

Military Technical College
Kobry El-Kobba,
Cairo, Egypt



11-th International Conference
on Aerospace Sciences &
Aviation Technology

THE GAS PRESSURE INFLUENCE ON COMBUSTION CHAMBER FLOW CHARACTERISTICS

Ola Rashed¹

ABSTRACT

The atomization process influences combustion. The Sauter Mean Diameter (SMD), the pressure, and the temperature of the combustor play an important role in the progress of the combustion. The objective of this study is to develop a mathematical model to evaluate the effect of the gas pressure on the combustion chamber flow characteristics. Based on this model a computer program has been designed taking into account the effects of droplet internal circulation, change in combustion chamber gas temperature, the non-unitary Lewis number, the changing gas film thickness surrounding the droplet, the slip, the variable thermo-physical properties. The effect of gas pressure on Liquid Reynolds number, gas Reynolds number, non-dimensional radii, Lewis number, droplet velocity and droplet surface temperature is established. Increasing the gas pressure increases the vapor mass fraction, which affects all other parameters (thermal conductivity, diffusivity, specific heat of liquid and gas, mass and heat transfer number) and leads to a decrease in combustion time. The droplet surface is found to reach its maximum temperature in a shorter time if the pressure is higher. By increasing the gas pressure the droplet-evaporating rate increases, which leads to a decrease in evaporating time. Consequently the combustion chamber volume is decreased. Iterations on the chamber dimensions are done until the program converges.

KEYWORDS: combustion, combustion chamber, gas pressure, combustion modeling, atomization, atomization modeling.

¹ Assistant Professor, Aerospace Department, Cairo University

NOMENCLATURE

A	Droplet surface area	t	Time
B_M	Mass transfer number	t_r	Ratio of convective time scale to diffusive time scale in gas-phase
B_T	Heat transfer number	T	Temperature
C_D	Drag coefficient	T_B	Boiling temperature
C_l	Specific heat of liquid	T_l	Liquid drop temperature
C_p	Specific heat at const pressure	u	Gas-phase velocity
C_v	Specific heat at const volume	v	Droplet velocity
d_k	Instantaneous diameter of k^{th} droplet	x	Droplet position
d_o	Initial droplet diameter	Y	Mass fraction
D_c	Diffusion coefficient	$\delta(x)$	Dirac delta function
D_k	Drag function	ΔV	Cell control volume
h	Enthalpy	μ	Dynamic viscosity
k	Thermal conductivity	ν	Kinematic viscosity
L	Latent heat of vaporization	ρ	Density
L_e	Lewis number	τ	Non-dimensional time
L_r	Ratio of gas-phase length scale and initial droplet radius	<u>Subscripts</u>	
L_c	Gas-phase length scale	a	air
m_k	Mass of k^{th} droplet	g	gas
\dot{m}_k	Mass evaporation rate of k^{th} droplet	l	liquid
Nu	Nusselt number	k	k^{th} droplet
p	Pressure	v	vapor
Pr	Prandtl number	o	initial
Q_g	Energy transferred from gas-phase to droplet	∞	Unperturbed conditions
Q_l	Liquid heating energy		
r_k	Radius of k^{th} droplet		
R	Ideal gas constant		
Re	Reynolds number		
Re_o	Initial Reynolds number		
S_e	Energy exchange term between two phases		
S_m	Mass exchange term between two phases		
S_M	Momentum exchange term between two phases		

INTRODUCTION

In the last few years, spray models have been coupled with different flow algorithms [1,2,3,4,5]. The obtained results depended strongly on the complexity of these algorithms, and needed extensive computer storage and capabilities. Thus developing an algorithm, which handles the spray problem in an economic and accurate manner, was the main goal of the present work. The primary objective of this work is to develop a realistic model for spray evaporation in a combustor flow of continuously changing temperature and velocity fields so that the gas pressure effect on the combustion characteristics is determined.

In the past decade numerous spray models have been proposed for different spray combustion problems [6,7]. Recent models are generally divided into two categories: the locally homogeneous flow models (LHF), and the separated flow models (SF). LHF models [8] represent the simplest treatment of a multiphase flow. The main drawback of this model is its key assumption, which implies that the inter-phase transport rates are fast in comparison to the flow development rate. This is only correct for flows containing infinitely small droplets. The inter-phase transport phenomena have been incorporated in the SF models [9,10]. In this work the SF approach has been extended to include the variation of the combustion chamber temperature, the existence of liquid spray ligaments and droplets with a wide diameter spectrum.

