

It is of interest to note in Table 3.3 that as q increases (toward a more uniform spray), the volume fraction Q at which $D = \text{SMD}$ also increases. This means that, relatively speaking, SMD is increasing. This somewhat curious trend occurs because increase in q causes all the drops in the spray to shift closer to $D_{0.5}$, which eliminates many of the smallest drops. For many combustion systems, where very small drops are needed to provide high initial fuel evaporation rates for rapid ignition, a more uniform distribution of drop sizes in the spray may not always be desirable.

It is clearly important to distinguish between the concept of a representative diameter and a diameter that provides an indication of atomization quality. Any of the diameters listed in Table 3.2 can be used as a representative diameter to describe drop size distribution, but only the SMD can properly indicate the fineness of the spray from a combustion viewpoint. Even if the mass median diameter ($D_{0.5}$) is decreasing, it is still not certain whether the spray is finer, because if q also changes it is possible that SMD will either increase or remain sensibly constant. For the same reason, other representative diameters such as D_{peak} , X , and $D_{0.9}$ cannot indicate the fineness of the spray. It is strongly recommended, therefore, that for combustion applications the Sauter mean diameter be used to describe atomization quality, since the use of any representative diameter could lead to erroneous conclusions about the fineness of the spray.

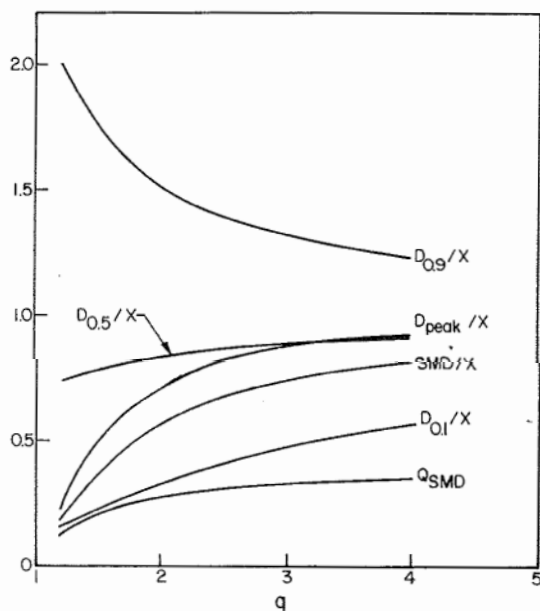


Figure 3.14 Relationship between Rosin-Rammler distribution parameter q and various spray characteristics (Chin and Lefebvre [11]).

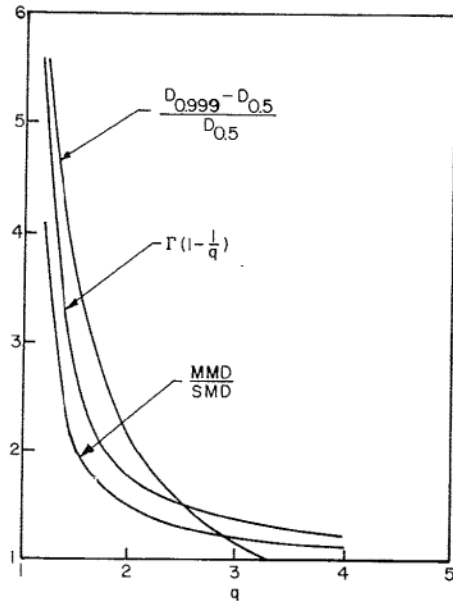


Figure 3.15 Relationship between Rosin-Rammler distribution parameter q and various spray characteristics (Chin and Lefebvre [11]).

