6.1000	-0.2000	1.5000
444.0000	6.1000	-0.2000	-1.5000
446.0000	6.2000	-0.2000	-1.5000
448.0000	6.2000	-0.2000	-1.5000
446.0000	6.4000	-0.2000	-1.5000
443.0000	6.4000	-0.2000	-1.0000
433.0000	7.0000	-0.1500	-0.5000
422.0000	7.0000	-0.1500	0.0000
412.0000	7.5000	-0.1500	0.5000
383.0000	7.5000	-0.1500	1.5000
371.0000	8.0000	-0.1500	1.5000
340.0000	7.1000	-0.1500	-1.5000
307.0000	7.2000	-0.1500	-2.0000
291.0000	0.0000	0.0000	0.0000

SEC. #8

TEMP	VZ	UT	VR
319.0000	0.0000	0.0000	0.0000
352.0000	3.0000	-0.1000	2.0000
355.0000	4.7000	-0.2500	3.5000
350.0000	4.8000	-0.4000	4.5000
365.0000	5.9000	-0.3500	5.0000
391.0000	6.1000	-0.2000	4.5000
420.0000	6.2000	-0.2500	2.0000
429.0000	7.2000	-0.2500	-1.5000
431.0000	7.5000	-0.2500	-1.5000
428.0000	7.5000	-0.2500	-1.0000
425.0000	7.6000	-0.2500	0.0000
420.0000	7.8000	-0.2500	0.0000
408.0000	8.1000	-0.2500	-1.0000
400.0000	8.0000	-0.2500	-3.5000
370.0000	7.5000	-0.2500	-4.7000
340.0000	6.5000	-0.2500	-3.7000

334.0000	5.5000	-0.3000	-1.0000
335.0000	4.0000	-0.4000	3.0000
340.0000	3.1000	-0.3000	3.5000
344.0000	0.0000	0.0000	0.0000

SEC. #9

TEMP	UZ	UT	VR
326.0000	0.0000	0.0000	0.0000
347.0000	3.6000	0.5000	0.5000
353.0000	3.7000	0.6000	0.5000
358.0000	3.8000	0.5500	0.7000
367.0000	5.9000	-0.4000	1.0000
377.0000	6.9000	0.2500	2.5000
392.0000	7.5000	0.3000	4.0000
410.0000	7.5000	0.4500	4.5000
410.0000	7.5000	0.5000	2.0000
409.0000	8.2000	0.5000	0.0000
408.0000	9.5000	0.5000	0.0000
402.0000	9.5000	0.5500	0.0000
386.0000	9.2000	0.6000	-1.0000
372.0000	6.1000	0.6500	-2.0000
358.0000	6.0000	0.7000	-2.0000
345.0000	6.0000	0.7500	-1.0000
342.0000	6.0000	0.7500	0.0000
343.0000	5.0000	0.0000	1.5000
342.0000	3.5000	0.0000	3.0000
342.0000	0.0000	0.0000	0.0000

SEC. #10

TEMP	UZ	UT	VR
351.0000	0.0000	0.0000	0.0000
388.0000	21.7000	0.4000	1.0000
405.0000	23.3000	0.4000	3.0000
415.0000	26.3000	0.5000	2.0000
425.0000	27.9000	0.7000	1.5000
419.0000	29.9000	0.8500	0.0000
406.0000	27.7000	0.7000	0.0000
386.0000	25.8000	0.5000	0.0000
361.0000	23.6000	0.5000	1.0000
334.0000	21.7000	0.6000	3.0000
291.0000	0.0000	0.0000	0.0000

RUN # 7

SEC. #1			
TEMP	VZ	VT	VR
793.0000	0.0000	0.0000	0.0000
994.0000	4.7000	-0.1500	3.0000
856.0000	6.6000	-0.6000	1.5000
885.0000	4.5000	-1.2000	-1.5000
841.0000	-2.6000	-1.4000	-2.0000
802.0000	-3.1000	-1.5000	0.0000
780.0000	-2.5000	-0.5000	0.0000
765.0000	-2.5000	-0.4000	0.0000
755.0000	-2.5000	-0.3500	0.0000
775.0000	-1.7000	-0.2000	0.0000
789.0000	2.7000	-0.2000	0.0000
818.0000	3.5000	-0.5000	-2.0000
868.0000	5.0000	-0.8000	2.0000
858.0000	6.0000	-1.2000	3.0000
873.0000	3.8000	-1.0000	1.0000
895.0000	0.0000	0.0000	0.0000

SEC. #2			
TEMP	VZ	VT	VR
702.0000	0.0000	0.0000	0.0000
902.0000	6.0000	1.2000	1.5000
912.0000	6.0000	0.8500	3.0000
881.0000	6.0000	0.6000	1.5000
803.0000	5.5000	0.3000	0.0000
661.0000	3.5000	0.1500	0.0000
550.0000	-0.9000	0.1500	0.0000
530.0000	-6.0000	0.1500	1.0000
556.0000	-5.5000	0.1500	1.5000
588.0000	-4.5000	0.1500	1.0000
630.0000	1.5000	0.0000	0.0000
701.0000	6.0000	-0.3000	0.0000
774.0000	7.1000	-0.7000	3.5000
785.0000	8.0000	-1.0000	2.0000
830.0000	7.5000	-1.2000	1.5000
851.0000	0.0000	0.0000	0.0000

SEC. #3			
TEMP	VZ	VT	VR
448.0000	0.0000	0.0000	0.0000
516.0000	8.2000	0.4000	-1.0000
515.0000	9.4000	0.4500	-1.0000
490.0000	-3.0000	0.5500	-1.5000
444.0000	-3.5000	0.9000	-2.0000
380.0000	12.0000	1.8000	-6.0000
324.0000	16.0000	1.2000	-5.0000
284.0000	13.0000	1.0000	0.0000
270.0000	16.0000	0.5000	5.0000
281.0000	14.0000	-0.2000	7.0000
323.0000	14.0000	-0.2500	8.0000

390.0000	-4.0000	-0.2500	8.0000
466.0000	-3.5000	-0.2500	6.5000
545.0000	9.0000	-0.3000	4.0000
582.0000	8.0000	-1.5000	2.5000
613.0000	7.0000	-1.5000	2.0000
615.0000	0.0000	0.0000	0.0000

SEC. #4

TEMP	VZ	VT	VR
357.0000	0.0000	0.0000	0.0000
462.0000	2.5000	-0.1000	-1.0000
482.0000	2.5000	-0.2000	-1.0000
453.0000	4.0000	-0.2000	-2.0000
430.0000	5.0000	-0.2000	-4.0000
400.0000	7.0000	-0.2000	-5.0000
360.0000	7.8000	-0.1000	-4.5000
330.0000	10.2000	0.0000	-2.5000
320.0000	11.1000	-0.1000	0.0000
329.0000	11.5000	-0.8000	3.0000
350.0000	10.2000	-1.5000	5.0000
386.0000	7.6000	-0.6000	6.5000
409.0000	6.8000	-0.6000	7.0000
461.0000	5.5000	-0.8000	5.0000
517.0000	4.0000	-1.0000	3.5000
555.0000	2.0000	-0.8000	3.0000
556.0000	0.0000	0.0000	0.0000

SEC. #5

TEMP	VZ	VT	VR
342.0000	0.0000	0.0000	0.0000
440.0000	2.8000	-0.3000	-1.0000
455.0000	3.9000	-0.4000	-2.0000
440.0000	3.9000	-0.4000	-2.5000
433.0000	5.0000	-0.5000	-2.7000
403.0000	5.5000	-0.3000	-3.0000
387.0000	7.5000	-0.2000	-2.5000
368.0000	11.1000	0.0000	-1.5000
356.0000	11.2000	0.0000	0.0000
352.0000	11.1000	0.0000	1.5000
373.0000	8.8000	0.0000	4.5000
388.0000	8.0000	-0.1000	5.0000
410.0000	7.0000	-0.2000	5.5000
411.0000	6.6000	-0.4000	6.0000
442.0000	6.4000	-0.5000	4.7000
448.0000	2.6000	-0.5000	4.5000
453.0000	0.0000	0.0000	0.0000

SEC. #6

TEMP	VZ	VT	VR
313.0000	0.0000	0.0000	0.0000
373.0000	3.4000	0.4000	-1.5000
376.0000	4.5000	0.8000	-1.5000
377.0000	4.5000	1.0000	-1.0000
394.0000	4.6000	0.7500	-1.0000

426.0000	5.1000	0.4000	-0.5000
445.0000	5.5000	0.3000	-2.0000
453.0000	6.0000	0.2000	-2.0000
456.0000	8.5000	0.1000	-2.0000
456.0000	9.1000	0.0000	0.0000
452.0000	7.2000	-0.1000	0.5000
449.0000	9.1000	-0.2000	1.0000
434.0000	9.2000	-0.2500	1.5000
411.0000	9.2000	-0.2500	2.0000
390.0000	8.0000	-0.4000	2.0000
372.0000	8.4000	-0.8000	2.0000
360.0000	6.8000	-1.2000	0.0000
364.0000	0.0000	0.0000	0.0000

SEC. #7

TEMP	VZ	UT	VR
261.0000	0.0000	0.0000	0.0000
274.0000	5.3000	0.2500	1.5000
275.0000	6.2000	0.2500	2.0000
319.0000	5.0000	0.2000	2.5000
385.0000	4.0000	0.1000	2.5000
400.0000	3.5000	0.0000	2.5000
404.0000	4.0000	0.0000	-2.0000
407.0000	4.1000	0.0000	-2.0000
410.0000	4.5000	0.1000	-1.5000
412.0000	5.0000	0.1500	-1.0000
411.0000	5.5000	0.1000	-1.0000
407.0000	6.0000	0.0000	0.0000
400.0000	6.0000	-0.1000	0.0000
390.0000	6.0000	-0.1000	0.0000
365.0000	8.0000	-0.1000	-1.0000
345.0000	8.5000	0.0000	-3.0000
317.0000	9.5000	-0.1000	-4.0000
289.0000	8.1000	-0.1000	0.0000
268.0000	0.0000	0.0000	0.0000

