

# A New Experimental Method for the Investigation of Fuel Spray Evaporation

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A phenomenological experimental method for the investigation of fuel spray evaporation is introduced. The method is based on the measurement of pressure decrease resulting from spray evaporation in a closed vessel at high temperature and pressure. If certain physical and thermodynamic properties of the fuel are available, the method gives the average lifetime of the droplets and a mixing characteristic of the spray. If these properties are unavailable, the method can be employed to compare injector designs. A significant advantage of this technique is that it serves as an automatic integrator of all the governing parameters of spray evaporation. Ethyl alcohol and water sprays evaporating in nitrogen gas have been investigated successfully. The initial conditions of these experiments were 400 psig, 450 psig, and 500 psig chamber pressures and 700°F, 800°F, and 900°F chamber temperatures. Fuel injection rates were approximately 30 g/sec. 16-hole and 8-hole self impinging injector-plates were used. The lifetime of the sprays was found to vary between 144 and 550 msec.

## Nomenclature

$a$  =  $\{[\partial e_2(T, P)/\partial T]_P - [\partial e_2(T, 1)/\partial T]_P\}$   
 $b$  =  $[\partial e_2(T, P)/\partial P]_T$   
 $B$  =  $m_1[1 + \chi_f(\gamma + \zeta + \zeta\chi_f\mu)]/2\dot{m}_2\alpha$   
 $C_p$  = specific heat at constant pressure  
 $C_v$  = specific heat at constant volume  
 $D$  = droplet diameter  
 $e$  = specific internal energy  
 $H$  = over-all heat-transfer coefficient  
 $L$  = latent heat of vaporization  
 $m$  = mass  
 $M$  = molecular weight  
 $P$  = pressure  
 $r$  = radius  
 $R$  = universal gas constant  
 $S$  = evaporation chamber inside surface area  
 $t$  = time  
 $t''$  = age of droplet  
 $T$  = temperature  
 $V$  = evaporation chamber volume  
 $\alpha$  =  $HS/2\dot{m}_2C_v$

$$\beta = \frac{1}{C_{v1}T_0} \left[ L_2 + \int_{T_{2,0}}^{T_b} C_{p2} dT + \int_{T_b}^{T_0} C_{p2} dT + e_2(T_0, P_0) - e_2(T_0, 1) - \bar{C}_{p2}T_0 - aT_0 - bP_0 \right]$$

$\gamma$  =  $[\bar{C}_{p2} + a - (R/M_2)]/C_{v1}$   
 $\zeta$  =  $(bP_0/C_{v1}T_0)$   
 $\kappa$  =  $\chi/\chi'$   
 $\lambda$  = evaporation constant  
 $\mu$  =  $M_1/M_2$   
 $\phi$  =  $0.4\dot{m}_2t_1/m_1$   
 $\chi$  =  $m_2/m_1$   
 $\chi'$  =  $m_2/m_1'$

## Subscripts

0 = initial condition  
 1 = inert gas  
 2 = fuel  
 $b$  = boiling point  
 $f$  = final value  
 $h$  = hypothetical value  
 $L$  = droplet lifetime  
 $r$  = reference point

## Superscripts

$\cdot$  = fuel injection rate  
 $'$  = fraction of inert gas cooled  
 $-$  = average value  
 $(l)$  = liquid state

## Introduction

IN the design and development of liquid fuel combustion devices, one of the components which requires considerable attention is the design of the fuel injector. The function of the injector is to introduce and meter the fuel flow. In most cases, it also serves to atomize and mix the fuel. Usually, the fuel is introduced in the form of a spray. Before combustion can take place, the fuel must evaporate and form a correctly proportioned combustible mixture. Thus the performance of an injector is related to the spray evaporation process, which in turn is governed partly by the injector characteristics. In practice, one approach to the design of combustion chambers<sup>1,2</sup> and prediction of combustion characteristics of sprays<sup>3,4</sup> is to utilize results of single droplet evaporation<sup>5-8</sup> and combustion<sup>9,10</sup> studies. One weakness of this approach is that single droplet evaporation and combustion phenomena depart grossly from the complex situation encountered in combustion chambers. Moreover, this approach often requires a knowledge of the physical properties such as vapor pressure, surface tension, latent heat of vaporization, specific heat, and diffusion coefficient, etc., of the fuel at near-critical, critical, and supercritical conditions. But these properties of fuels are not known. Thus, any analysis or experimental technique for high-pressure spray evaporation studies would be useful only if it is independent of the preceding data. The present method satisfies this requirement by circumventing the preceding difficulties in the analysis.