In this paper Kerosene injected with airblast atomizers in a combustion chamber of variable cross section is considered. The initial combustion chamber shape is chosen according to Mellor [16]. Then iterations on the shape is to be performed until compatibility with the obtained parameters is reached. The aim of this paper is to determine the gas pressure influence on the different flow characteristics, so that preliminary designs of combustors can be achieved. Upon injection, the injected ligaments start to break up into droplets. These droplets accelerate; the gas quickly adjusts to their presence and surrounds each one by a boundary layer. Internal circulation within the droplet is soon established. Fuel droplets and hot air enriched with fuel vapor are moving from left to right with the gas moving at a greater speed. The gas motion is primarily caused by its expansion. The continuous transfer of momentum results in a reduction of the relative velocity between the gas and the droplets, causing a drop in the relative droplet Reynolds number. The surrounding gas supplies energy, which heats the droplet interior until the vaporization is initiated.

The smaller droplets are seen to vaporize more quickly and to be dragged by gas motion more readily. Vaporization occurs so that the fuel-vapor mass fraction increases with time and with down-stream distance from the entry point. As fuel vapor is created, it diffuses away from the droplet in all directions. The gradual decay of fuel vapor mass fraction in the direction of the flow occurs since the gas velocity is greater than the droplet velocity. The equations governing this problem include the continuity, momentum, energy, and species equations for the gas and liquid phases. The equation of state is taken into account also. Strong coupling exists between these non-linear equations because of the existence of inter-phase friction, and inter-phase transport modes at the droplet-gas interface. So an iterative technique is employed to solve these equations.

MATHEMATICAL FORMULATION

The model considers a spray of liquid droplets with interfacial exchange transport exposed to forced convective gas flow. It is assumed that near the exit of the atomizer, the flow is in the form of ligaments surrounded by hot gases. Then after an increment of time these ligaments reach a certain distance, where they start to break up and form droplets possessing internal circulation. By knowing the fuel properties, air properties and the atomizer dimensions this distance is determined from the graphs developed by Lefevbre[11] and equals $x=0.5 D_p$ in the present case. Thus the computational domain starts from $x= 0.5 D_p$ until the droplet diminishes. Exposed to drag, shear forces, and to uneven heating to its surface the droplet surrounded by thermal and mass boundary layers, gains velocity during evaporation. The liquid droplets consist of a single species, namely Kerosene. The gas phase consists of two species one from the liquid fuel droplet evaporation and the other from the ambient flow. The liquid- phase, gas -phase and droplet continuity, momentum, energy and species equations as well as the heat and mass transfer equations, in addition to the state equation, therefore govern the flow.

Due to the presence of the relative motion between the gases and the droplets, the existing shear stress and the heat and mass flux at the interface, the liquid - phase and gas phase equations are coupled and require an iterative procedure to solve them simultaneously.

GOVERNING EQUATIONS

Gas-Phase Equations

The droplets are injected in a gaseous medium, so they are considered to be sources of mass and momentum, as well as sinks of energy. The gas phase governing equations describing the model can be written in general as following

$$\frac{\partial \rho_g}{\partial t} + \nabla(\rho_g v_g) = S_m \quad (1)$$

$$\frac{\partial(\rho_g v_g)}{\partial t} + \nabla(\rho_g v_g v_g) = \nabla(\rho_g v_g \nabla v_g) - \nabla(p) + S_M \quad (2)$$

$$\frac{\partial(\rho_g h_g)}{\partial t} + \nabla(\rho_g v_g h_g) = \nabla(k_g \nabla T_g) + S_e \quad (3)$$

$$\frac{\partial(\rho_g Y_v)}{\partial t} + \nabla(v_g \rho_g Y_v) = \nabla(\rho_g D_c \nabla Y_v) + S_m \quad (4)$$

where p , ρ , v , T , h , and t are the pressure, density, velocity, temperature, enthalpy and time respectively. D , k , ν are the diffusivity, the thermal conductivity, and the kinematic viscosity. The vapor-mass fraction Y_v is defined as $Y_v = \frac{\rho_v}{\rho_g}$, and the

subscript g denotes the gaseous phase.