SEC. #8

TEMP	VZ	UT	VR
302.0000	0.0000	0.0000	0.0000
327.0000	2.1000	0.1500	1.0000
327.0000	3.6000	0.1500	2.0000
325.0000	5.5000	0.2000	3.0000
340.0000	8.0000	0.2000	4.5000
364.0000	4.5000	0.2000	5.5000
389.0000	3.0000	0.2000	4.5000
400.0000	4.1000	0.2000	2.5000
403.0000	5.1000	0.2000	-1.0000
403.0000	5.2000	0.1000	-1.0000
401.0000	5.5000	0.0000	-1.5000
398.0000	6.1000	0.0000	-1.0000
385.0000	6.5000	0.0000	-0.5000
369.0000	6.5000	0.0000	-1.0000
341.0000	7.0000	0.1000	-3.0000
314.0000	7.1000	0.2000	-3.0000

308.0000	7.5000	0.2500	-2.5000
309.0000	6.0000	0.0000	-1.0000
312.0000	5.0000	0.2000	0.0000
315.0000	0.0000	0.0000	0.0000

SEC. #9

TEMP	VZ	VT	VR
337.0000	0.0000	0.0000	0.0000
343.0000	3.1000	0.2500	0.5000
351.0000	4.6000	0.2500	0.5000
362.0000	5.5000	0.2500	0.5000
375.0000	5.0000	0.2500	0.5000
386.0000	6.0000	0.2500	2.5000
392.0000	6.5000	0.2500	4.0000
392.0000	6.0000	0.2500	3.5000
388.0000	5.5000	0.3000	2.0000
379.0000	5.0000	0.3000	-1.0000
369.0000	5.5000	0.2500	-1.0000
350.0000	6.0000	0.2500	-1.5000
334.0000	7.0000	0.2500	-2.0000
328.0000	7.0000	0.3500	-2.5000
324.0000	7.0000	0.3500	-2.0000
324.0000	6.9000	0.3000	-1.0000
320.0000	6.7000	0.2500	0.0000
317.0000	4.0000	0.2000	2.5000
315.0000	3.1000	0.1500	2.0000
314.0000	0.0000	0.0000	0.0000

SEC. #10

TEMP	VZ	VT	VR
339.0000	0.0000	0.0000	0.0000
357.0000	18.5000	0.3000	1.5000
369.0000	21.2000	0.7500	4.0000
374.0000	24.0000	0.8000	3.8000
373.0000	24.5000	0.6500	2.0000
373.0000	26.0000	0.5000	1.0000
367.0000	26.1000	0.4000	0.0000
360.0000	26.0000	0.2000	-1.0000
344.0000	24.7000	0.2000	1.0000
320.0000	23.0000	0.4000	4.0000
322.0000	0.0000	0.0000	0.0000

RUN # 8

SEC. #1			
TEMP	VZ	VT	VR
862.0000	0.0000	0.0000	0.0000
1025.0000	1.7000	-0.1000	-0.5000
952.0000	4.0000	-0.5000	1.0000
972.0000	5.0000	-1.0000	2.5000
1003.0000	4.5000	-1.0000	3.5000
1003.0000	2.5000	0.0000	1.5000
1025.0000	-1.5000	0.5000	0.0000
1081.0000	-2.0000	1.0000	0.0000
1137.0000	-2.1000	1.5000	0.0000
1137.0000	-1.1000	0.8000	0.0000
1112.0000	1.1000	0.0000	0.0000
1158.0000	6.0000	1.2000	1.0000
1275.0000	8.1000	1.3000	1.0000
1295.0000	7.9000	0.7000	1.0000
1331.0000	4.2000	0.1500	1.0000
1340.0000	0.0000	0.0000	0.0000

SEC. #2			
TEMP	VZ	VT	VR
765.0000	0.0000	0.0000	0.0000
888.0000	5.0000	0.6000	-0.5000
930.0000	5.1000	0.8000	-1.0000
885.0000	5.3000	0.8500	-1.5000
780.0000	4.5000	0.9000	-2.0000
720.0000	-1.0000	1.2000	-2.5000
680.0000	-5.0000	1.1500	-1.0000
670.0000	-5.5000	1.0000	2.5000
695.0000	-4.0000	0.8500	1.0000
725.0000	2.0000	2.0000	0.5000
770.0000	6.0000	1.5000	0.0000
788.0000	6.5000	1.2000	0.0000
818.0000	7.5000	1.0000	0.0000
835.0000	8.0000	0.8000	0.0000
870.0000	7.5000	0.7000	0.0000
887.0000	0.0000	0.0000	0.0000

SEC. #3			
TEMP	VZ	VT	VR
508.0000	0.0000	0.0000	0.0000
600.0000	8.0000	0.2000	1.0000
575.0000	14.0000	0.2000	1.0000
525.0000	-1.2000	0.4000	1.0000
457.0000	-3.0000	0.8000	1.0000
388.0000	-4.0000	2.0000	1.5000
350.0000	10.0000	1.2000	3.0000
331.0000	22.0000	0.8000	4.5000
343.0000	24.1000	0.4000	6.0000
392.0000	25.5000	0.3000	7.0000
454.0000	23.0000	0.2500	4.5000

552.0000	18.0000	0.2000	2.0000
652.0000	-4.0000	-0.6000	-2.0000
730.0000	-10.0000	-1.4000	-1.5000
755.0000	-6.0000	-1.6000	-1.0000
780.0000	8.2000	-1.2000	0.0000
782.0000	0.0000	0.0000	0.0000

SEC. #4

TEMP	VZ	VT	VR
480.0000	0.0000	0.0000	0.0000
566.0000	3.2000	-0.1500	-0.5000
530.0000	3.4000	-0.2000	-1.5000
477.0000	4.5000	-0.4000	-3.5000
430.0000	6.0000	-0.4500	-5.0000
400.0000	8.0000	-0.3000	-5.0000
372.0000	10.1000	-0.2000	-4.5000
360.0000	11.5000	-0.1000	-2.5000
363.0000	12.5000	0.0000	0.0000
377.0000	11.5000	0.0000	4.0000
438.0000	10.1000	-0.1000	6.0000
475.0000	8.9000	-0.2500	7.5000
515.0000	7.0000	-0.4000	7.0000
582.0000	6.5000	-0.5000	5.0000
662.0000	5.0000	-0.4000	4.5000
702.0000	4.0000	-0.3000	2.0000
696.0000	0.0000	0.0000	0.0000

SEC. #5

TEMP	VZ	VT	VR
400.0000	0.0000	0.0000	0.0000
525.0000	2.0000	-0.4000	-2.0000
513.0000	3.1000	-0.4000	-3.5000
477.0000	3.5000	-0.4000	-4.5000
446.0000	5.7000	-0.4000	-5.0000
416.0000	6.8000	-0.4500	-5.0000
405.0000	8.8000	-0.5000	-5.0000
396.0000	10.2000	-0.5000	-4.5000
399.0000	11.5000	-0.5000	-2.5000
410.0000	10.5000	-0.5000	0.0000
420.0000	10.1000	-0.5000	2.0000
440.0000	9.8000	-0.5000	4.0000
460.0000	8.7000	-0.5000	5.5000
495.0000	6.5000	-0.5000	6.0000
530.0000	5.1000	-0.5000	5.5000
605.0000	3.5000	-0.5000	4.5000
629.0000	0.0000	0.0000	0.0000

SEC. #6

TEMP	VZ	VT	VR
290.0000	0.0000	0.0000	0.0000
352.0000	2.1000	0.2000	-1.0000
349.0000	3.5000	0.5000	-1.5000
343.0000	4.0000	0.5000	-1.0000
362.0000	4.5000	0.4500	0.0000

414.0000	5.8000	0.4000	-1.5000
435.0000	7.1000	-0.3500	-2.0000
440.0000	7.2000	0.3500	-1.5000
437.0000	7.5000	0.3000	-1.0000
437.0000	8.5000	0.2500	0.0000
439.0000	8.6000	0.2000	1.0000
437.0000	9.4000	0.1500	1.5000
434.0000	9.2000	0.1000	1.0000
410.0000	9.1000	0.0000	0.0000
370.0000	8.8000	-0.1000	1.0000
360.0000	7.6000	-0.2000	1.5000
347.0000	4.4000	-0.1000	1.0000
345.0000	0.0000	0.0000	0.0000

SEC. #7

TEMP	VZ	VT	VR
239.0000	0.0000	0.0000	0.0000
235.0000	4.1000	0.3000	1.0000
266.0000	4.2000	0.2500	2.5000
355.0000	5.4000	0.2000	4.5000
393.0000	5.5000	0.1500	5.5000
396.0000	6.6000	0.1000	2.0000
396.0000	7.1000	0.1000	-2.0000
397.0000	7.2000	0.1000	-2.0000
399.0000	7.3000	0.1000	-1.5000
399.0000	7.5000	0.1000	-1.5000
396.0000	8.0000	0.0000	-1.0000
388.0000	8.2000	0.0000	0.0000
374.0000	8.1000	0.0000	0.0000
355.0000	7.5000	0.0000	-1.0000
333.0000	7.5000	-0.1000	0.0000
316.0000	6.5000	-0.2000	1.5000
279.0000	6.5000	-0.2500	1.0000
246.0000	5.5000	0.2000	1.0000
233.0000	0.0000	0.0000	0.0000

SEC. #8

TEMP	VZ	VT	VR
264.0000	0.0000	0.0000	0.0000
295.0000	2.5000	0.1500	1.5000
293.0000	3.6000	0.3000	1.5000
299.0000	4.0000	0.3000	2.0000
317.0000	5.0000	0.3000	3.0000
350.0000	7.0000	0.2500	3.5000
372.0000	8.5000	0.2000	4.5000
382.0000	7.5000	0.1500	2.5000
381.0000	7.5000	0.1500	-2.0000
379.0000	7.0000	0.1500	-2.5000
376.0000	7.6000	0.1500	-1.5000
367.0000	7.6000	0.1500	-1.0000
357.0000	7.5000	0.1500	-1.0000
342.0000	7.8000	0.1500	-2.5000
310.0000	8.0000	0.2000	-5.0000
280.0000	8.0000	0.3000	-4.0000

273.0000	6.2000	0.4000	-2.5000
278.0000	6.0000	0.2000	2.0000
279.0000	4.8000	0.1000	3.0000
283.0000	0.0000	0.0000	0.0000

SEC. #9.

