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ATOMIZER AIR VELOCITY EFFECT ON MULTI-COMPONENT DROP EVAPORATION AND FLOW CHARACTERISTICS

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ABSTRACT

A computer-aided approach for the evaluation of the atomizer air velocity effect on the combustion chamber flow characteristics is described. This work considers an Airblast atomizer and models the problem by developing a finite difference scheme to solve the equations governing the combustion chamber flow. This presently developed model takes into account the effects of the variable thermo-physical properties, the gas film non-unitary Lewis number, the Stefan flow, the transient liquid heating, internal circulation, and the changes of combustion chamber gas temperature. The air velocity effect on liquid Reynolds number, gas Reynolds number, the regression rate, the droplet velocity, the droplet surface temperature, and Lewis number is determined by examining air velocities ranging from 50 to 190 m/s. Air velocity increase results in a decrease in Sauter mean diameter (SMD) and evaporating time. The tables and figures resulting from this work will help in the early design stages of the combustion chamber. Comparing the present results to other theoretical and experimental results indicate that this research provides an efficient tool for the evaluation of the air velocity effect on the combustion chamber flow characteristics.

KEYWORDS:

drop evaporation, combustion characteristics, atomizer air velocity, combustion chamber, and atomizer.

1. INTRODUCTION

The theory of fuel droplet vaporization and spray combustion has been under development for several decades. During recent years, it received a large amount of attention. Previously, the classical droplet vaporization theory treated spherically-symmetric, quasi-steady, single-component isolated droplets. In the past decades numerous spray models have been proposed for different spray combustion problems. Kuo, Lefebvre [1,2] studied sprays in real combustors where the local field around a droplet is not spherically-symmetric but multi-dimensional and fuels are multi-compositional with varying volatility. Feath (1983) ref. [3] and Sirignano (1983) ref. [4] were among the most famous who have investigated the droplet vaporization problem. Our knowledge to date on droplet vaporization and spray combustion is summarized in their review articles ref [3,4], in which they addressed and discussed a variety of spray combustion issues. Generally recent spray models are divided into two categories, the locally homogeneous flows (LHF) and the separated flow models (SF). LHF

models represent the simplest treatment of a multiphase flow and were widely used to analyze sprays ref. [5]. Numerous SF models have been proposed to consider the complex inter-phase transport phenomena, the multiple droplet size and the coupled phase properties ref. [6,7,8].

Nowadays most of the liquid fuels utilized are mixtures of many compounds with wide variation in properties. Many differences between the single vaporization and multi-component models exist. The main difference in the vaporization and combustion characteristics between these two models can be attributed to the transient droplet transport effects. For the multi-component case, the transient thermal and liquid mass transport is usually rate controlling, and must be included in the model ref [9-12]. The micro-explosion phenomenon, rising from the volatile component trapping inside the droplet, presents another important difference. This phenomenon used to enhance the combustion of heavy fuels cannot be explained by the single component theory ref [12]. Of course the droplet dynamics itself is affected by the presence of multi-components.

Droplet vaporization research is now a highly developed science ref (13-16), in which a large amount of time and resources have been invested in the account of the droplet vaporization modeling.

2. OBJECTIVE OF THE PRESENT WORK

The primary objective of this work is to study the atomizer air velocity effect on the combustion chamber flow characteristics. The atomizer governing equations are solved to determine the most appropriate atomizer characteristics for the flow condition considered. Then a separated flow model is applied, which solves the equations governing the flow emerging from an airblast atomizer using dodecane (Kerosene) as single fuel and a mixture of decane-dodecane for the multi-component case.

This model takes into account the interfacial surface tension and the drop exposure to forced convective gas flow, drag, shear forces and to uneven heating to its surface. It is assumed that the injected fuel ligaments start to break up into fully developed spherical droplets at certain distance from the nozzle. This distance varies according to operating parameters and is obtained according to graphs developed by Lefebvre ref (2). The gas adjusts to droplet presence by forming a boundary layer around each one. Due to internal circulation, these droplets accelerate. Continuous momentum transfer reduces the relative motion between the gas and droplets, causing a drop in the relative droplet Reynolds number. Heat transfer tends to trigger the change of droplet interior temperature and consequently vaporization starts. This vaporization process starts initially slow, but accelerates later, due to the rise in droplet temperature, drop in pressure caused by accelerating particles, and other reasons.