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The aim of this paper is to introduce a phenomenological experimental method for the investigation of spray evaporation and to report the results of the studies conducted at moderate temperature and pressure ranges, with ethyl alcohol and water, using this method. This work forms the first part of a study directed toward the development of a method which is intended to correlate spray evaporation at moderate temperature and pressure ranges to spray combustion at pressures including the supercritical range.

The foundation of the method is the following observed phenomenon. When liquid fuel is sprayed through an injector into a vessel filled with an inert gas at high temperature and pressure, heat transfer takes place between the droplets of the liquid spray and the surrounding gas. Since combustion is prevented, the ensuing evaporation of the droplets decreases the gas temperature and consequently causes a pressure decrease in the vessel. The pressure decrease at any time after the beginning of fuel injection will always be smaller than that which would occur if the fuel evaporated instantaneously. The latter is an idealized process which cannot be achieved in practice. However, it serves as a standard for comparison purposes. This limiting case will be referred to as the hypothetical instantaneous evaporation in the following analysis. In other words, the actual spray evaporation process lags behind the idealized process. Making use of this fact, an analytical treatment can be performed to derive a semiempirical formula for the prediction of the average droplet lifetime of a spray, provided that certain physical properties of the fuel are known. To simplify the analysis, all droplets are assumed to have the same size at formation. In reality, droplet sizes vary and the spray droplet size at formation determined from results obtained by this method is actually an average size of the drops.

## Assumptions and Basic Equations

### Assumptions

The theoretical model spray is assumed to possess the following characteristics: 1) the liquid fuel mass injection rate is constant; 2) the initial droplet size is uniform at formation; 3) the mass evaporation rate of all droplets having the same size is the same; 4) there is no wall effect; 5) the mass evaporation rate of the spray during quasi-steady state is constant (i.e., changes in evaporation chamber temperature and pressure are both small compared to the initial conditions) and is the same as the mass injection rate; and 6) a portion of the inert gas is cooled uniformly to a lower temperature  $T$  while the remaining undisturbed portion is at the initial temperature. This cooled portion increases with fuel injection time.

For the analysis, the evaporation process of a spray is divided into four distinct periods. These are the following:

- Liquid fuel build-up within the evaporation chamber, i.e., from the start of fuel injection until a full size spray is formed and the total liquid mass of the spray becomes constant.
- Quasi-steady-state fuel injection and evaporation (i.e., the evaporation rate of the spray) equals the fuel injection rate, and the changes in evaporation chamber temperature and pressure are small with regard to the initial conditions.
- After fuel injection, evaporation of remaining droplets results in further temperature and pressure decrease in the evaporation chamber.
- After fuel injection: warm-up period of the mixture.

### Basic Equations

The Ideal-Gas-Law describes the initial conditions of the inert gas

$$P_0 V = (m_1/M_1)RT_0 \quad (1)$$

After fuel evaporation, one has for the mixture

$$PV = \frac{m_1 - m_1'}{M_1} RT_0 + \left( \frac{m_1'}{M_1} + \frac{m_2}{M_2} \right) RT \quad (2)$$

### Conservation of Energy

The internal energy of the gas and fuel system must be constant in the absence of wall effects and is given by the internal energy balance

$$(m_1 - m_1')e_{1,0} + m_1'e_1 + m_2e_2 = m_1e_{1,0} + m_2e_{2,0} \quad (3)$$

The quadratic law of droplet evaporation

$$D^2 = D_0^2 - \lambda t \quad (4)$$

When Eq. (4) is expressed in droplet radius at  $t = t''$ , the age of the droplet is

$$r^2 = r_0^2(1 - t''/t_L) \quad (4a)$$

## Analysis

### Derivation of Hypothetical Instantaneous Temperature and Pressure Decrease Formulas

These formulas are applicable in the fuel injection period with an idealized spray evaporation process. They are required to serve as a standard for comparison purposes with the actual case. To derive these formulas, specific internal energy expressions which relate the initial and final states of the gas and fuel are needed. For the portion of the gas which is cooled

$$e_1 = e_{1,0} + C_{v_1}(T_h - T_0) \quad (5)$$

at the end of fuel evaporation.