The quantities S_m , S_M , and S_e represent the mass, momentum, and energy exchange terms, respectively. The ideal-gas equation of state is also adopted.

$$P = \rho_g \frac{[R_a + Y_v(R_v - R_a)]}{1/T_g} \tag{5}$$

where R denotes the gas constant, subscripts a is for air and v for vapor.

The gas-phase governing equations are solved to obtain the gas flow characteristics, by using a 2nd order Runge-Kutta technique for the equation of motion and a numerical explicit finite difference scheme for the species and momentum equations. The liquid-phase solution is used as input to this scheme and then iteration is performed to insure the correctness of the obtained results.

Liquid - Phase Equations

Droplet Motion Equations

Each droplet, labeled by a subscript k is assumed to obey the following equations:

$$\frac{dx}{dt} = V_k \tag{6}$$

$$\frac{dV_k}{dt} = \rho_r L_r^2 t_r \left(\frac{3 C_D \mu \text{Re}_k (V - V_k)}{16 r_k^2} \right) \tag{7}$$

$$\frac{dr_k^2}{dt} = -2 \rho_r L_r^2 t_r M_k \tag{8}$$

where $\rho_r = \rho_{\text{gas initial}} / \rho_{\text{liquid initial}}$, L_r is the gas-phase length scale divided by the initial drop radius, t_r is the convective by the diffusive time scale.

M_k represents the non-dimensional vaporization rate where the term multiplying the logarithmic term represents the correction factor modifying the spherically-

symmetric results obtained from the heat transfer equation solution and is defined as follows

$$M_k = \left(1 + 0.3 \text{Re}_k^{\frac{1}{2}} \right) \ln(1 + B_M) \quad \text{with} \quad B_M = \frac{Y_{fs} - Y_{fo}}{1 - Y_{fs}}$$

B_M is the mass transfer number (referred to as Spalding number), and Y_{fs} is the fuel vapor mass fraction at the droplet surface, $Y_{f\infty}$ is the mean fuel mass fraction of the ambient gas. According to Feath [6] and Sirignano [7], the particle Reynolds number Re_k and the drag coefficient C_D are defined as follows

$$C_D = \frac{24}{Re_k} \left(1 + \frac{Re_k^{\frac{2}{3}}}{6} \right) \quad \text{and} \quad Re_k = \frac{2r_k \rho}{t_r L_r \mu} |V - V_k|$$

In order to obtain the velocity of the droplets, the distance traveled, the droplet radii, these governing equations are solved using a 2nd order Runge- Kutta Scheme.

The obtained results are then used as input for the gas phase problem, and the iterative procedure is performed until convergence is reached.

Droplet Mass & Heat Transfer Equations

As described by Feath [6], the droplet heat-up and evaporation calculations use the following correlations to obtain the mass & heat transfer rates

$$\frac{\dot{m}_k d}{\rho D_f} = 2N_s \ln(1 + B_M) \tag{9}$$

$$\frac{h_r d}{k} = 2N_p \frac{\ln(1 + B_M)^{Le-1}}{((1 + B_M)^{Le-1} - 1)} \tag{10}$$

where h_r , d , D_f , k , and \dot{m}_k are the heat transfer coefficient, the droplet diameter, the fuel-mass diffusivity, thermal conductivity and mass evaporation rate per unit area. The N_s and N_p are defined as:

$$N_s = 1 + \frac{0.276 Re^{\frac{1}{2}} Pr^{\frac{1}{3}}}{\left(1 + \frac{1.232}{Re(Pr^{\frac{4}{3}})} \right)^{\frac{1}{2}}} \quad \text{and} \quad N_p = 1 + \frac{0.276 Re^{\frac{1}{2}} Sc^{\frac{1}{3}}}{\left(1 + \frac{1.232}{Re(Sc^{\frac{4}{3}})} \right)^{\frac{1}{2}}}$$

Sc , Pr and Le are the Schmidt, Prandtl and Lewis numbers.