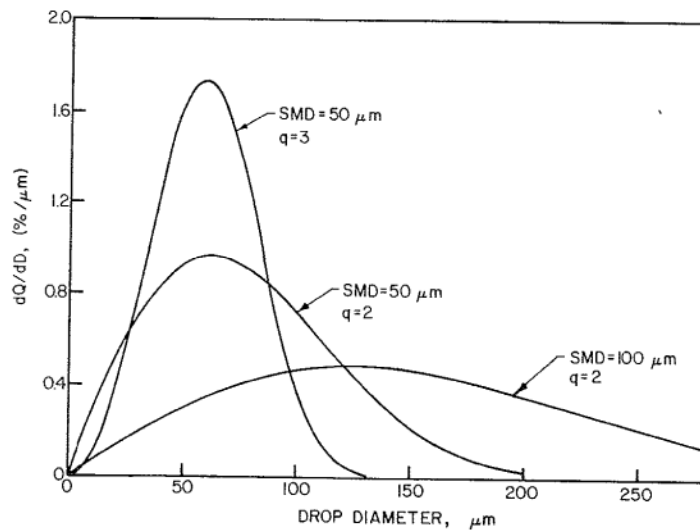


Figure 3.16 Influence of SMD and q on drop size frequency distribution curves.

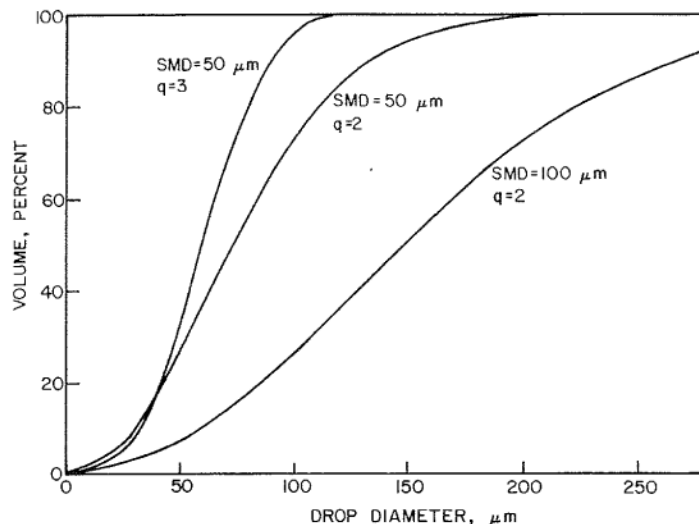


Figure 3.17 Influence of SMD and q on cumulative drop size distribution curves.

DROP SIZE DISPERSION

The term *dispersion* is sometimes used as an alternative to *distribution* to express the range of drop sizes in a spray. The concept of *width* is also employed to indicate drop size range, but it can be misleading. For example, the width of a spray having a certain SMD may be less than that of another spray having a higher SMD, but it does not necessarily follow that the former has a smaller drop size dispersion than the latter, because their average values are different.

It is generally recognized that the ratio MMD/SMD provides a good indication of drop size dispersion (see, for example, reference [13]) but, as discussed above, a slight change in this ratio could correspond to a large change in dispersion (or q).

Droplet Uniformity Index

Tate [14] has proposed a *droplet uniformity index* to describe the spread of drop sizes in a spray. It is defined as

$$\text{Droplet uniformity index (volumetric basis)} = \frac{\sum_i V_i(D_{0.5} - D_i)}{D_{0.5}} \quad (3.39)$$

where D_i is the midpoint of size class i and V_i is the volume fraction in the size class.

This expression indicates the spread relative to the mass median diameter and

takes into account all the discrete size classes. It can be shown that the droplet uniformity index depends mainly on q and to a lesser extent on mean diameter.

Relative Span Factor

This parameter, defined by

$$\Delta = \frac{D_{0.9} - D_{0.1}}{D_{0.5}} \quad (3.40)$$

has much to commend it. First, it provides a direct indication of the range of drop sizes relative to the mass median diameter. Second, from Eqs. (3.28)–(3.30) we have, for a Rosin-Rammler distribution,

$$\Delta = (3.322)^{1/q} - (0.152)^{1/q} \quad (3.41)$$

which is also a unique function of q .

The manner and extent of change in the relative span factor with q is shown in Table 3.3.

Dispersion Index

Another parameter that is used to express the range of drop sizes is the *dispersion index*, which is defined for a Rosin-Rammler distribution as

$$\delta = \int_0^{D_m} D \frac{dQ}{dD} dD = \int_0^{D_m} q \left(\frac{D}{X} \right)^q \exp \left[- \left(\frac{D}{X} \right)^q \right] dD \quad (3.42)$$

which is essentially the same as the *uniformity index* proposed by Tate [14].