For the fuel, an arbitrary path is chosen such that the physical properties of the liquid fuel being investigated are available for all processes between the end states. Since the specific internal energy is a property, its change between any two states is independent of the processes which bring about the change. The following is a description of the arbitrarily chosen path: a) decrease of initial pressure to one atmosphere at constant temperature (the specific internal energy is assumed constant); b) heating-up of the liquid fuel to its boiling temperature at constant atmospheric pressure; c) isobaric evaporation of the liquid fuel at atmospheric pressure; d) heating-up of the fuel vapor to its final temperature by a constant pressure process; and e) increase of vapor pressure to its final value by a constant temperature process. If the vapor were considered to be an ideal gas, the internal energy would not change. When it is considered a nonideal gas, the change is calculated in two parts. The first part consists of a change from the initial state of the process,  $T = T_0$  and  $P = 1$  atm, to the initial conditions existing in the chamber,  $T = T_0$  and  $P = P_0$ . The second part consists of changes due to deviations of temperature from  $T_0$  and pressure from  $P_0$ .

In equation form, the specific internal energy of the fuel is therefore

$$e_2 = e_{2,0} + \int_{T_{2,0}}^{T_h} C_2 dT + L_2 - \frac{RT}{M_2} + \int_{T_h}^{T_b} C_{p_2} dT + e_2(T_0, P_0) - e_2(T_0, 1) + a(T_h - T_0) + b(P_h - P_0) \quad (6)$$

or

$$e_2 = e_{2,0} + \gamma C_{v_1} T_h + \beta C_{v_1} T_0 + b P_h \quad (6a)$$

where  $\gamma$  and  $\beta$  are dimensionless constants given by

$$\beta = \frac{1}{C_{v_1} T_0} \left[ L_2 + \int_{T_{2,0}}^{T_h} C_2 dT + \int_{T_h}^{T_0} C_{p_2} dT + e_2(T_0, P_0) - e_2(T_0, 1) - \bar{C}_{p_2} T_0 - a T_0 - b P_0 \right] \quad (7a)$$

$$\gamma = [\bar{C}_{p_2} + a - (R/M_2)]/C_{v_1} \quad (7b)$$

and

$$a = \{[\partial e_2(T, P)/\partial T]_P - [\partial e_2(T, 1)/\partial T]_P\} \quad (8a)$$

$$b = [\partial e_2(T, P)/\partial P]_T \quad (8b)$$

Combining Eqs. (3, 5, and 6a) and rearranging gives

$$\Delta T_h = T_0 - T_h = T_0 \chi' \frac{[\gamma + \beta + (b P_h / C_{v_1} T_0)]}{1 + \chi'} \quad (9)$$

§ Neglecting the energy of injection.

Since it is more meaningful to measure pressure rather than temperature changes in the evaporation chamber during spray evaporation, the corresponding pressure change formula is required. This is obtained by solving Eqs. (1, 2, and 9) and is given by

$$\Delta P_h = P_0 - P_h = P_0 \chi \left[ \frac{(\beta + \gamma + \zeta - \mu) + (\zeta + \beta) \mu \chi'}{1 + \chi'(\gamma + \kappa \zeta) + \chi \chi' \mu \zeta} \right] \quad (10)$$

where  $\mu = M_1/M_2$ ,  $\zeta = bP_0/C_{V_1}T_0$ , and  $\kappa = m_1'/m_1 = \chi/\chi'$ .  $\kappa$  can be thought of as a spray spread coefficient. It is a mass ratio of the inert gas within the volume of the spray and its vapor to the total inert gas in the evaporation chamber. It is also a volume ratio between the volume occupied by the two-phase fuel jet and the evaporation chamber volume.

#### Liquid Fuel Mass Existing in the Evaporation Chamber during Fuel Injection

From the quadratic law, Eq. (4a), the volume of each droplet having an age  $t''$  when compared to its initial volume is proportional to

$$[1 - (t''/t_L)]^{3/2} \quad (11)$$

In the period of time,  $dt''$ , the liquid fuel mass injected is  $\dot{m}_2 dt''$ . Assuming that the liquid density is constant, (for ethyl alcohol, the error introduced is less than 2% if thermal expansion is neglected) the mass of all droplets of an age between  $t''$  and  $t'' + dt''$  is

$$dm^{(l)} = \dot{m}_2 dt'' (1 - t''/t_L)^{3/2} \quad (12)$$

The total mass of liquid fuel in the chamber during the quasi-steady-state fuel injection period is

$$m^{(l)} = \int_0^{t_L} dm^{(l)} = \frac{2}{5} \dot{m}_2 t_L \quad (13)$$

#### Real Conditions during Fuel Injection

In deriving the actual pressure change formula, consideration is given to the finite lifetime of the droplets and heat transfer from the evaporation chamber wall to the gaseous mixture. The following additional assumptions are made: 1) the liquid droplets are considered to remain at their initial temperature throughout their lifetime; 2) the fuel vapor is considered to be at the temperature  $T$  of the cooled gas; 3) the chamber wall remains at constant temperature,  $T_0$ ; 4) the volume of the liquid spray droplets is negligibly small.