TEMP	VZ	UT	VR
265.0000	0.0000	0.0000	0.0000
288.0000	2.2000	0.2500	-1.5000
297.0000	2.5000	0.2000	-2.0000
307.0000	3.8000	0.1000	-1.5000
314.0000	4.5000	0.0000	0.0000
324.0000	6.0000	0.1000	1.0000
341.0000	7.5000	0.2000	1.5000
350.0000	7.8000	0.2500	1.0000
353.0000	7.8000	0.3000	0.0000
350.0000	8.0000	0.3000	0.0000
351.0000	8.2000	0.3000	-0.5000
342.0000	8.5000	0.4000	-1.0000
331.0000	7.5000	0.4000	-3.0000
312.0000	7.1000	0.4000	-3.5000
299.0000	7.6000	0.4000	-3.0000
287.0000	7.8000	0.3500	-2.0000
283.0000	8.0000	0.4000	0.0000
286.0000	6.1000	0.3000	1.5000
284.0000	4.5000	0.2000	2.0000
281.0000	0.0000	0.0000	0.0000

SEC. #10

TEMP	VZ	UT	VR
258.0000	0.0000	0.0000	0.0000
273.0000	13.7000	1.0000	1.0000
286.0000	19.7000	1.2000	1.5000
291.0000	24.2000	1.3000	2.0000
296.0000	24.0000	1.1500	2.5000
290.0000	24.0000	1.0000	1.5000
286.0000	25.0000	0.8000	0.0000
276.0000	23.0000	0.6000	-1.0000
266.0000	21.2000	0.5000	-1.0000
255.0000	16.5000	0.5000	1.0000
242.0000	0.0000	0.0000	0.0000

RUN # 9

SEC. #1

TEMP	VZ	VT	VR
1011.0000	0.0000	0.0000	0.0000
1138.0000	2.8000	0.4000	0.0000
1207.0000	6.1000	0.6000	-1.5000
1166.0000	7.0000	0.7500	-2.5000
952.0000	5.0000	0.8000	-2.5000
826.0000	-2.0000	-0.1000	0.0000
863.0000	-2.0000	-0.1000	0.0000
926.0000	-2.0000	-0.1000	0.0000
945.0000	-2.0000	-0.1000	1.5000
1016.0000	-2.0000	-0.1000	2.0000
1164.0000	-2.0000	-0.3000	-1.0000
1291.0000	7.5000	-1.0000	3.0000
1305.0000	6.1000	-0.7000	4.0000
1312.0000	5.5000	-0.5000	2.0000
1315.0000	3.0000	-0.3000	1.0000
1323.0000	0.0000	0.0000	0.0000

SEC. #2

TEMP	VZ	VT	VR
759.0000	0.0000	0.0000	0.0000
950.0000	5.0000	0.1500	-2.0000
980.0000	5.0000	0.2000	-2.0000
950.0000	5.2000	0.2500	-1.7000
920.0000	6.8000	0.5000	-1.5000
850.0000	7.2000	0.8000	-1.0000
740.0000	-3.0000	1.0000	0.0000
680.0000	-2.5000	-0.1000	0.0000
670.0000	-3.2000	-0.2000	0.5000
720.0000	-3.1000	-0.2000	1.0000
790.0000	4.0000	-0.2000	1.5000
880.0000	7.1000	-0.3000	2.5000
990.0000	8.2000	-0.2000	3.0000
1020.0000	8.4000	-0.2500	4.0000
1000.0000	7.2000	-0.2000	2.5000
960.0000	0.0000	0.0000	0.0000

SEC. #3

TEMP	VZ	VT	VR
480.0000	0.0000	0.0000	0.0000
560.0000	11.0000	0.1000	1.0000
553.0000	13.5000	0.3000	2.0000
510.0000	10.0000	0.5000	2.5000
450.0000	-2.0000	1.0000	1.5000
370.0000	-4.1000	1.8000	1.0000
320.0000	-6.5000	1.5000	1.0000
300.0000	22.1000	1.2000	2.5000
302.0000	23.1000	0.7000	3.0000
333.0000	24.1000	-0.2000	5.0000
392.0000	-6.0000	-0.2000	5.5000

467.0000	-4.0000	-0.2000	5.0000
555.0000	-3.5000	-0.2000	3.0000
635.0000	-1.4000	0.4000	-2.0000
698.0000	9.0000	0.0000	0.0000
706.0000	8.5000	-1.5000	0.0000
695.0000	0.0000	0.0000	0.0000

SEC. #4

TEMP	VZ	VT	VR
468.0000	0.0000	0.0000	0.0000
595.0000	3.0000	-0.3000	-1.5000
595.0000	3.0000	-0.3500	-3.0000
550.0000	3.0000	-0.4000	-4.5000
495.0000	4.9000	-0.5000	-5.5000
440.0000	8.1000	-0.5000	-6.0000
408.0000	9.1000	-0.5000	-5.5000
390.0000	9.5000	-0.5000	-2.5000
386.0000	9.9000	-0.7000	0.0000
403.0000	10.0000	-1.0000	4.5000
434.0000	9.5000	-1.2000	6.5000
474.0000	9.1000	-1.2000	7.0000
520.0000	9.0000	-1.3000	7.5000
590.0000	9.2000	-1.4000	6.5000
660.0000	9.5000	-1.8000	5.0000
695.0000	6.0000	-2.6000	4.5000
700.0000	0.0000	0.0000	0.0000

SEC. #5

TEMP	VZ	VT	VR
452.0000	0.0000	0.0000	0.0000
564.0000	2.0000	-0.3000	-1.5000
560.0000	3.0000	-0.3000	-2.5000
527.0000	4.2000	-0.2500	-3.5000
494.0000	5.8000	-0.2000	-4.5000
457.0000	6.2000	-0.1000	-4.5000
433.0000	7.9000	0.0000	-4.5000
425.0000	9.8000	0.1000	-3.5000
420.0000	9.5000	0.1000	-2.0000
427.0000	10.6000	0.0000	0.0000
440.0000	10.1000	-0.1000	2.5000
464.0000	8.6000	-0.2000	5.0000
495.0000	7.2000	-0.3000	5.5000
530.0000	6.2000	-0.4000	6.0000
580.0000	4.6000	-0.4500	5.0000
620.0000	3.1000	-0.5000	5.0000
635.0000	0.0000	0.0000	0.0000

SEC. #6

TEMP	VZ	VT	VR
292.0000	0.0000	0.0000	0.0000
346.0000	2.7000	0.3000	-1.0000
346.0000	2.9000	0.4000	-1.5000
242.0000	3.5000	0.6000	-1.0000
397.0000	4.1000	0.7000	0.0000

440.0000	6.1000	0.6500	-1.5000
455.0000	8.5000	0.5500	-2.0000
459.0000	9.2000	0.5500	-2.0000
462.0000	9.9000	0.5500	-1.0000
466.0000	10.1000	0.3500	0.0000
468.0000	10.2000	0.2500	1.0000
470.0000	10.2000	0.2000	2.0000
460.0000	10.1000	0.1500	2.5000
470.0000	8.8000	0.1000	2.0000
436.0000	8.1000	0.1000	1.0000
430.0000	7.0000	0.0000	1.5000
413.0000	5.9000	-0.1000	0.1000
410.0000	0.0000	0.0000	0.0000

SEC. #7

TEMP	VZ	VT	VR
260.0000	0.0000	0.0000	0.0000
265.0000	2.1000	0.1500	5.0000
280.0000	3.5000	0.5000	6.0000
352.0000	4.0000	0.5000	6.5000
408.0000	5.5000	0.5000	6.5000
415.0000	5.1000	0.1000	2.5000
416.0000	5.5000	0.0000	0.0000
417.0000	6.1000	0.1500	-2.0000
417.0000	6.2000	0.2500	-2.0000
420.0000	6.9000	0.3000	-2.0000
418.0000	8.1000	0.2500	-2.0000
410.0000	8.4000	0.2500	-3.5000
402.0000	9.1000	0.2000	-5.0000
382.0000	9.2000	0.0000	-6.5000
358.0000	9.2000	0.0000	-6.5000
336.0000	9.5000	0.4000	-6.5000
297.0000	9.1000	0.5000	-6.0000
255.0000	6.1000	0.3000	-5.0000
242.0000	0.0000	0.0000	0.0000

SEC. #8

TEMP	VZ	VT	VR
292.0000	0.0000	0.0000	0.0000
328.0000	2.1000	0.0500	0.5000
324.0000	4.5000	0.1500	1.0000
327.0000	5.1000	0.1500	1.5000
338.0000	6.4000	0.1000	3.0000
369.0000	6.5000	0.0800	5.5000
398.0000	5.1000	0.0600	5.0000
394.0000	6.2000	0.0400	3.0000
392.0000	7.1000	0.0200	-1.5000
392.0000	7.8000	0.0100	-1.5000
397.0000	8.5000	0.0000	-1.5000
398.0000	8.8000	0.0400	-1.0000
385.0000	8.7000	0.0600	-1.0000
364.0000	8.0000	0.0800	-1.5000
345.0000	8.2000	0.1000	-3.0000
332.0000	8.5000	0.2000	-5.0000

320.0000	7.1000	0.1000	-4.5000
321.0000	5.1000	0.0400	-2.0000
320.0000	4.5000	0.0200	1.0000
323.0000	0.0000	0.0000	0.0000