The governing equations are solved to obtain the characteristics of the prefilming atomizer and the flow characteristics inside the combustion chamber. 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. Thus to solve the equations efficiently an iterative technique is employed, to tackle simultaneously the gas phase, liquid phase and droplet equations using a Runge-Kutta scheme and an implicit finite element technique in a Lagrangian approach. The effect of variable thermo-physical properties, gas film, non unitary Lewis number, the Stefan flow influence on heat and mass transfer, droplet-gas heat exchange, and transient liquid heating were taken into consideration. The variation of the combustion chamber temperature though complicating the algorithm, and therefore usually ignored by researchers is incorporated in this work. Due to the excessive amount of computer time needed for group droplet analysis in the advanced numerical models, the major contribution resulting from this work is the development of a new approximate droplet vaporization and spray model under such complex circumstances.

3. GOVERNING EQUATIONS

3.1- Atomizer Equations

Referring to Nuyama and Tanasawa ref [18], a relative simple mathematical equation that adequately describes the droplet distribution is used

$$\frac{dN}{dD} = aD^p \exp(-(bD)^q) \quad (1)$$

where N denotes the number of droplets, D droplet diameter, and a,b,p,q are constants dependent on the atomizer geometry and operating conditions and are obtained referring to Lefebvre ref[2], and El Kotb ref [19]. The distribution function is thus obtained and the sauter mean diameter is calculated for each case according to

$$SMD = \frac{\sum N_i D_i^3}{\sum N_i D_i^2} \quad (2)$$

3.2- 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 generally written as following

$$\frac{\partial \rho_g}{\partial t} + \nabla(\rho_g v_g) = S_m \quad (3)$$

$$\frac{\partial(\rho_g v_g)}{\partial t} + \nabla(\rho_g v_g v_g) = \nabla(\rho_g v_g v_g) - \nabla p + S_M \quad (4)$$

$$\frac{\partial(\rho_g h_g)}{\partial t} + \nabla(\rho_g v_g h_g) = \nabla(k_g \nabla T_g) + S_e \quad (5)$$

$$\frac{\partial(\rho_g Y_v)}{\partial t} + \nabla(v_g \rho_g Y_v) = \nabla(\rho_g D_c \nabla Y_v) + S_m \quad (6)$$

where p , ρ , v , T , and t are the pressure, density, velocity, temperature and time respectively. The

vapor-mass fraction Y_v is defined as $Y_v = \frac{\rho_v}{\rho_g}$

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.

$$P = \rho_g (R_a + Y_v (R_v - R_a)) T_g \quad (7)$$

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.

3.3- Liquid - Phase Equations

3.3.1 Droplet Motion Equations

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

$$\frac{dx}{dt} = V_k \quad (8)$$

$$\frac{dV_k}{dt} = \rho_r L_r^2 t_r \left(\frac{3 C_D \mu \text{Re}_k (V - V_k)}{16 r_k^2} \right) \quad (9)$$

$$\frac{dr_k^2}{dt} = -2 \rho_r L_r^2 t_r M_k \quad (10)$$

where the M_k is obtained from the heat transfer equation solution and is defined as follows

$$M_k = \left(1 + 0.3 \text{Re}_k^{\frac{1}{2}} \right) \ln(1 + B_M) \quad , \quad B_M = \frac{Y_{v,k} - Y_{v,\infty}}{1 - Y_{v,k}}$$

B_M is the mass transfer number (referred to as Spalding number).

$\rho_r, L_r, t_r, \text{Re}_k, C_D$ are the gas density ratio, the gas-phase length scale, and the convective to diffusive time scale ratio, the particle Reynolds number and the drag coefficient respectively, and are defined in ref [3,8,14].

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.

3.3.2 Droplet Mass & Heat Transfer Equations

As described by Feath ref (3), the droplet heat-up and evaporation calculations use the following correlations to obtain the mass & heat transfer rates

$$\frac{\dot{m}_k d}{\rho D_f} = 2 N_s \ln(1 + B_M) \quad (11)$$

$$\frac{hd}{k} = 2 N_p \frac{\ln(1 + B_M)^{Le-1}}{\left((1 + B_M)^{Le-1} - 1 \right)} \quad (12)$$

where $h, d, D_f, k,$ 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 in ref [3].