The energy balance equation is

$$m_1'(e_1 - e_{1,0}) + \left[ m_2(t) - \int_0^{t_L} dm^{(l)} \right] (e_2 - e_{2,0}) = HS \int_{t_L}^t (T_0 - T) dt \quad (14)$$

or

$$\kappa(T - T_0) + (\chi - \phi)[\gamma T + \beta T_0 + (1 - \kappa)\zeta T_0 + (\kappa + \chi\mu - \phi\mu)\zeta T] = \frac{HS}{m_1 C_{V_1}} \int_0^t (T_0 - T) dt \quad (15)$$

where  $\phi = 0.4\dot{m}_2 t_L / m_1$ . In formulating the heat-transfer term on the R.H.S. of Eq. (14), we choose to use the evaporation chamber inner surface area,  $S$ , and an over-all heat transfer coefficient,  $H$ , between the chamber wall and the vapor-gas mixture, because the chamber wall constitutes the only possible source of heat transfer to the vapor-gas system. Equation (15) can best be solved by approximation. The term on the R.H.S., although not negligible, is still small compared to the other terms. Hence one finds a first approximation for  $T$  by letting the R.H.S. equal to zero and solving the equation for  $T$ . Next the expression for  $T$  is substituted into the integral in Eq. (15) and integration is performed. Then the resulting expression is solved to give

$$T = T_0 \frac{[(\chi/\kappa)(\beta + \gamma + \zeta)(\chi^2 - 2\phi\chi) + \kappa - (\chi - \phi)\beta - (\chi - \phi)(1 - \kappa)\zeta]}{\kappa + (\chi - \phi)\gamma + (\chi - \phi)(\kappa + \chi\mu - \phi\mu)\zeta} \quad (16)$$

where

$$\alpha = HS/2\dot{m}_2 C_{V_1}$$

The corresponding formula for the pressure is

$$P = P_0 [1 - (\chi - \phi)(\gamma + \beta + \zeta - \mu) + (\chi - \phi)^2 \zeta (\beta + \zeta - \gamma - \mu) + [(\chi - \phi)^2 / \kappa] (\beta + \gamma + \zeta)(\alpha + \gamma - \mu)] \quad (17)$$

In the absence of heat transfer (i.e.,  $\alpha = 0$ ) and for sprays with a lifetime approaching zero (i.e.,  $t_L = 0$ ), Eq. (17) reduces to the same relationship as in Eq. (10).

#### Lifetime of the Spray: An Average Lifetime of Droplets

The method in deriving the formula for the average lifetime of the droplets in the spray (i.e., the lifetime of the spray) is to use the hypothetical instantaneous pressure change as the standard and to compare it with the actual pressure change. This gives the amount by which the actual pressure change lags behind the idealized case. From this difference and the measured rate of pressure change, a formula for the lifetime of spray is obtained. Consequently, we require

$$\frac{-(P - P_h)}{(dP/d\chi)} \quad (18)$$

Making use of Eqs. (10) and (17) and the derivative of Eq. (17) with respect to  $\chi$  and substituting into Eq. (18) gives, after rearranging

$$t_L = 2.5 \left\{ -\frac{P - P_h}{(dP/dt)} - \frac{\alpha m_1 (\gamma + \beta + \zeta) \chi^2}{\kappa \dot{m}_2 (\gamma + \beta + \zeta - \mu)} - \zeta \chi^2 \frac{m_1}{\dot{m}_2} \left[ \frac{(\beta + \zeta - \gamma - \mu)}{(\gamma + \beta + \zeta - \mu)} - \frac{1}{\kappa} \right] \right\} \quad (19)$$