SEC. #9

TEMP	VZ	VT	VR
292.0000	0.0000	0.0000	0.0000
299.0000	1.5000	0.0500	1.0000
321.0000	2.2000	0.0500	1.0000
325.0000	3.6000	0.0500	1.0000
332.0000	4.8000	0.0500	1.0000
328.0000	5.2000	0.0500	1.0000
354.0000	6.1000	0.0500	2.5000
380.0000	6.0000	0.0500	3.5000
384.0000	6.0000	0.0500	3.5000
384.0000	6.9000	0.0500	1.5000
375.0000	7.9000	0.0500	0.0000
364.0000	8.5000	0.0500	0.5000
347.0000	9.1000	0.1000	-1.5000
339.0000	8.0000	0.1500	-2.0000
321.0000	8.0000	0.2000	-3.5000
292.0000	8.4000	0.2500	-4.0000
322.0000	8.5000	0.3000	-3.5000
321.0000	6.5000	0.3500	-2.0000
315.0000	4.5000	0.4000	0.0000
312.0000	0.0000	0.0000	0.0000

SEC. #10

TEMP	VZ	VT	VR
271.0000	0.0000	0.0000	0.0000
281.0000	17.2000	1.0000	2.0000
291.0000	19.0000	1.0000	3.0000
291.0000	21.5000	1.0000	2.5000
289.0000	21.5000	1.0000	1.5000
278.0000	24.5000	1.0000	0.0000
264.0000	22.0000	1.1000	-0.5000
252.0000	20.5000	1.0000	-1.0000
245.0000	19.0000	0.9500	0.0000
231.0000	17.0000	0.5000	5.0000
229.0000	0.0000	0.0000	0.0000

RUN #10

SEC. #1			
TEMP	VZ	VT	VR
919.0000	0.0000	0.0000	0.0000
992.0000	2.9000	0.4000	-1.0000
1139.0000	3.8000	1.2000	-2.5000
1266.0000	4.5000	1.4000	-2.0000
1185.0000	4.6000	1.2000	0.0000
961.0000	-1.0000	1.0000	3.0000
893.0000	-2.5000	0.3000	2.5000
908.0000	-3.5000	0.2000	0.0000
947.0000	-3.1000	0.0000	0.0000
922.0000	-2.1000	-0.3000	0.0000
947.0000	-1.0000	-0.3000	-3.0000
1051.0000	4.0000	0.0000	0.0000
1201.0000	6.1000	0.0000	4.5000
1266.0000	5.2000	-0.2000	4.5000
1286.0000	5.0000	-0.8000	3.0000
1256.0000	0.0000	0.0000	0.0000

SEC. #2			
TEMP	VZ	VT	VR
740.0000	0.0000	0.0000	0.0000
940.0000	5.1000	0.5000	0.0000
980.0000	6.2000	0.7000	0.0000
985.0000	6.5000	1.0000	-1.5000
953.0000	6.5000	1.2000	-2.0000
903.0000	6.5000	1.0000	-2.5000
840.0000	2.5000	0.5000	0.0000
788.0000	-3.1000	0.4000	0.0000
780.0000	-3.0000	0.2000	2.5000
800.0000	-2.1000	0.1000	2.0000
815.0000	2.5000	0.0000	1.5000
860.0000	4.0000	-0.2000	1.0000
925.0000	5.5000	-0.3000	0.5000
981.0000	6.5000	-0.5000	0.0000
1024.0000	5.0000	-1.0000	0.0000
1010.0000	0.0000	0.0000	0.0000

SEC. #3			
TEMP	VZ	VT	VR
497.0000	0.0000	0.0000	0.0000
555.0000	6.2000	0.2500	0.0000
552.0000	8.5000	0.4000	0.0000
500.0000	7.0000	0.6000	-0.5000
435.0000	1.0000	0.8000	-0.5000
366.0000	-3.9000	1.7000	1.0000
316.0000	-5.5000	1.8000	2.5000
295.0000	25.0000	1.8000	2.0000
298.0000	24.5000	1.0000	4.0000
351.0000	23.5000	0.7500	8.0000
400.0000	20.5000	0.7000	8.5000

480.0000	-6.5000	0.5000	3.0000
562.0000	-4.0000	-0.5000	-3.0000
636.0000	-2.0000	-0.5000	-2.0000
670.0000	12.0000	-0.5000	-1.0000
675.0000	9.5000	-0.4000	0.0000
670.0000	0.0000	0.0000	0.0000

SEC. #4

TEMP	VZ	VT	VR
434.0000	0.0000	0.0000	0.0000
540.0000	3.2000	0.1000	-2.0000
525.0000	3.2000	0.1000	-3.5000
480.0000	4.1000	0.2000	-4.5000
435.0000	5.1000	0.2500	-5.0000
400.0000	6.2000	0.3000	-5.5000
365.0000	8.1000	0.3500	-5.0000
360.0000	10.1000	0.2000	-2.5000
360.0000	11.0000	-0.2000	0.0000
386.0000	10.5000	-0.5000	3.0000
412.0000	8.8000	-0.5500	5.0000
450.0000	8.1000	-0.5000	6.5000
488.0000	7.6000	-0.4500	7.0000
544.0000	6.8000	-0.4000	6.0000
613.0000	5.5000	-0.3500	4.7000
640.0000	4.5000	-0.3500	3.5000
630.0000	0.0000	0.0000	0.0000

SEC. #5

TEMP	VZ	VT	VR
443.0000	0.0000	0.0000	0.0000
516.0000	2.7000	0.1500	-1.5000
511.0000	3.1000	0.2000	-2.5000
483.0000	5.1000	0.2000	-4.0000
451.0000	7.5000	0.2000	-5.0000
430.0000	9.1000	0.1500	-5.0000
407.0000	9.5000	0.1000	-4.0000
400.0000	10.0000	0.0000	-2.5000
403.0000	9.1000	0.0000	0.0000
413.0000	9.0000	-0.1000	2.0000
430.0000	8.5000	-0.1500	4.0000
454.0000	8.1000	-0.3000	5.5000
480.0000	7.5000	-0.4000	6.0000
505.0000	7.1000	-0.4000	5.5000
577.0000	6.5000	-0.4000	5.0000
595.0000	3.5000	-0.4000	4.0000
585.0000	0.0000	0.0000	0.0000

SEC. #6

TEMP	VZ	VT	VR
258.0000	0.0000	0.0000	0.0000
309.0000	2.1000	0.1500	-0.8000
320.0000	3.5000	0.1500	-0.8000
336.0000	4.5000	0.2000	-1.5000
369.0000	6.5000	0.2000	-2.0000

403.0000	7.5000	0.2500	-3.0000
420.0000	8.2000	0.2500	-3.0000
422.0000	8.5000	0.2000	-2.0000
426.0000	9.1000	0.1500	-1.5000
424.0000	9.0000	0.1000	0.0000
430.0000	8.8000	0.0000	1.0000
428.0000	8.4000	0.0000	1.5000
416.0000	8.3000	-0.1000	2.0000
388.0000	8.3000	-0.1500	1.5000
365.0000	7.9000	-0.1500	0.0000
330.0000	5.2000	-0.3000	0.0000
320.0000	3.0000	-0.2000	1.0000
345.0000	0.0000	0.0000	0.0000

SEC. #7

TEMP	VZ	VT	VR
198.0000	0.0000	0.0000	0.0000
207.0000	2.0000	0.4000	6.5000
233.0000	3.0000	0.8000	7.0000
313.0000	5.8000	0.7000	6.5000
360.0000	6.5000	0.5000	6.0000
365.0000	6.5000	0.3000	0.0000
368.0000	6.6000	0.2500	-3.0000
373.0000	6.5000	0.2500	-2.0000
372.0000	6.5000	0.2500	-1.5000
376.0000	6.9000	0.2500	-1.0000
376.0000	7.1000	0.2500	0.0000
365.0000	7.2000	0.2500	0.0000
357.0000	7.5000	0.3000	-1.0000
335.0000	8.0000	0.5000	-3.0000
299.0000	8.2000	0.6000	-4.5000
280.0000	8.5000	0.7000	-6.0000
247.0000	9.0000	0.7500	-7.0000
210.0000	5.1000	0.7500	-5.0000
228.0000	0.0000	0.0000	0.0000

SEC. #8

TEMP	VZ	VT	VR
252.0000	0.0000	0.0000	0.0000
286.0000	2.5000	0.5000	0.5000
298.0000	4.5000	0.8000	1.0000
304.0000	6.1000	0.7500	1.5000
317.0000	7.5000	0.5000	2.5000
350.0000	5.5000	0.4000	5.0000
370.0000	4.5000	0.2000	4.5000
382.0000	4.5000	0.1500	1.0000
386.0000	4.5000	0.1500	-1.5000
385.0000	5.1000	0.1500	-1.5000
382.0000	5.1000	0.2000	-1.5000
377.0000	5.4000	0.2500	-1.0000
363.0000	6.1000	0.4000	-1.0000
342.0000	6.5000	0.6000	-2.0000
317.0000	6.7000	0.6000	-5.0000
290.0000	8.5000	0.5000	-6.0000

285.0000	9.5000	0.3000	-5.0000
286.0000	6.8000	0.2000	-1.0000
290.0000	4.5000	0.1000	2.0000
292.0000	0.0000	0.0000	0.0000

SEC. #9

TEMP	VZ	VT	VR
250.0000	0.0000	0.0000	0.0000
275.0000	2.5000	-0.1000	-1.0000
285.0000	3.9000	-0.2000	-1.0000
291.0000	4.9000	-0.2000	-1.0000
298.0000	5.9000	-0.2000	1.0000
310.0000	5.9000	-0.1000	2.0000
330.0000	5.9000	0.0000	2.5000
340.0000	5.9000	0.0000	2.0000
349.0000	5.9000	0.1000	1.0000
345.0000	5.9000	0.1500	0.0000
345.0000	5.9000	0.1500	-0.5000
342.0000	5.7000	0.1500	-1.0000
328.0000	5.7000	0.1500	-1.5000
310.0000	6.1000	0.2000	-3.0000
290.0000	7.1000	0.3000	-3.5000
282.0000	7.5000	0.4000	-2.5000
282.0000	7.5000	0.4000	-1.0000
285.0000	6.6000	0.3000	0.0000
286.0000	2.5000	0.2000	1.0000
283.0000	0.0000	0.0000	0.0000