In order to evaluate the mass and the thermal boundary layer thickness δ_T and δ_m , the following relations are used

$$\delta_m = F_M * \delta_{Mo} \tag{11}$$

$$\delta_T = F_T * \delta_{To} \tag{12}$$

F_M and F_T are factors used to account for the Stefan flow effect on the heat and mass transfer and are defined as follows:

$$F_M = (1 + B_M)^{0.7} \frac{\ln B_M}{B_M} \quad \text{and} \quad F_T = (1 + B_T)^{0.7} \frac{\ln B_T}{B_T}$$

and

$$\delta_{Mo} = \frac{2r_s}{(Nu_o - 2)} \quad \text{and} \quad \delta_{To} = \frac{2r_s}{(sh_o - 2)}$$

with $B_T = \frac{c_p(T_\infty - T_K)}{L}$ where L is the latent heat of evaporation

the Nusselt and Sherwood numbers at the initial condition (o) are obtained as follows

$$Nu_o = 2 + 0.552 Re^{\frac{1}{2}} Pr^{\frac{1}{3}} \quad \text{and} \quad sh_o = 2 + 0.552 Re^{\frac{1}{2}} Sc^{\frac{1}{3}}$$

Droplet Internal Temperature Equations

The temperature distribution within the droplet is obtained by solving the energy equation subject to the convective boundary conditions at the droplet surface by a Crank- Niclson scheme.

$$\frac{\partial T_1}{\partial t_b} = \frac{\partial^2 T_1}{\partial \tilde{r}^2} + \left(\frac{2}{\tilde{r}} - \tilde{r} M_k K \right) \frac{\partial T_1}{\partial \tilde{r}} \quad (13)$$

with the following initial and boundary condition

$$T_1(\tilde{r}, 0) = 0$$

$$\frac{\partial T_1}{\partial \tilde{r}} = \frac{1}{k} [h(T_{gas} - T_{k-surface}) - \dot{m}_k L]$$

Where the non-dimensional radial distance \tilde{r} , time t_b , and liquid temperature T_1

are defined, respectively, as : $\tilde{r} = \frac{r}{r_{kinitial}}$, $t_b = \alpha \int_0^t \frac{dt}{r_k^2}$, and $T_1 = \frac{T_1^l - T_o^l}{T_B^l - T_o^l}$

T_1^l , T_o^l , T_B^l , r_k , are the droplet temperature, the initial temperature, the boiling temperature, and the radial distance within the droplet respectively. L is the latent

heat of evaporation h, k, \dot{m}_k are obtained from the heat transfer equations.

K is defined as

$$K = \frac{(\rho D)_{initial-gas-phase}}{(\alpha_l \rho_k)_{initial-liquid-phase}} \quad \alpha_l = \text{liquid - thermal - diffusivity}$$

The choice of the increments used in the Partial Differential Equations is of crucial importance to the solution; smaller increments tend to give accurate results. Staggered meshes are applied in the physical domain to facilitate the discretization schemes. The increments yielding good results were 0.001mm for distance, and ranges between 0.001 ms and 0.0001 ms for time.

It is worth mentioning that:

The average physical properties (T , C_{p_g} , μ_g , K_g , D , C_{p_f} , ρ) are calculated by the following 1/3 rule [12]

$$\Phi = \Phi_s + 1/3 (\Phi_g - \Phi_s)$$

The following input parameters are chosen as follows:

Atomizer input conditions $P_a = 1$ atm, $U_a = 92$ m/s, $T_a = 300^\circ\text{K}$, $D_p = 38.1$ mm
 $SMD_{\text{initial}} = 64.9$ μm and updated in the solution according to:

$$SMD = (0.073 \left(\frac{\sigma_L}{\rho_A U^2} \right)^{0.6} \left(\frac{\rho_L}{\rho_A} \right)^{0.1} D_p^{0.4} + 0.15 \left(\frac{\mu_L^2 D_p}{\sigma_L \rho_L} \right)^{0.5}$$

Fuel properties: Liquid/air ratio = 0.25
 $\rho = 752.3$ kg/m³, $\sigma = 24e^{-3}$ N/m,
 $\mu = 0.00129$ kg/ms

RESULTS

The effect of the gas pressure on the combustion characteristics has been evaluated considering the case of a tubular combustion chamber. For the different gas pressures the variation of the non-dimensional radii with time is shown in Fig.1. As expected the increase in gas pressure increases the regression rate, which leads to a decrease in the evaporating time. But this favorable gas pressure increase is limited by the combustion process itself, because beyond a certain range this increase will lead to instantaneous droplet evaporation, which will cause combustion instabilities.