Equation (19) is a semiempirical formula for the lifetime of the spray. The second term in the parentheses reflects the heat transfer associated with the process and the third term reflects departures from ideality of the fuel vapor. However, in order to determine  $t_L$ , the quantity  $\alpha$  of Eq. (19) must be known. Since  $\alpha$  is a function of the heat-transfer characteristic of the equipment, it may be estimated from the data obtained during the warm-up period of an experiment. This implies that the amount of heat transferred to the gaseous mixture during the fuel evaporation period and the warm-up period are the same. During the warm-up period, sufficient time has elapsed such that the entire mass of inert gas may be cooled. Assuming this is the case, then  $m_1' \approx m_1$ , and  $\kappa \approx 1$ . The energy balance equation is

$$C_{V_1} [m_1 + m_2 (\gamma + \zeta + \zeta \chi \mu)] dT = HS(T_0 - T) dt \quad (20)$$

The solution of Eq. (20) is

$$T_0 - T = (T_0 - T_r) \exp[-(t - t_r)/B] \quad (21)$$

where

$$B = m_1 [1 + \chi_f (\gamma + \zeta + \zeta \chi_f \mu)] / 2\dot{m}_2 \alpha \quad (22)$$

Equation (21) when expressed in terms of pressure becomes

$$P_\infty - P = (P_\infty - P_r) \exp[-(t - t_r)/B] \quad (23)$$

where the constant  $B$  is to be determined from the experimental pressure vs time curve. Once  $B$  is known,  $\alpha$  can be calculated from Eq. (22) and Eq. (19) can be solved.

#### Spray Evaporation Experiments

Spray evaporation experiments were conducted by injecting predetermined masses of liquid into a high temperature and high pressure evaporation chamber filled with nitrogen gas. Fuel injection is accomplished by a displacement technique and the injection rate is monitored by means of a linear displacement transducer. Pressure change within the evaporation chamber is monitored by means of a differential pressure transducer. Solid cone sprays which were produced by 16-hole (0.015-in.-diam) and 8-hole (0.021-in.-diam) self-impinging injector-plates were investigated. The holes are drilled at an angle of 30° to the axis so that all liquid jets converge into an apex  $\frac{1}{4}$  in. above the

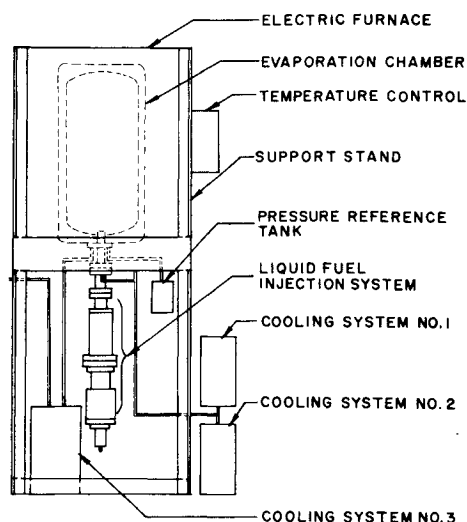


Fig. 1 Apparatus layout.

injector-plate. Figure 1 shows the general layout of the apparatus assembly. Figure 2 illustrates the liquid fuel injection system.

Referring to Fig. 2, when the diaphragm is ruptured by the piercing rod, pressure is applied to the lower surface of the number two piston. This forces piston number one into the fuel reservoir and displaces an identical volume of fuel. A total of six sets of initial temperature and pressure combinations ranging from 700°F to 900°F at 100°F intervals and 400 psig to 500 psig at 50 psi intervals were used. The injection pressure for all experiments was approximately 200 psia.

#### Data Evaluation and Analysis of Results

Figure 3 is an example photographic record of the data output. The upper trace shows the total fuel mass injection at any time. The lower trace indicates the pressure change in the evaporation chamber vs time. All recordings were evaluated by means of a projector-comparator. These data were then used to calculate the average lifetime of the droplets from Eq. (19). However, examination of this equation reveals that it contains two unknowns, namely  $t_L$  and  $\kappa$ . One must therefore resort to the curve fitting

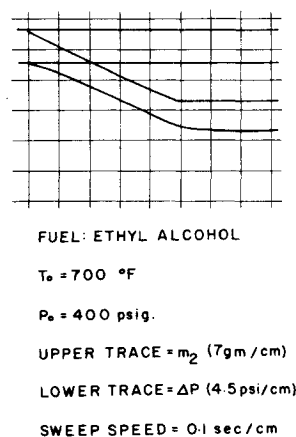


Fig. 3 Camera recording.

technique to find  $t_L$ . The following procedure is employed. First, a set of spray lifetimes were calculated as functions of  $\chi$  and a set of arbitrarily chosen  $\kappa$ . Results so obtained are then presented in a  $t_L$  vs  $\chi$  plot for constant  $\kappa$ 's. Next, one must analyze the physical phenomena in order to predict the correct  $t_L$ . Carrying out this procedure, our main task therefore is to determine from the data of Fig. 4, which is a typical plot of the calculated results, the actual  $t_L$ .