SEC. #10

TEMP	VZ	VT	VR
255.0000	0.0000	0.0000	0.0000
266.0000	19.8000	0.5500	1.5000
276.0000	20.1000	0.8000	2.0000
281.0000	20.1000	0.8000	3.5000
279.0000	20.1000	1.0000	2.5000
278.0000	20.1000	1.0000	0.0000
274.0000	20.1000	0.8500	-0.5000
260.0000	20.0000	0.8000	-1.0000
248.0000	19.6000	0.7500	1.0000
237.0000	17.8000	0.7000	3.0000
222.0000	0.0000	0.0000	0.0000

SHAPE FACTOR FOR EXIT SECTION TEMPERATURE
PROFILE FOR RUNS 1-10

DIST.	RUN #									
	1	2	3	4	5	6	7	8	9	10
0.0	0.023	-0.003	-0.11	-0.035	-0.018	-0.043	-0.051	-0.064	0.025	-0.026
0.2	0.055	0.083	-0.08	0.020	0.053	0.065	0.005	-0.004	0.066	0.022
0.4	0.037	0.126	-0.02	0.062	0.116	0.117	0.045	0.044	0.107	0.062
0.6	-0.003	0.117	0.045	0.070	0.147	0.147	0.062	0.064	0.107	0.084
0.8	-0.014	0.091	0.083	0.090	0.13	0.130	0.056	0.084	0.098	0.075
1.0	-0.017	0.043	0.094	0.090	0.095	0.104	0.056	0.06	0.053	0.07
1.2	0.014	0.029	0.088	0.070	0.035	0.039	0.04	0.044	0.008	0.053
1.4	0.0	-0.032	0.06	0.02	-0.035	-0.26	0.017	0.004	-0.053	-0.004
1.6	-0.008	-0.104	0.032	-0.02	-0.116	-0.091	-0.04	-0.036	-0.082	-0.053
1.8	0.043	-0.157	-0.08	-0.16	-0.183	-0.225	-0.107	-0.076	-0.139	-0.101
1.9	-0.125	-0.184	-0.11	-0.21	-0.235	-0.225	-0.102	-0.129	-0.184	-0.163

TEMPERATURE AT EXIT SECTION FOR RUNS 11-20

DIST.	RUN #									
	11	12	13	14	15	16	17	18	19	20
0.0	321	182	419	506	356	278	357	362	345	365
0.2	339	186	428	503	366	285	365	368	359	380
0.4	354	187	428	493	375	299	374	373	367	381
0.6	371	189	405	480	378	307	383	389	372	385
0.8	366	200	410	468	383	305	387	400	374	389
1.0	350	197	405	470	383	282	383	411	378	397
1.2	340	187	405	465	372	292	400	405	388	401
1.4	308	192	418	474	375	288	418	402	397	410
1.6	297	178	416	493	371	287	374	396	380	389
1.8	270	173	421	506	366	278	365	378	377	372
1.8	254	172	372	454	361	278	365	369	369	356

Listing of computer program CHECK.FOR as obtained from DEC computer

LSI 11/23

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      VIRTUAL T1(25),T2(25),T3(25),T4(25),T5(25),T6(25),T7(25),T8(25)
2     ,T9(25),T10(25),VZ1(25),VZ2(25),VZ3(25),VZ4(25),VZ5(25),VZ6(25),
3     VZ7(25),VZ8(25),VZ9(25),VZ10(25),VT1(25),VT2(25),VT3(25),VT4(25),
4     ,VT5(25),VT6(25),VT7(25),VT8(25),VT9(25),VT10(25),VR1(25),
5     VR2(25),VR3(25),VR4(25),VR5(25),VR6(25),VR7(25),VR8(25),VR9(25),
6     VR10(25),D1(25),
7     D2(25),D3(25),D4(25),D5(25),D6(25),D7(25),D8(25),D9(25),D10(25)
      REAL NZMD
      COMMON XCO,XCO2,XH2,FMASSD,AMASSD,FRAPRE,FRAPRI,FRASEC,FRADIL
      COMMON/G/RCG1,RCG2,RCG3,RCG4,RCG5,RCG6,RCG7,RCG8,RCG9,RCG10
      COMMON/BLK1/AD1,AD2,AD3,AD4,AD5,AD6,AD7,AD8,AD9,AD10
      COMMON/BLK2/TOMAS1,TOMAS2,TOMAS3,TOMAS4,TOMAS5,TOMAS6,TOMAS7,
2     TOMAS8,TOMAS9,TOMAS10
      COMMON/BLK2/ANMOM1,ANMOM2,ANMOM3,ANMOM4,ANMOM5,ANMOM6,ANMOM7,
2     ANMOM8,ANMOM9,ANMOM10
E *****
C GLOSSARY OF THE VARIOUS TERMS AND SYMBOLS USED IN THIS LISTING OR THE
C SURROUTINES LINKED TO IT
C *****
C THE NUMERALS 1 THROUGH 10 BESIDE ANY VARIABLE INDICATES THE SECTION
C NUMBER FOR THE VARIABLE
C AD      =MASS FLOW RATE AT A PARTICULAR SECTION IN THE COMBUSTOR FROM INLET
C          FLOW RATES
C AMASSD=MASS FLOW RATE OF AIR
C ANMOM  =FLUX OF ANGULAR MOMENTUM AT A COMBUSTOR SECTION
C CO2MD  =MASS FLOW RATE OF CO2 AT EXIT SECTION
C D       =(SUBSCRIPTED)DIAMETER FOR VARIOUS POINTS AT ANY SECTION
C DAREA  =DIFFERENTIAL AREA AT ANY MEASURING SECTION
C DR      =RADIAL INCREMENT OF MEASUREMENTS
C DTMEAN =MEAN TEMP RISE FROM INLET AT FIRST SECTION
C EPS     =EMISSIVITY OF COMBUSTOR SURFACE
C FMASSD =FUEL MASS FLOW RATE
C FRAPRE =FRACTION OF AIR PREMIXED WITH FUEL
C FRAPRI =FRACTION OF AIR IN THE PRIMARY ZONE
C FRASEC =FRACTION OF AIR IN THE SECONDARY ZONE
C FRADIL =FRACTION OF AIR IN THE DILUTION ZONE
C H2OMD  =MASS FLOW RATE OF H2O AT EXIT SECTION
C NZMD   =MASS FLOW RATE OF N2 AT EXIT SECTION
C O2MD   =MASS FLOW RATE OF O2 AT EXIT SECTION
C PI      =3.14159
C RCG     =AVERAGE VALUE OF CHARACTERISITIC GAS CONSTANT
C RHO     =GAS DENSITY
C SM      =TOTAL MASS AT EXIT SECTION
C T       =(SUBSCRIPTED)TEMPERATURE READINGS AT EACH SECTION
C TMEAN   =MEAN TEMPERATURE
C TO      =AMBIENT TEMPERATURE
C TOMAS   =MASS FLOW RATE AT ANY SECTION AS CALCULATED FROM MEASUREMENTS
C VR      =(SUBSCRIPTED) RADIAL VELOCITY COMPONENT AT THE VARIOUS SECTIONS
C          OF THE COMBUSTOR
C VT      =(SUBSCRIPTED) CIRCUMFERENTIAL VELOCITY COMPONENT AT THE VARIOUS
C          SECTIONS OF THE COMBUSTOR
C VTMEAN  =MEAN CIRCUMFERENTIAL VELOCITY
C VZ      =(SUBSCRIPTED)AXIAL VELOCITY COMPONENT AT THE VARIOUS SECTIONS
C          OF THE COMBUSTOR

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C VZMEAN=MEAN AXIAL VELOCITY
C XCO   =MOL FRACTION OF CO IN FUEL
C XCO2  =MOL FRACTION OF CO2 IN FUEL
C XH2   =MOL FRACTION OF H2 IN FUEL
C XMCO  =XCO*MOL WT.
C XMCO2 =XCO2*MOL WT.
C XMH2  =XH2*MOL WT.
C XMTOT =XMCO+XMCO2+XMH2
      READ(5,50)II
      WRITE(6,65)II
50    FORMAT(I2)
65    FORMAT(5X,'RUN #',I2)
C
C SUBROUTINE INPUT READS IN THE INPUT VALUES AT THE TEN SECTIONS FOR
C ANY SINGLE RUN
C
      CALL INPUT(T1,T2,T3,T4,T5,T6,T7,T8,T9,T10,VZ1,VZ2,VZ3,VZ4,
2     VZ5,VZ6,VZ7,VZ8,VZ9,VZ10,VT1,VT2,VT3,VT4,VT5,VT6,VT7,VT8,VT9,
3     VT10,VR1,VR2,VR3,VR4,VR5,VR6,VR7,VR8,VR9,VR10,D1,D2,D3,D4,D5,
4     D6,D7,D8,D9,D10)
C
C SUBROUTINE GASCON CALCULATES AN AVERAGE VALUE FOR THE GAS CONSTANT
C AND THE MASS FLOW RATE AT EACH SECTION FROM THE FUEL AND AIR MASS
C FLOW RATES AND THE LINER OPEN HOLE AREA THAT ADMITS THE SECONDARY
C AND THE DILUTION AIR
C
      CALL GASCON
      DR=0.005
      TO=293.0
      PI=3.14159
      XMCO=28.0*XCO
      XMCO2=44.0*XCO2
      XMH2=2.0*XH2
      XMTOT=XMCO+XMCO2+XMH2
      GCO=XMCO/XMTOT
      GCO2=XMCO2/XMTOT
      GH2=XMH2/XMTOT
      COIN=GCO*FMASSD
      CO2IN=GCO2*FMASSD
      H2IN=GH2*FMASSD
      CO2MD=(44.0/28.0)*GCO*FMASSD+GCO2*FMASSD
      H2MD=9.0*GH2*FMASSD
      O2MD=0.233*AMASSD-16.0/28.0*GCO*FMASSD-8.0*GH2*FMASSD
      N2MD=0.767*AMASSD
      TOMAS1=0.0
      TOMAS2=0.0
      TOMAS3=0.0
      TOMAS4=0.0
      TOMAS5=0.0
      TOMAS6=0.0
      TOMAS7=0.0
      TOMAS8=0.0
      TOMAS9=0.0
      TOMAS0=0.0