In order to evaluate the mass and the thermal boundary layer thickness the effect of the Stefan flow on the heat and mass transfer are accounted for by a correction factor F on both thicknesses as described in ref [3]

Then the thermal and boundary film thickness δ_T and δ_m are evaluated by using the following relations:

$$\delta_m = F_M \delta_{Mo} \quad (13)$$

$$\delta_T = F_T \delta_{To} \quad (14)$$

where

$$\delta_{Mo} = \frac{2r_s}{(Nu_o - 2)}$$

$$\delta_{To} = \frac{2r_s}{(sh_o - 2)}$$

Nu and sh are the Nusselt and Sherwood numbers respectively.

3.3.3. 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. The liquid mass fraction is evaluated by solving the species equation. A Crank- Niclson scheme is adapted to solve these equations.

$$\frac{\partial T_l}{\partial t_b} = \frac{\partial^2 T_l}{\partial \tilde{r}^2} + \left(\frac{2}{\tilde{r}} - \tilde{r} MK \right) \frac{\partial T_l}{\partial \tilde{r}} \quad (15)$$

$$\frac{\partial Y_{li}}{\partial t_b} = \frac{1}{Le} \frac{\partial^2 Y_{li}}{\partial \tilde{r}^2} + \left(\frac{2}{\tilde{r} Le} - \tilde{r} MK \right) \frac{\partial Y_{li}}{\partial \tilde{r}} \quad (16)$$

Where the radial distance within the droplet \tilde{r} , time t_b , liquid temperature T_l , and K are defined as

$$T_l = \frac{T_l - T_o}{T_b - T_o}, \quad t_b = \alpha \int_0^t \frac{dt}{r_k^2}, \quad \tilde{r} = \frac{r}{r_{kinitial}}$$

$$K = \frac{\rho_o D_{c_o}}{\alpha_l \rho_k}$$

Staggered meshes are applied in the physical domain to facilitate the discretization schemes. The average physical properties ($T, C_{p_g}, \mu_g, K_g, D, C_{p_f}, \rho$) are calculated by the following 1/3 rule ref. (17):

$$\phi = \phi_s + \left(\frac{\phi_g - \phi_s}{3} \right)$$

4. RESULTS

The developed mathematical model solves the flow emerging from the prefilming atomizer through the combustion chamber. By this atomizer, droplets become finer and spray becomes uniform. The

governing equations were solved to determine the air velocity effect and the variation of fuel composition on the flow characteristics. The SMD throughout the droplet lifetime for different air velocity is calculated and a sample of the results is summarized in Table 1. The SMD is sensitive to any change in air velocity and it is noticed that a small increase in air velocity leads to a great reduction in SMD. It is noted that when air velocity increases, the spray maximum droplet diameter decreases, which means that the spray distribution function is shifted towards smaller droplet diameters. This shift occurs because by increasing the air velocity, the heat exchange becomes higher thereby increasing the generated vortices inside the droplet, which enhances the evaporating rate; i.e. increasing the air velocity leads to a decrease in the droplet evaporating and combustion time. According to results obtained by this work the liquid Reynolds number increases initially and then decreases. This trend is expected because by injecting the spray in the combustion chamber heat is transferred to the droplet interior causing the droplet to heat up. During this heat up period the droplet diameter change is negligible and its velocity increases which is reflected as an increase in its Reynolds number. Then with the continuous heat transfer the droplet surface regression is initiated and consequently the Reynolds number decreases.

Table 1. Variation of air velocity versus SMD and evaporating time

Air velocity m/s	SMD μ m	Evapor time ms
50	130.0	18.5
70	86.8	14.8
90	64.9	12.0
110	51.5	10.0
130	42.6	8.5
150	36.3	7.7
170	31.6	7.0
190	27.9	6.6

If the air velocity increases, a significant decrease in the droplet lifetime is noticed, but the liquid Reynolds number showing the same trend for each case is not drastically affected. The gas Reynolds number experiences a rapid drop. The drop in the gas Reynolds number is caused mainly by the decrease in gas density and velocity. Though there is a decrease in the drag coefficient accompanied by the decrease in droplet diameter, their effect on the gas Reynolds number doesn't overcome the decreasing effect caused by the gas velocity and density. This increase in the air velocity leads to a decrease in gas Reynolds number. As expected from the obtained results the gas Reynolds number is not affected by the droplet composition but is affected by swirling. The effect of air velocity on the regression rate with respect to time is shown in Fig (1a). Knowing that the initial vaporization rate is slow because the droplet is cold,