We know that for any given set of operating conditions, there can only be one  $t_L$  associated with the spray of a particular injector-plate. A second criterion is that corresponding to any instant of fuel injection, there is only one value of  $\kappa$ . Also,  $\kappa$  can only increase with time starting from the moment of fuel injection. It increases from its lower limit, which is fixed by the initial spray volume, to a maximum value of unity, when the entire inert gas in the system is cooled. Another condition is that quasi-steady-state spray evaporation (i.e., a full size liquid spray having a constant total liquid mass is formed; the spray evaporation rate equals the fuel injection rate; and changes in mixture pressure is small compared to the initial pressure) occurs only when the fuel injection time is equal to or greater than the spray lifetime. Since for  $t_L > \text{injection time}$ , there is not enough time available for a full size spray to form, the minimum fuel injection time required is then identical to the spray lifetime,  $t_L$ .

Referring to Fig. 4, the inclined straight line specifies the time required for the injection of a liquid mass corresponding to  $\chi$  subjected to the particular experimental conditions listed. To the upper left of this line,  $t_L > \text{injection time}$  and the region

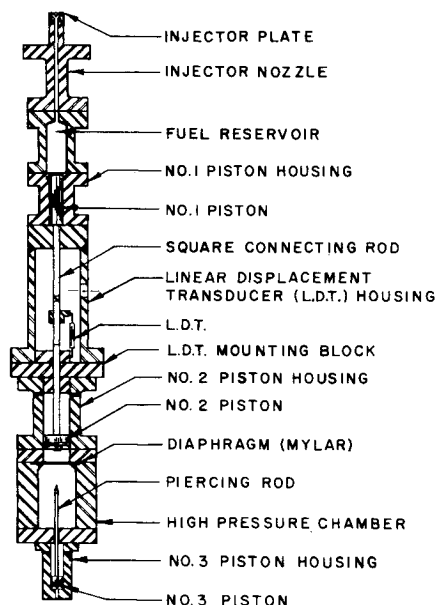
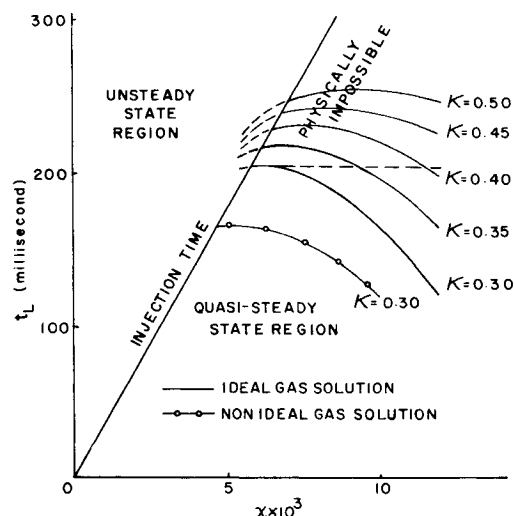


Fig. 2 Liquid fuel injection system.

Fig. 4 Lifetime of ethyl alcohol spray, 16-hole injector,  $T_0 = 900^\circ\text{F}$ ,  $P_0 = 400$  psig.

**Table 1** Spray lifetime  $t_L$ 

Injector	Fuel	Pressure (psig)	Temp. (°F)	Injection rate (g/sec)	$\kappa$	$t_L$ (msec)
16-hole	Ethyl Alcohol	400	700	32.48	$0.23 \pm 0.01$	$180 \pm 3$
		450	700	31.96	$0.22 \pm 0.01$	$149 \pm 3$
		500	700	32.36	$0.22 \pm 0.01$	$144 \pm 2$
		400	800	32.31	$0.25 \pm 0.01$	$158 \pm 3$
		400	800	29.90	$0.25 \pm 0.01$	$169 \pm 4$
		450	800	32.60	$0.22 \pm 0.01$	$144 \pm 3$
		400	900	29.66	$0.30 \pm 0.01$	$165 \pm 4$
8-hole	Water	400	700		$0.40 \pm 0.01$	$512 \pm 12$
			800	35.79	$0.44 \pm 0.01$	$480 \pm 12$
			900		$0.45 \pm 0.01$	$422 \pm 10$
	Ethyl alcohol	400	700		$0.26 \pm 0.01$	$196 \pm 4$
			800	32.52	$0.27 \pm 0.01$	$169 \pm 4$
			900	29.89	$0.33 \pm 0.01$	$178 \pm 4$
			700		$0.43 \pm 0.01$	$558 \pm 14$
	Water	400	800	36.01	$0.48 \pm 0.01$	$516 \pm 12$
			900		$0.48 \pm 0.01$	$462 \pm 12$

represents unsteady-state spray evaporation. To the lower right,  $t_L < \text{injection time}$  and the region signifies quasi-steady states. The spray lifetime,  $t_L$ , should therefore be located on this line.