Fig.2 shows the gas pressure effect on the evolution of the surface temperature. As seen from the figure the change in droplet surface temperature is not drastically affected by the increases of the gas pressure. The decrease of droplet evaporating time noticed for the increased gas pressure is due to its increased vapor formation.

Fig.3 depicts the slow respond of the droplet interior to heating. The droplet interior doesn't experience its main temperature rise before the surface has almost reached the boiling temperature.

The Lewis number versus time at different gas pressures is displayed in Fig 4.

The Lewis number decreases rapidly until it reaches a certain value, then it levels off. The decrease is due to the variation of density, specific heat, thermal conductivity and diffusivity, which happens within the first few milli-seconds. Since the slope of the curves for different pressures is almost constant, the pressure increase has affected the Lewis number in a minor way.

Fig.5 shows the liquid Reynolds number versus time for different gas pressures. The liquid Reynolds number though starting with same initial value differs

depending on the droplet diameter evolution, for each pressure case. For increased gas pressure the liquid Reynolds number increases due to heat and mass transfer and then decreases quickly which indicates the diminishing droplet diameter. While for low gas pressures the rise in Reynolds number takes a longer period and the fall is slower because the droplets consume more time in evaporation.

Changing the gas pressure on the other hand did not affect the gas Reynolds number drastically, as depicted in Fig.6

As seen from the results, the increase of the gas pressure can enhance the combustion process. However this increase is restricted by the combustion instabilities that may occur at the higher pressures. At these pressures the evaporation process tend to consume less time causing instantaneous vaporization and consequently instability of the combustion process.

Comparisons with the model developed by Chiang [13,14] are seen in Fig.7. In spite of the small difference, the present model is more accurate for it took into consideration the varying combustion chamber temperature, the droplet internal circulation, which Chiang ignored. As reflected in Fig.8 the present work results are also in good agreement with the data obtained by Abramazon and Sirignano [15], which proves the validity of the developed model.

CONCLUSIONS

The increase in gas pressure leads to an increase in the droplet regression rate, which in turn leads to the decrease in evaporating time, and consequently the combustion chamber size decreases. Iterations on chamber size are done by running the developed program until compatibility with flow characteristics is reached. Thus the need for a tool to provide for the appropriate estimate for the combustion chamber size is satisfied in this work.