and that the rate of evaporation will increase as the droplet heats up, and knowing also that for pure liquid droplets heating is mostly over in the early part of the droplet lifetime, such that the subsequent regression rate of its surface area or equivalently its diameter squared remains constant with time, it is not surprising to see a change of slope in the radius- time curve, which becomes steeper with the progress of time. Increasing the air velocity shifts the spray diameters towards smaller diameter distributions, which takes less time to complete combustion. Fig (1b) displays the radius time-wise variation for the multi-component fuel mixtures. Changing the droplet composition shifts the diameter distribution, and alters the evaporation time also, which leads to different radius profiles for the multi-component cases. Calculations show that the Lewis number decreases throughout the droplet lifetime and that it decreases with a higher rate with the increased air velocity. Initially Lewis number is very high because the droplet surface is cold, and the vapor concentration in the film is low, and the hydrocarbon diffusion coefficient in air is much lower than the air thermal diffusivity. With the growth of the gas film fuel concentration the droplet surface is heated and the Lewis number decreases. Increasing the air velocity, the Lewis number decreases with a higher rate as a result of higher vaporization due to the accelerated heat exchange between the droplet and the gas. For multi-component Fuel the change in Lewis number is not significant because the thermal diffusivity and the diffusion coefficients for the different hydrocarbons are within the same order and magnitude. The droplet surface exposed to hot gases changes its temperature rapidly, until reaching the boiling temperature. Fig. (2a) shows the increase in the droplet surface temperature with time for different droplet velocities. Smaller droplets are formed with higher air velocities, which leads to smaller evaporating time; this explains the different evaporating times displayed in the figure. Fig (2b) displays the droplet surface temperature for the multi-component case. The droplet composition changes the diameter distribution and consequently the decane droplet reaches its boiling temperature quicker. On the other hand the droplet center temperature, experiences a gradual increase, because it depends on the core region heat up mode.

The drag coefficient affected by the local droplet concentration in the gas flow, decreases significantly as the droplet vaporizes. In spite of the fact that the drag tends to decrease the droplet velocity, the droplet gains Kinetic energy due to the evaporation and the decrease in size, through heat transfer, which reduces the drag acting on it. On the other hand the gas tend to loose kinetic energy due to heat omission. Fig. (3a) represents the droplet velocity versus time at different air velocity. As seen from the figure the droplet velocity increases with time. The increase in air velocity leads to an increase in the droplet velocity, because on one hand it shifts the distribution curve towards smaller diameters possessing higher velocities and on the other hand the resulting decrease in relative velocity leads to an increase in heat and mass transfer rates. As seen from the figure the droplet velocity