Next step is to identify the correct  $\kappa$ . When  $\kappa$  is known, the intersection of the constant  $\kappa$  curve and the inclined straight line meet all the necessary criteria and thus yields the actual  $t_L$ . In Fig. 4, the family of solid curves for  $\kappa$  values at 0.05 intervals and within the quasi-steady-state region were obtained by treating the fuel vapor as an ideal gas [i.e., by letting  $a = b = 0$ , and  $e_2(T_0, P_0) = e_2(T_0, 1)$ , in the governing equations]. It can be readily observed that at small values of  $\chi$ , this family of curves tends to converge to a narrow range of  $t_L$ . As  $\chi$  increases, the curves tend to diverge and become concave downward. This family exhibits a trend for which the curves of constant  $\kappa$  have a positive slope at their intersection with the injection time line when  $\kappa$  is large and a negative slope for lower  $\kappa$ . Those for which the slope is positive are physically impossible since  $\kappa$  can hardly decrease as  $\chi$  increases at constant  $t_L$ .<sup>11,12</sup> Therefore, all  $\kappa$  curves intersecting the injection time line above the dotted horizontal line in Fig. 4 are physically impossible solutions. The other curves all have a maximum which falls on the inclined line and below the dotted line. These curves are bounded above by the maximum physically possible  $\kappa$  curve and below by the limiting  $\kappa$  curve.

Now, in order to identify the correct  $\kappa$ , one must consider the actual physical phenomenon. As injection proceeds, turbulent mixing along with mass and momentum transfer of the spray tend to increase the cooled fraction of the inert gas within the evaporation chamber. A second factor, which also promotes the increase of  $\kappa$ , is diffusion of the fuel vapor due to the existence of a concentration gradient. Based on these observations, the maximum possible  $\kappa$  curve should be chosen. The spray lifetime,  $t_L$  so obtained, is a maximum possible, and from the standpoint of designers, this number then is the spray lifetime of interest.

When the fuel vapor is considered a nonideal gas, a similar procedure can be followed to determine  $t_L$ . One such curve, which was obtained from the same experimental results as in the ideal gas solutions, was drawn for  $\kappa = 0.30$ . This curve appears as the lowest curve in Fig. 4. Some spray lifetimes determined by this method are shown in Table 1.

### Discussion

In the formulation of the governing equations, both the liquid fuel vapor and the inert gas were assumed to obey the Ideal Gas Law. For the inert gas used, this assumption is accurate even at high pressures. However, for the fuel vapor, deviation from this assumption is quite considerable at high pressures. Nevertheless, the Ideal Gas Law can still be applied in Eq. (2), because  $m_1'$  is much greater than  $m_2$  and a large deviation in

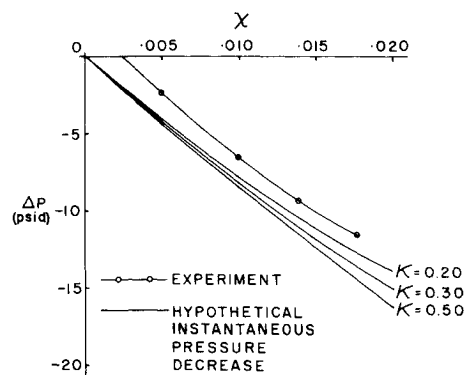
the term containing  $m_2$  cannot affect the entire equation by any significant amount.