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ANMOM1=0.0
ANMOM2=0.0
ANMOM3=0.0
ANMOM4=0.0
ANMOM5=0.0
ANMOM6=0.0
ANMOM7=0.0
ANMOM8=0.0
ANMOM9=0.0
ANMOM0=0.0

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C
C INTEGRATION OF MASS FLOW RATE AND AXIAL FLUX OF ANGULAR MOMENTUM
C OVER THE AREA OF THE SECTION FOR SECTIONS 1-10
C

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```

DO 300 I=1,15
  TMEAN=0.5*(T1(I)+T1(I+1))+273.0
  RHO=101300.0/(RCG1*TMEAN)
  VZMEAN=0.5*(VZ1(I)+VZ1(I+1))
  VTMEAN=0.5*(VT1(I)+VT1(I+1))
  DAREA=0.5*PI*D1(I)*DR
  TOMAS1=TOMAS1+3600.0*DAREA*RHO*VZMEAN
  ANMOM1=ANMOM1+0.5*D1(I)*DAREA*RHO*VZMEAN*VTMEAN
300 CONTINUE
DO 301 I=1,15
  TMEAN=0.5*(T2(I)+T2(I+1))+273.0
  RHO=101300.0/(RCG2*TMEAN)
  VZMEAN=0.5*(VZ2(I)+VZ2(I+1))
  VTMEAN=0.5*(VT2(I)+VT2(I+1))
  DAREA=0.5*PI*D2(I)*DR
  TOMAS2=TOMAS2+3600.0*DAREA*RHO*VZMEAN
  ANMOM2=ANMOM2+0.5*D2(I)*DAREA*RHO*VZMEAN*VTMEAN
301 CONTINUE
DO 302 I=1,16
  TMEAN=0.5*(T3(I)+T3(I+1))+273.0
  RHO=101300.0/(RCG3*TMEAN)
  VZMEAN=0.5*(VZ3(I)+VZ3(I+1))
  VTMEAN=0.5*(VT3(I)+VT3(I+1))
  DAREA=0.5*PI*D3(I)*DR
  TOMAS3=TOMAS3+3600.0*DAREA*RHO*VZMEAN
  ANMOM3=ANMOM3+0.5*D3(I)*DAREA*RHO*VZMEAN*VTMEAN
302 CONTINUE
DO 303 I=1,16
  TMEAN=0.5*(T4(I)+T4(I+1))+273.0
  RHO=101300.0/(RCG4*TMEAN)
  VZMEAN=0.5*(VZ4(I)+VZ4(I+1))
  VTMEAN=0.5*(VT4(I)+VT4(I+1))
  DAREA=0.5*PI*D4(I)*DR
  TOMAS4=TOMAS4+3600.0*DAREA*RHO*VZMEAN
  ANMOM4=ANMOM4+0.5*D4(I)*DAREA*RHO*VZMEAN*VTMEAN
303 CONTINUE
DO 304 I=1,16
  TMEAN=0.5*(T5(I)+T5(I+1))+273.0
  RHO=101300.0/(RCG5*TMEAN)
  VZMEAN=0.5*(VZ5(I)+VZ5(I+1))

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VTMEAN=0.5*(VT5(I)+VT5(I+1))
DAREA=0.5*PI*D5(I)*DR
TOMAS5=TOMAS5+3600.0*DAREA*RHO*VZMEAN
ANMOM5=ANMOM5+0.5*D5(I)*DAREA*RHO*VZMEAN*VTMEAN
304 CONTINUE
DO 305 I=1,17
TMEAN=0.5*(T6(I)+T6(I+1))+273.0
RHO=101300.0/(RCG6*TMEAN)
VZMEAN=0.5*(VZ6(I)+VZ6(I+1))
VTMEAN=0.5*(VT6(I)+VT6(I+1))
DAREA=0.5*PI*D6(I)*DR
TOMAS6=TOMAS6+3600.0*DAREA*RHO*VZMEAN
ANMOM6=ANMOM6+0.5*D6(I)*DAREA*RHO*VZMEAN*VTMEAN
305 CONTINUE
DO 306 I=1,18
TMEAN=0.5*(T7(I)+T7(I+1))+273.0
RHO=101300.0/(RCG7*TMEAN)
VZMEAN=0.5*(VZ7(I)+VZ7(I+1))
VTMEAN=0.5*(VT7(I)+VT7(I+1))
DAREA=0.5*PI*D7(I)*DR
TOMAS7=TOMAS7+3600.0*DAREA*RHO*VZMEAN
ANMOM7=ANMOM7+0.5*D7(I)*DAREA*RHO*VZMEAN*VTMEAN
306 CONTINUE
DO 307 I=1,19
TMEAN=0.5*(T8(I)+T8(I+1))+273.0
RHO=101300.0/(RCG8*TMEAN)
VZMEAN=0.5*(VZ8(I)+VZ8(I+1))
VTMEAN=0.5*(VT8(I)+VT8(I+1))
DAREA=0.5*PI*D8(I)*DR
TOMAS8=TOMAS8+3600.0*DAREA*RHO*VZMEAN
ANMOM8=ANMOM8+0.5*D8(I)*DAREA*RHO*VZMEAN*VTMEAN
307 CONTINUE
DO 308 I=1,19
TMEAN=0.5*(T9(I)+T9(I+1))+273.0
RHO=101300.0/(RCG9*TMEAN)
VZMEAN=0.5*(VZ9(I)+VZ9(I+1))
VTMEAN=0.5*(VT9(I)+VT9(I+1))
DAREA=0.5*PI*D9(I)*DR
TOMAS9=TOMAS9+3600.0*DAREA*RHO*VZMEAN
ANMOM9=ANMOM9+0.5*D9(I)*DAREA*RHO*VZMEAN*VTMEAN
308 CONTINUE
DO 309 I=1,10
TMEAN=0.5*(T10(I)+T10(I+1))+273.0
RHO=101300.0/(RCG10*TMEAN)
VZMEAN=0.5*(VZ10(I)+VZ10(I+1))
VTMEAN=0.5*(VT10(I)+VT10(I+1))
DAREA=0.5*PI*D10(I)*DR
TOMAS0=TOMAS0+3600.0*DAREA*RHO*VZMEAN
ANMOM0=ANMOM0+0.5*D10(I)*DAREA*RHO*VZMEAN*VTMEAN
309 CONTINUE

```

C
C SUBROUTINE OUTPUT PRINTS THE CALCULATED RESULTS AS WELL AS THE INPUT
C VALUES FOR TEMPERATURE AND VELOCITY COMPONENTS
C

CALL DU PUT(T1,T2,T3,T4,T5,T6,T7,T8,T9,T10,VZ1,VZ2,VZ3,VZ4
2 ,VZ5,VZ6,VZ7,VZ8,VZ9,VZ10,VT1,VT2,VT3,VT4,VT5,VT6,VT7,VT8,VT9,
3 VT10,VR1,VR2,VR3,VR4,VR5,VR6,VR7,VR8,VR9,VR10)
STOP
END

4 4

OF / DE

```

SUBROUTINE INPUT(T1,T2,T3,T4,T5,T6,T7,T8,T9,T10,VZ1,VZ2,VZ3,VZ4,
2 VZ5,VZ6,VZ7,VZ8,VZ9,VZ10,VT1,VT2,VT3,VT4,VT5,VT6,VT7,VT8,VT9,
3 VT10,VR1,VR2,VR3,VR4,VR5,VR6,VR7,VR8,VR9,VR10,D1,D2,D3,D4,D5,
4 D6,D7,D8,D9,D10)

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C
C THIS SUBROUTINE READS IN THE INPUT DATA
C

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```

      VIRTUAL T1(25),T2(25),T3(25),T4(25),T5(25),T6(25),T7(25),T8(25)
2      ,T9(25),T10(25),VZ1(25),VZ2(25),VZ3(25),VZ4(25),VZ5(25),VZ6(25),
3      VZ7(25),VZ8(25),VZ9(25),VZ10(25),VT1(25),VT2(25),VT3(25),VT4(25),
4      ,VT5(25),VT6(25),VT7(25),VT8(25),VT9(25),VT10(25),VR1(25),
5      VR2(25),VR3(25),VR4(25),VR5(25),VR6(25),VR7(25),VR8(25),VR9(25),
6      VR10(25),D1(25),
7      D2(25),D3(25),D4(25),D5(25),D6(25),D7(25),D8(25),D9(25),D10(25)
      COMMON XCO,XCO2,XH2,FMASSD,AMASSD,FRAPRE,FRAPRI,FRASEC,FRADIL
      DATA='000
      LOGICAL *1 DATA
      OPEN(UNIT=4,NAME='DATA',TYPE='OLD',READONLY)
      DO 1 I=1,16
      READ(4,100)T1(I)
1      CONTINUE
      DO 2 I=1,16
      READ(4,100)T2(I)
2      CONTINUE
      DO 3 I=1,17
      READ(4,100)T3(I)
3      CONTINUE
      DO 4 I=1,17
      READ(4,100)T4(I)
4      CONTINUE
      DO 5 I=1,17
      READ(4,100)T5(I)
5      CONTINUE
      DO 6 I=1,18
      READ(4,100)T6(I)
6      CONTINUE
      DO 7 I=1,19
      READ(4,100)T7(I)
7      CONTINUE
      DO 8 I=1,20
      READ(4,100)T8(I)
8      CONTINUE
      DO 9 I=1,20
      READ(4,100)T9(I)
9      CONTINUE
      DO 10 I=1,11
      READ(4,100)T10(I)
10     CONTINUE
      DO 11 I=1,16
      READ(4,100)VZ1(I)
11     CONTINUE
      DO 12 I=1,16
      READ(4,100)VZ2(I)
12     CONTINUE

```