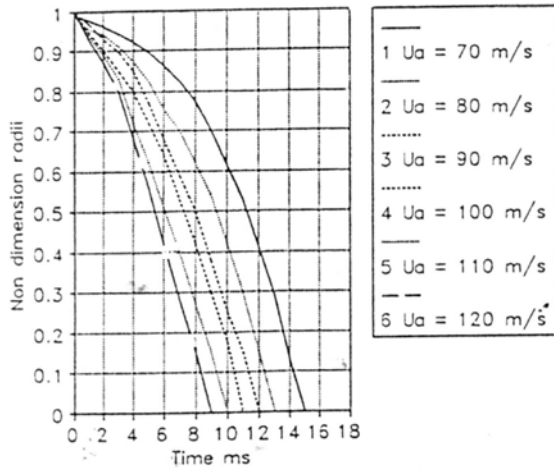


Fig.1a. Drop Radius vs. Time

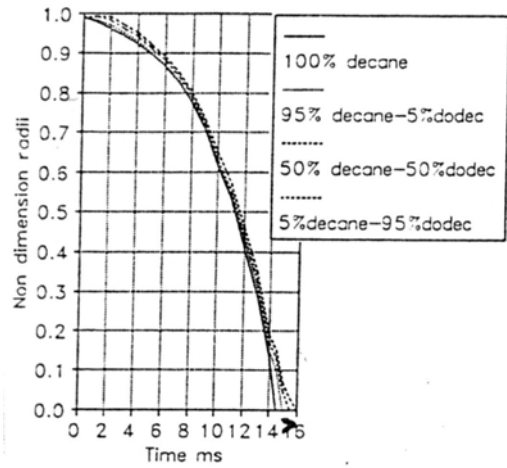


Fig.1b. Multi-comp. Radius vs. Time

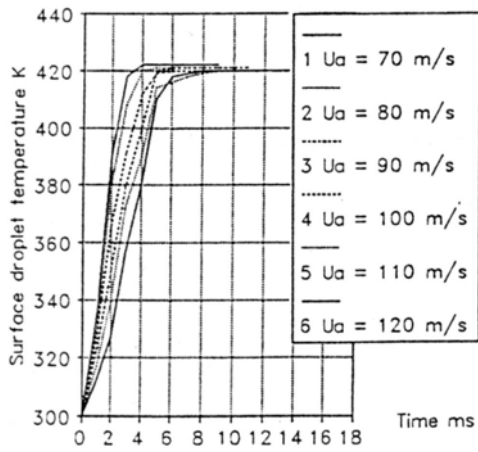


Fig.2a. Drop Surface Temp. vs. Time

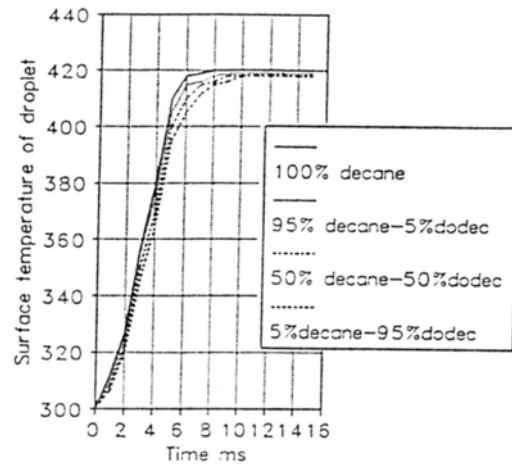


Fig.2b. . Multi-comp. Surface Temp.

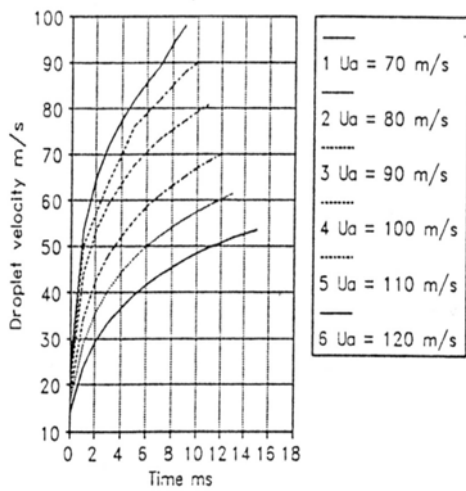


Fig.3a. Drop Velocity vs. Time

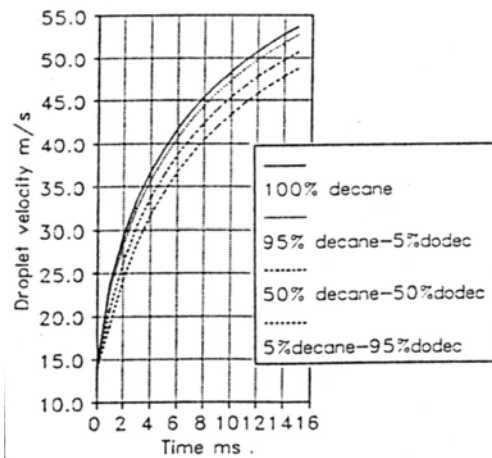


Fig.3b. Multi-component Drop Velocity vs Time

increases with time. The increase in air velocity leads to an increase in the droplet velocity, because on one hand it shifts the distribution curve towards smaller diameters possessing higher velocities and on the other hand the resulting decrease in relative velocity leads to an increase in heat and mass transfer rates. Fig (3b) shows the effect of the multi-component fuel mixture on the velocity propagation

To test the developed computer code the obtained results were compared with Chiang and Sir

gnano ref. [8,13,15]; results are in accord, which reflects our dependable work.

5. CONCLUSIONS

The developed program examined the air-velocity-operating limit for each flow characteristic by examining air velocities ranging from 50 to 190 m/s. The air velocity increase results in a decrease in Sauter mean diameter (SMD) and its evaporating time. With the help of this work the use of the air velocity corresponding to the evaporating time needed to fulfill the combustion chamber design requirements is facilitated. The tables and figures resulting from this work will help in the early stages the engineer in designing the combustion chamber. The developed program is a powerful tool in the determination of the different parameters affecting the combustion chamber design.

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