In solving the basic equations, the approximation technique was employed. It is based on the assumptions that the ratio of the total injected liquid fuel mass to the inert gas mass  $\chi$  and the amount of heat transfer from the chamber wall to the gaseous mixture are small. This approach is adopted because of the complexity of the analytical mathematical model. To obtain small values of  $\chi$ , the total injected liquid fuel mass can be adjusted. This and the rate of fuel injection can be used to control the amount of heat transfer to the gaseous mixture to a minimum by keeping the inert gas temperature decrease small. The influence of  $\chi$  on  $t_L$  can be observed in Fig. 4. At small  $\kappa$ , the constant  $\kappa$  curves tend to converge to a narrow range when extended into the unsteady-state region. As  $\chi$  increases, these curves would tend to give negative  $t_L$  values. Since  $t_L$  is a function of both  $\chi$  and  $\kappa$  and that the influence of  $\chi$  is much greater than  $\kappa$ , this indicates that the approximation method of solving the basic equations becomes less accurate as  $\chi$  increases. Therefore care must be taken in an experiment to ensure small  $\chi$  values. On the other hand, one must keep the  $\chi$  values large enough so that the calculated lifetime of the spray lies outside of the unsteady-state region for the result to be meaningful.

Solution of the basic equations also involves the determination of the end state specific internal energy of the liquid vapors considered. For the five processes of the arbitrarily chosen path, the specific internal energy of the liquid vapor is allowed to vary. The fuel vapor is considered a nonideal gas in this case. However, if the fuel vapor can be considered as an ideal gas, the present analysis still applies. The only difference between these two cases lies in the last process as mentioned earlier.

Based on identical experimental data,  $t_L$  determined using non-ideal gas formula is shorter than that calculated from the ideal gas expression (see Fig. 4). This is understandable because at the temperature and pressure ranges investigated, the specific internal energy of the fuel vapor at the final state is substantially less than that of an ideal gas subjected to an identical isothermal compression process.

In the calculation of  $t_L$  from experimental data, increments of  $\kappa$  were chosen to be 0.01. This was considered accurate enough because the maximum error introduced is about  $\pm 2.0\%$  for small values of  $\chi$ . Since the solutions were approximations only, it was considered unnecessary to computerize the calculating procedure. A second source of error is in the evaluation of the pressure data. The maximum error introduced through the recording instrument is approximately  $\pm 3.0\%$ . Other sources of error that set a limit to the accuracy of the present method are introduced through the simplifying assumptions. These include the size distributions, unequal evaporation rate of same size droplets, deviations from the Ideal Gas Law and others. Though they are fairly well known, there seems to be no easy way of substantially diminishing the absolute size of the errors.



**Fig. 5** Comparison of experimental and calculated pressure decreases, ethyl alcohol,  $T_0 = 900^\circ\text{F}$ ,  $P_0 = 400$  psig.

A limitation of this method is its application to predict  $t_L$  of actual fuels having many species, since most of the required thermodynamic properties are not available. Even then, this method can still be used to compare different injector plate designs. Referring to Fig. 5, from the theoretical curves, it can be readily seen that  $\Delta P$  is a function of both the evaporated fuel mass and the spread of the two-phase fuel jet in the form of  $\kappa$ . However, the influence of  $\chi$  on  $\Delta P$  is at least one order of magnitude greater than that of  $\kappa$ . Also, the experimental  $\Delta P$  curve lags behind the theoretical  $\Delta P$  curve. Both these features can be utilized to compare injector plates using actual fuels. The better injector plate will produce a pressure decrease curve closer to the idealized curve. Such a curve would also exhibit a smaller evaporation lag. This is an important practical feature of this method.

A second limitation is that the evaporation chamber temperature had to be kept relatively low compared to the combustion temperature. This prevents one from obtaining directly pressure change curves corresponding to combustion temperature conditions. However, from Table 1, it can be seen that spray lifetimes decrease with increasing pressure and temperature. Therefore, it is reasonable to expect that an injector plate which performs better at low temperatures should also perform better at higher temperatures. In the next phase of this study, it is intended to correlate the performance of spray evaporation to that of spray combustion.

### Conclusions

A technique has been described that yields experimental data, which, when combined with an appropriate analysis with certain physical properties, enables the designer to determine the average lifetime of a spray of liquid droplets produced by any injector. The method has been proved to work satisfactorily when pure fuels were used. In these cases, an estimate can be made on the spray lifetime and its spread coefficient. It was also proved that the better injector will produce a pressure decrease in the evaporation chamber closer to the idealized pressure decrease under identical operating conditions, i.e., a better injector has smaller evaporation lag and greater rate of pressure decrease.

The experimental apparatus and procedure is simple, extremely repeatable, and capable of detecting small differences

in injection characteristics quite accurate for many pure fuels. Even in the absence of such thermophysical property data for actual fuels of many species, the experimental method may still be used to evaluate different injector plate designs by comparing their pressure decrease curves directly.

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