```
DO 13 I=1,17
READ(4,100)VZ3(I)
13 CONTINUE
DO 14 I=1,17
READ(4,100)VZ4(I)
14 CONTINUE
DO 15 I=1,17
READ(4,100)VZ5(I)
15 CONTINUE
DO 16 I=1,18
READ(4,100)VZ6(I)
16 CONTINUE
DO 17 I=1,19
READ(4,100)VZ7(I)
17 CONTINUE
DO 18 I=1,20
READ(4,100)VZ8(I)
18 CONTINUE
DO 19 I=1,20
READ(4,100)VZ9(I)
19 CONTINUE
DO 20 I=1,41
READ(4,100)VZ10(I)
20 CONTINUE
DO 21 I=1,16
READ(4,100)VT1(I)
21 CONTINUE
DO 22 I=1,16
READ(4,100)VT2(I)
22 CONTINUE
DO 23 I=1,17
READ(4,100)VT3(I)
23 CONTINUE
DO 24 I=1,17
READ(4,100)VT4(I)
24 CONTINUE
DO 25 I=1,17
READ(4,100)VT5(I)
25 CONTINUE
DO 26 I=1,18
READ(4,100)VT6(I)
26 CONTINUE
DO 27 I=1,19
READ(4,100)VT7(I)
27 CONTINUE
DO 28 I=1,20
READ(4,100)VT8(I)
28 CONTINUE
DO 29 I=1,20
READ(4,100)VT9(I)
29 CONTINUE
DO 30 I=1,11
READ(4,100)VT10(I)
30 CONTINUE
```

```

DO 1100 I=1,16
1100 READ(4,100)VR1(I)
CONTINUE
DO 1200 I=1,16
1200 READ(4,100)VR2(I)
CONTINUE
DO 1300 I=1,17
1300 READ(4,100)VR3(I)
CONTINUE
DO 1400 I=1,17
1400 READ(4,100)VR4(I)
CONTINUE
DO 1500 I=1,17
1500 READ(4,100)VR5(I)
CONTINUE
DO 1600 I=1,18
1600 READ(4,100)VR6(I)
CONTINUE
DO 1700 I=1,19
1700 READ(4,100)VR7(I)
CONTINUE
DO 1800 I=1,20
1800 READ(4,100)VR8(I)
CONTINUE
DO 1900 I=1,20
1900 READ(4,100)VR9(I)
CONTINUE
DO 2000 I=1,11
2000 READ(4,100)VR10(I)
CONTINUE
READ(4,110)XC0
READ(4,110)XC02
READ(4,110)XH2
READ(4,120)FMASDD
READ(4,120)AMASDD
READ(4,110)FRAPRE
READ(4,110)FRAPRI
READ(4,110)FRASEC
READ(4,110)FRADIL
100 FORMAT(F7.2)
110 FORMAT(F7.5)
120 FORMAT(F7.3)
130 FORMAT(E10.5)
D1(1)=71.0
D1(2)=61.0
D1(3)=31.0
D1(4)=41.0
D1(5)=31.0
D1(6)=21.0
D1(7)=11.0
D1(8)=1.0
D1(9)=11.0
D1(10)=21.0
D1(11)=31.0

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32 D1(12)=41.0
D1(13)=51.0
D1(14)=61.0
D1(15)=71.0
D0 32 I=1.15
D2(I)=D1(I)
CONTINUE

D3(1)=77.6
D3(2)=67.6
D3(3)=57.6
D3(4)=47.5
D3(5)=37.6
D3(6)=27.6
D3(7)=17.6
D3(8)=7.6
D3(9)=7.6
D3(10)=17.6
D3(11)=27.6
D3(12)=37.6
D3(13)=47.6
D3(14)=57.6
D3(15)=67.6
D3(16)=77.6
D0 33 I=1.16
D4(I)=D3(I)
D5(I)=D3(I)

33

CONTINUE
D6(1)=83.9
D6(2)=73.9
D6(3)=63.9
D6(4)=53.9
D6(5)=43.9
D6(6)=33.9
D6(7)=23.9
D6(8)=13.9
D6(9)=3.9
D6(10)=13.9
D6(11)=23.9
D6(12)=33.9
D6(13)=43.9
D6(14)=53.9
D6(15)=63.9
D6(16)=73.9
D6(17)=83.9
D7(1)=87.3
D7(2)=77.3
D7(3)=67.3
D7(4)=57.3
D7(5)=47.3
D7(6)=37.3
D7(7)=27.3
D7(8)=17.3
D7(9)=7.3
D7(10)=7.3

```

D7(11)=17.3
D7(12)=27.3
D7(13)=37.3
D7(14)=47.3
D7(15)=57.3
D7(16)=67.3
D7(17)=77.3
D7(18)=87.3
D8(1)=96.6
D8(2)=86.6
D8(3)=76.6
D8(4)=66.6
D8(5)=56.6
D8(6)=46.6
D8(7)=36.6
D8(8)=26.6
D8(9)=16.6
D8(10)=6.6
D8(11)=16.6
D8(12)=26.6
D8(13)=36.6
D8(14)=46.6
D8(15)=56.6
D8(16)=66.6
D8(17)=76.6
D8(18)=86.6
D8(19)=96.6
DO 34 I=1,19
D9(I)=D8(I)
34 CONTINUE
D10(1)=46.0
D10(2)=36.0
D10(3)=26.0
D10(4)=16.0
D10(5)=6.0
D10(6)=6.0
D10(7)=16.0
D10(8)=26.0
D10(9)=36.0
D10(10)=46.0
DO 35 I=1,20
D1(I)=D1(I)/1000.0
D2(I)=D2(I)/1000.0
D3(I)=D3(I)/1000.0
D4(I)=D4(I)/1000.0
D5(I)=D5(I)/1000.0
D6(I)=D6(I)/1000.0
D7(I)=D7(I)/1000.0
D8(I)=D8(I)/1000.0
D9(I)=D9(I)/1000.0
D10(I)=D10(I)/1000.0
35 CONTINUE
RETURN
END

```



```

SUBROUTINE OUTPUT(T1,T2,T3,T4,T5,T6,T7,T8,T9,T10,VZ1,VZ2,VZ3,VZ4
2  ,VZ5,VZ6,VZ7,VZ8,VZ9,VZ10,VT1,VT2,VT3,VT4,VT5,VT6,VT7,VT8,VT9,
3  VT10,VR1,VR2,VR3,VR4,VR5,VR6,VR7,VR8,VR9,VR10)
C
C THIS SUBROUTINE PRINTS THE RESULTS OF THE VARIOUS CALCULATIONS AS WELL
C AS A TABLE OF THE TEMPERATURE AND, AXIAL AND CIRCUMFERENTIAL VELOCITIES
C
      VIRTUAL T1(25),T2(25),T3(25),T4(25),T5(25),T6(25),T7(25),T8(25)
2    ,T9(25),T10(25),VZ1(25),VZ2(25),VZ3(25),VZ4(25),VZ5(25),VZ6(25),
3    VZ7(25),VZ8(25),VZ9(25),VZ10(25),VT1(25),VT2(25),VT3(25),VT4(25)
4    ,VT5(25),VT6(25),VT7(25),VT8(25),VT9(25),VT10(25),VR1(25),
5    VR2(25),VR3(25),VR4(25),VR5(25),VR6(25),VR7(25),VR8(25),VR9(25),
6    VR10(25)
      COMMON/RCG1,RCG2,RCG3,RCG4,RCG5,RCG6,RCG7,RCG8,RCG9,RCG10
      COMMON/BLK1/AD1,AD2,AD3,AD4,AD5,AD6,AD7,AD8,AD9,AD10
      COMMON/BLK2/TOMAS1,TOMAS2,TOMAS3,TOMAS4,TOMAS5,TOMAS6,TOMAS7,
2    TOMAS8,TOMAS9,TOMAS0
      COMMON/BLK2/ANMOM1,ANMOM2,ANMOM3,ANMOM4,ANMOM5,ANMOM6,ANMOM7,
2    ANMOM8,ANMOM9,ANMOM0
      WRITE(6,100)
      WRITE(6,210)
      DO 200 I=1,16
      WRITE(6,110)T1(I),VZ1(I),VT1(I),VR1(I)
200    CONTINUE
      WRITE(6,101)
      WRITE(6,210)
      DO 201 I=1,16
      WRITE(6,110)T2(I),VZ2(I),VT2(I),VR2(I)
201    CONTINUE
      WRITE(6,102)
      WRITE(6,210)
      DO 202 I=1,17
      WRITE(6,110)T3(I),VZ3(I),VT3(I),VR3(I)
202    CONTINUE
      WRITE(6,103)
      WRITE(6,210)
      DO 203 I=1,17
      WRITE(6,110)T4(I),VZ4(I),VT4(I),VR4(I)
203    CONTINUE
      WRITE(6,104)
      WRITE(6,210)
      DO 204 I=1,17
      WRITE(6,110)T5(I),VZ5(I),VT5(I),VR5(I)
204    CONTINUE
      WRITE(6,105)
      WRITE(6,210)
      DO 205 I=1,18
      WRITE(6,110)T6(I),VZ6(I),VT6(I),VR6(I)
205    CONTINUE
      WRITE(6,106)
      WRITE(6,210)
      DO 206 I=1,19
      WRITE(6,110)T7(I),VZ7(I),VT7(I),VR7(I)
206    CONTINUE

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```

WRITE(6,107)
WRITE(6,210)
DO 207 I=1,20
WRITE(6,110)TB(I),UZ8(I),VT8(I),VR8(I)
207 CONTINUE
WRITE(6,108)
WRITE(6,210)
DO 208 I=1,20
WRITE(6,110)T9(I),UZ9(I),VT9(I),VR9(I)
208 CONTINUE
WRITE(6,109)
WRITE(6,210)
DO 209 I=1,11
WRITE(6,110)T10(I),UZ10(I),VT10(I),VR10(I)
209 CONTINUE
GO TO 500
WRITE(6,310)
WRITE(6,311)RCG1,RCG2,RCG3,RCG4,RCG5,RCG6,RCG7,RCG8,RCG9,RCG10
WRITE(6,312)
WRITE(6,311)AD1,AD2,AD3,AD4,AD5,AD6,AD7,AD8,AD9,AD10
WRITE(6,313)
WRITE(6,311)TOMAS1,TOMAS2,TOMAS3,TOMAS4,TOMAS5,TOMAS6,TOMAS7,
2 TOMAS8,TOMAS9,TOMAS10
WRITE(6,314)
WRITE(6,320)ANMOM1,ANMOM2,ANMOM3,ANMOM4,ANMOM5
WRITE(6,316)
WRITE(6,320)ANMOM6,ANMOM7,ANMOM8,ANMOM9,ANMOM10
500 CONTINUE
100 FORMAT(15X,'SEC. #1')
101 FORMAT(15X,'SEC. #2')
102 FORMAT(15X,'SEC. #3')
103 FORMAT(15X,'SEC. #4')
104 FORMAT(15X,'SEC. #5')
105 FORMAT(15X,'SEC. #6')
106 FORMAT(15X,'SEC. #7')
107 FORMAT(15X,'SEC. #8')
108 FORMAT(15X,'SEC. #9')
109 FORMAT(15X,'SEC. #10')
110 FORMAT(10X,4F10.4)
210 FORMAT(13X,'TEMP',8X,'UZ',9X,'VT',9X,'VR',/)
310 FORMAT(2X,'RCG1',3X,'RCG2',3X,'RCG3',3X,'RCG4',3X,'RCG5',3X,
2 'RCG6',3X,'RCG7',3X,'RCG8',3X,'RCG9',3X,'RCG10')
311 FORMAT(10F7.2,/)
312 FORMAT(3X,'AD1',4X,'AD2',4X,'AD3',4X,'AD4',4X,'AD5',4X,'AD6',
2 4X,'AD7',4X,'AD8',4X,'AD9',4X,'AD10')
313 FORMAT(1X,'TOMAS1',1X,'TOMAS2',1X,'TOMAS3',1X,'TOMAS4',1X,
2 'TOMAS5',1X,'TOMAS6',1X,'TOMAS7',1X,'TOMAS8',1X,'TOMAS9',1X,
3 'TOMAS10')
314 FORMAT(3X,'ANMOM1',6X,'ANMOM2',6X,'ANMOM3',6X,'ANMOM4',6X,
2 'ANMOM5')
316 FORMAT(3X,'ANMOM6',6X,'ANMOM7',6X,'ANMOM8',6X,'ANMOM9',6X,
3 'ANMOM10')
320 FORMAT(5E12.5)
RETURN
END