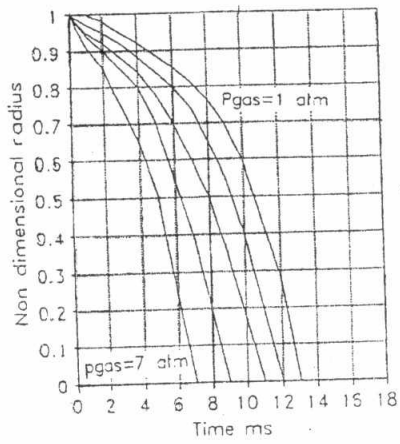


Fig.1. Drop Radius vs. Time

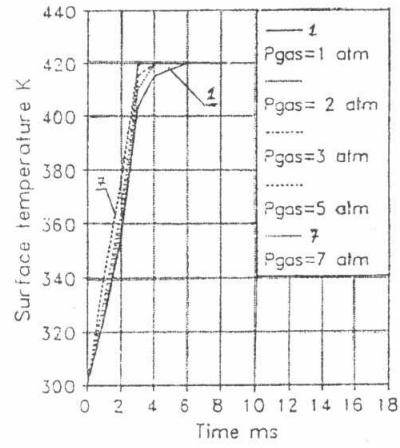


Fig.2. Drop Surface Temp. vs. Time

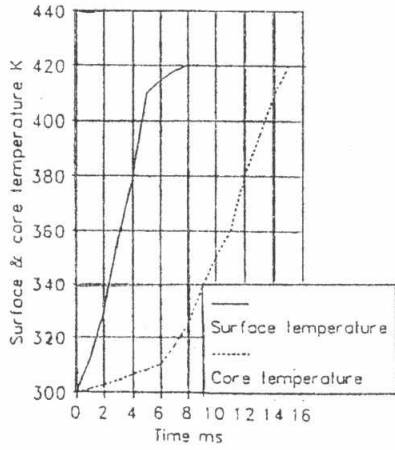


Fig.3. Drop Core Temp. vs. Time

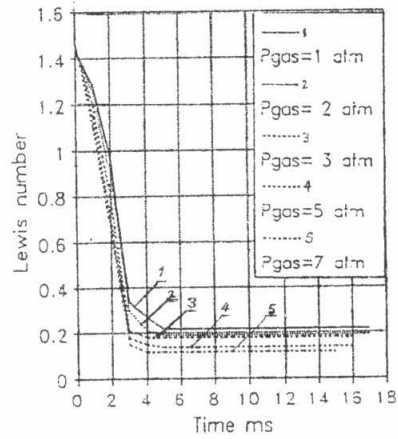


Fig.4. Lewis Number vs. Time

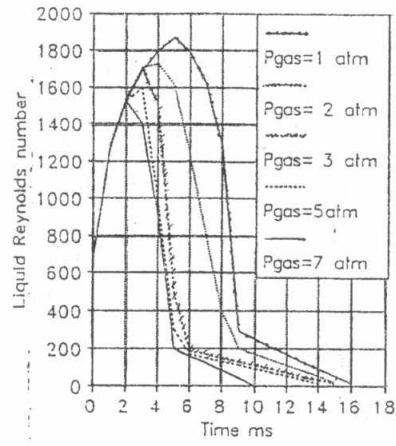


Fig. 5. Liquid Re Number vs. Time

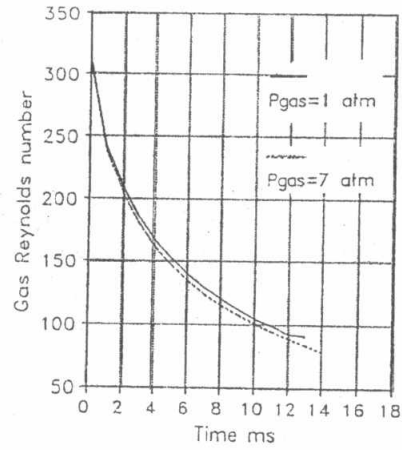


Fig. 6. Gas Re Number vs. Time

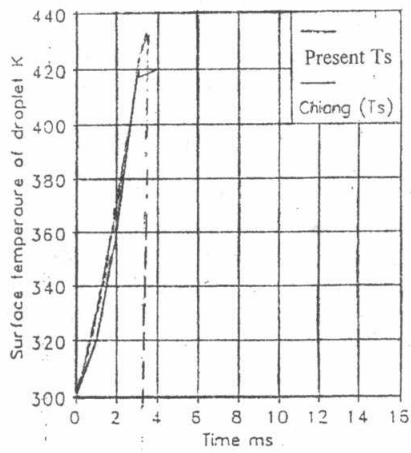


Fig. 7. Drop Surface Temp. Comparison

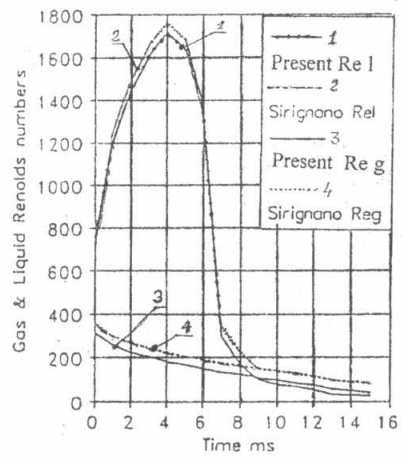


Fig. 8. Reynolds Number Comparison

REFERENCES

- [1] Shuen, J.S., Chen, K.H. and Choi, Y., "A Time Accurate Algorithm for Chemical Non-Equilibrium Viscous Flows at All Speeds", *AIAA paper 92-3639*, American Institute for Aeronautics and Astronautics, New York, N.Y., (1992).
- [2] Celik, C., Arisoy, A., and Genceli, O., "Modeling of Turbulent Reacting Flows with Gaseous Fuels in a Combustion Chamber", *Fuel*, Vol. 75, No. 4, p. 431, (1996).
- [3] Solomon, A.S.P., Shuen, J.S., Zhang, Q.F. and Faeth, G.M., "Measurement and Predictions of the Structure of Evaporating Sprays", *J. of Heat Tran.*, Vol. 107, p. 679, (1985).
- [4] Shuen, J.S., "Prediction of the Structure of Fuel Sprays in Cylindrical Combustion Chambers", *J. of Propulsion*, Vol. 3, p. 105, (1987).
- [5] Raju, M.S. and Sirignano, W.A., "Multi-component Spray Computations in a modified Center-body Combustor", *J. of Propulsion*, Vol. 6, p. 97, (1990).
- [6] Faeth, G.M., "Evaporation and Combustion of Sprays", *Prog. In Energy and Comb. Sci.*, Vol. 9, p.1, (1983).
- [7] Sirignano, W.A., "Fuel Vaporization and Spray Combustion Theory", *Prog. in Energy and Comb. Sci.*, Vol. 9, p. 291, (1983).
- [8] Shearer, A.J., Tamura, H. and Faeth, G.M., "Evaluation of a Locally Homogeneous Flow Model of Spray Evaporation", *J. of Energy*, Vol. 3, p. 271, (1979).
- [9] Gosman, A.D. and Johns, R.J.R., "Computer Analysis of Fuel-Air Mixing in Direct-Injection Engines", *Diesel Comb. And Emission*, SEA P-86, SAE, Warrendale, PA., (1980).
- [10] Crow, C.T., "Review-Numerical Models for Dilute Gas-Particle Flows," *J. of Fluid Eng.*, Vol.104, p. 297, (1982).
- [11] Lefebvre, A.H., "Airblast Atomization", *Prog. in Energy Combustion Sci*, Vol. 6, p. 233, (1980).
- [12] Sparrow, E.M., and Gregg, J.L., "The Variable Fluid Property problem in free Convection," *Trans. Am. Soc. Engrs*, Vol. 80, p. 879, (1958)
- [13] Chiang, C.H., "Isolated and interacting, vaporizing fuel Droplet: Field calculation with variables Properties", *Ph.D. Dissertation*, Dept. of Mechanical Engineering, University of California, Irvine, CA., (1990).
- [14] Chiang, C.H., Raju, M.S., and Sirignano, W.A., "Numerical Analysis of Convecting, Vaporization Fuel Droplet with Variable Properties," *International Journal of Heat and mass Transfer*, Vol.35, No.5, p.1307, (1992).
- [15] Abramzon, B and Sirignano, W.A., "Droplet vaporization Model for Spray combustion Calculations", *International J. Heat and Mass Transfer*, Vol. 32, No. 9, p. 1605, (1989).
- [16] Mellor, A.M., *Design of Modern Turbine Combustors*, Academic Press, San Diego, (1990).

THERMAL SCIENCES

CONTENTS

TH-01	THEORETICAL INVESTIGATION OF AN ADIABATIC CAPILLARY TUBE WORKING WITH PROPANE / n-BUTANE / iso-BUTANE BLENDS M. Fatouh	493
TH-02	ASSESSMENT OF PROPANE/COMMERCIAL BUTANE MIXTURES AS POSSIBLE ALTERNATIVES TO R134a IN DOMESTIC REFRIGERATORS M. Fatouh , M. El Kafafy	513
TH-03	SIMULATION OF SATELLITE HEAT PIPE RADIATOR Ahmed Fathy, Mohamed Afify, Karam El-Shazly	529
TH-04	PERFORMANCE OF WICKLESS HEAT PIPE WORKING WITH R134a Ahmed Fathy, Mohamed Afify, Karam El-Shazly	553
TH-05	THE EFFECT OF USING ARGON IN THE INTAKE CHARGE ON THE PERFORMANCE OF A DIESEL ENGINE UNDER SYNTHETIC ATMOSPHERE CONDITIONS W. M. Nessim, M.H. Khobeiz., M. A. Shahin	573