```

SUBROUTINE GASCON

REAL NZMD1,NZMD2,NZMD3,NZMD4,NZMD5,NZMD6,NZMD7,NZMD8,NZMD9

2 ,NZMD10

C

C THIS SUBROUTINE CALCULATES THE AVERAGE GAS CONSTANT AT THE TEN MEASURING
C SECTIONS OF THE COMBUSTOR

C

COMMON XCO,XCO2,XH2,FMASSD,AMASSD,FRAPRE,FRAPRI,FRASEC,FRADIL

COMMON/G/RCG1,RCG2,RCG3,RCG4,RCG5,RCG6,RCG7,RCG8,RCG9,RCG10

COMMON/BLK1/AD1,AD2,AD3,AD4,AD5,AD6,AD7,AD8,AD9,AD10

AD1=FMASSD*(FRAPRE+FRAPRI)*AMASSD

AD2=AD1+0.45715*FRASEC*AMASSD

AD3=AD2+0.54285*FRASEC*AMASSD

AD4=AD3

AD5=AD3

AD6=AD5+0.46341*FRADIL*AMASSD

AD7=AD6+0.46341*FRADIL*AMASSD

AD8=FMASSD*AMASSD

AD9=AD8

AD10=AD9

XMC0=28.0*XCO

XMC02=44.0*XCO2

XMH2=2.0*XH2

XMTOT=XMC0+XMC02+XMH2

GCO=XMC0/XMTOT

GCO2=XMC02/XMTOT

GH2=XH2/XMTOT

CO2MD=(44.0/28.0)*GCO*FMASSD+GCO2*FMASSD

H2OMD=9.0*GH2*FMASSD

O2MD1=0.233*(AD1-FMASSD)-16.0/28.0*GCO*FMASSD-8.0*GH2*FMASSD

NZMD1=0.767*(AD1-FMASSD)

SM1=CO2MD+H2OMD+O2MD1+NZMD1

GCO2=CO2MD/SM1

GH20=H2OMD/SM1

GO2=O2MD1/SM1

GN2=NZMD1/SM1

RCG1=(GCO2/44.0+GH20/18.0+GO2/32.0+GN2/28.0)*8314.7

O2MD2=0.233*(AD2-FMASSD)-16.0/28.0*GCO*FMASSD-8.0*GH2*FMASSD

NZMD2=0.767*(AD2-FMASSD)

SM2=CO2MD+H2OMD+O2MD2+NZMD2

GCO2=CO2MD/SM2

GH20=H2OMD/SM2

GO2=O2MD2/SM2

GN2=NZMD2/SM2

RCG2=(GCO2/44.0+GH20/18.0+GO2/32.0+GN2/28.0)*8314.7

O2MD3=0.233*(AD3-FMASSD)-16.0/28.0*GCO*FMASSD-8.0*GH2*FMASSD

NZMD3=0.767*(AD3-FMASSD)

SM3=CO2MD+H2OMD+O2MD3+NZMD3

GCO2=CO2MD/SM3

GH20=H2OMD/SM3

GO2=O2MD3/SM3

GN2=NZMD3/SM3

RCG3=(GCO2/44.0+GH20/18.0+GO2/32.0+GN2/28.0)*8314.7

O2MD4=0.233*(AD4-FMASSD)-16.0/28.0*GCO*FMASSD-8.0*GH2*FMASSD

$N2MD4 = 0.767 * (AD4 - FMASSD)$
 $SM4 = CQ2MD + H2QMD + Q2MD4 + N2MD4$
 $GCQ2 = CQ2MD / SM4$
 $GH2Q = H2QMD / SM4$
 $GQ2 = Q2MD4 / SM4$
 $GN2 = N2MD4 / SM4$
 $RCG4 = (GCQ2 / 44.0 + GH2Q / 18.0 + GQ2 / 32.0 + GN2 / 28.0) * 8314.7$
 $Q2MD5 = 0.233 * (AD5 - FMASSD) - 16.0 / 28.0 * GCQ * FMASSD - 8.0 * GH2 * FMASSD$
 $N2MD5 = 0.767 * (AD5 - FMASSD)$
 $SM5 = CQ2MD + H2QMD + Q2MD5 + N2MD5$
 $GCQ2 = CQ2MD / SM5$
 $GH2Q = H2QMD / SM5$
 $GQ2 = Q2MD5 / SM5$
 $GN2 = N2MD5 / SM5$
 $RCG5 = (GCQ2 / 44.0 + GH2Q / 18.0 + GQ2 / 32.0 + GN2 / 28.0) * 8314.7$
 $Q2MD6 = 0.233 * (AD6 - FMASSD) - 16.0 / 28.0 * GCQ * FMASSD - 8.0 * GH2 * FMASSD$
 $N2MD6 = 0.767 * (AD6 - FMASSD)$
 $SM6 = CQ2MD + H2QMD + Q2MD6 + N2MD6$
 $GCQ2 = CQ2MD / SM6$
 $GH2Q = H2QMD / SM6$
 $GQ2 = Q2MD6 / SM6$
 $GN2 = N2MD6 / SM6$
 $RCG6 = (GCQ2 / 44.0 + GH2Q / 18.0 + GQ2 / 32.0 + GN2 / 28.0) * 8314.7$
 $Q2MD7 = 0.233 * (AD7 - FMASSD) - 16.0 / 28.0 * GCQ * FMASSD - 8.0 * GH2 * FMASSD$
 $N2MD7 = 0.767 * (AD7 - FMASSD)$
 $SM7 = CQ2MD + H2QMD + Q2MD7 + N2MD7$
 $GCQ2 = CQ2MD / SM7$
 $GH2Q = H2QMD / SM7$
 $GQ2 = Q2MD7 / SM7$
 $GN2 = N2MD7 / SM7$
 $RCG7 = (GCQ2 / 44.0 + GH2Q / 18.0 + GQ2 / 32.0 + GN2 / 28.0) * 8314.7$
 $Q2MD8 = 0.233 * (AD8 - FMASSD) - 16.0 / 28.0 * GCQ * FMASSD - 8.0 * GH2 * FMASSD$
 $N2MD8 = 0.767 * (AD8 - FMASSD)$
 $SM8 = CQ2MD + H2QMD + Q2MD8 + N2MD8$
 $GCQ2 = CQ2MD / SM8$
 $GH2Q = H2QMD / SM8$
 $GQ2 = Q2MD8 / SM8$
 $GN2 = N2MD8 / SM8$
 $RCG8 = (GCQ2 / 44.0 + GH2Q / 18.0 + GQ2 / 32.0 + GN2 / 28.0) * 8314.7$
 $Q2MD9 = 0.233 * (AD9 - FMASSD) - 16.0 / 28.0 * GCQ * FMASSD - 8.0 * GH2 * FMASSD$
 $N2MD9 = 0.767 * (AD9 - FMASSD)$
 $SM9 = CQ2MD + H2QMD + Q2MD9 + N2MD9$
 $GCQ2 = CQ2MD / SM9$
 $GH2Q = H2QMD / SM9$
 $GQ2 = Q2MD9 / SM9$
 $GN2 = N2MD9 / SM9$
 $RCG9 = (GCQ2 / 44.0 + GH2Q / 18.0 + GQ2 / 32.0 + GN2 / 28.0) * 8314.7$
 $Q2MD10 = 0.233 * (AD10 - FMASSD) - 16.0 / 28.0 * GCQ * FMASSD - 8.0 * GH2 * FMASSD$
 $N2MD10 = 0.767 * (AD10 - FMASSD)$
 $SM10 = CQ2MD + H2QMD + Q2MD10 + N2MD10$
 $GCQ2 = CQ2MD / SM10$
 $GH2Q = H2QMD / SM10$
 $GQ2 = Q2MD10 / SM10$
 $GN2 = N2MD10 / SM10$

RCG10*(GC02/44.0+GM20/18.0+G02/32.0+GM2/28.0)*8314.7
RETURN
END

END

2	8	H	1	0	H	8	5
---	---	---	---	---	---	---	---

FIN