It should be noted that the dispersion index depends not only on q but also to a lesser degree on X . Due to its more complicated form, δ has no advantage over the relative span factor Δ . Thus for most engineering purposes the relative span factor provides an adequate description of dispersion.

Dispersion Boundary Factor

To give some indication of the maximum possible drop size, it is useful to define a dispersion boundary factor given by

$$\Delta_B = \frac{D_{0.999} - D_{0.5}}{D_{0.5}} \quad (3.43)$$

By assuming a Rosin-Rammler distribution, from Eq. (3.14) we have

$$\frac{D_{0.999}}{X} = (6.90775)^{1/q} \quad (3.44)$$

Hence, from Eqs. (3.34) and (3.44) we have

$$\Delta_B = (9.9665)^{1/q} - 1 \quad (3.45)$$

which is also a unique function of q . Values of Δ_B are listed in Table 3.3.

From Table 3.3, which shows the variation of dispersion boundary factor with q , it is possible to estimate the maximum drop size in a spray for any given value of q . For example, for

$$q = 2, \quad D_{0.999} = 3.16D_{0.5}$$

$$q = 3, \quad D_{0.999} = 2.15D_{0.5}$$

$$q = 4, \quad D_{0.999} = 1.77D_{0.5}$$

The rationale employed by Chin and his colleagues [11, 12] in developing methods for defining drop size distributions solely in terms of a representative diameter and the Rosin-Rammler dispersion parameter q could easily be extended to more complicated distribution functions such as the upper-limit function. However, this would add complexity without necessarily improving accuracy and would be difficult to justify unless some situation arose where another distribution function demonstrated a clear superiority over the Rosin-Rammler distribution parameter.

CONCLUDING REMARKS

Much of this chapter has been devoted to a review of available methods for characterizing drop size distributions in sprays. It is difficult to judge whether any one method is superior to all others; such a conclusion must await theoretical determinations based on a more complete understanding of the basic mechanisms involved in drop formation. In the meantime, the following observations may be of guidance in avoiding some of the more commonly encountered pitfalls:

1. No single parameter can completely define a drop size distribution. For example, two sprays are not necessarily similar just because they have the same Sauter mean diameter (SMD) or the same mass median diameter (MMD). In many practical applications it is the smallest drop sizes in a spray or the largest drop sizes that are of paramount importance, and neither SMD nor MMD can provide this information.
2. There is no universal correlation between the mean diameter (or representative diameter) of a spray and its drop size distribution. They are completely independent of each other.
3. Mean diameters and representative diameters are different in nature. The MMD is not a mean diameter; it is a representative diameter. In particular, MMD ($D_{0.5}$) should not be confused with the mass mean diameter D_{30} .
4. If the Rosin-Rammler distribution is used, the distribution of drop sizes in a spray is defined by two parameters, a representative diameter and a measure of drop size dispersion.

5. Relative span factor can be used to indicate the spread of drop sizes in a spray.
6. Dispersion boundary factor can be used to define a meaningful maximum drop size.

NOMENCLATURE

D	drop diameter
$\bar{D}$	mean drop diameter
$D_{0.1}$	drop diameter such that 10% of total liquid volume is in drops of smaller diameter
$D_{0.5}$	drop diameter such that 50% of total liquid volume is in drops of smaller diameter
$D_{0.632}$	drop diameter such that 63.2% of total liquid volume is in drops of smaller diameter
$D_{0.9}$	drop diameter such that 90% of total liquid volume is in drops of smaller diameter
$D_{0.999}$	drop diameter such that 99.9% of total liquid volume is in drops of smaller diameter
D_{peak}	value of D corresponding to peak of drop size frequency distribution curve
D_m	maximum drop diameter
D_0	minimum drop diameter
D_{ng}	geometric number mean diameter
D_{sg}	geometric surface mean diameter
D_{vg}	geometric volume mean diameter
MMD	mass median diameter ($D_{0.5}$)
N	number of drops
Q	liquid volume fraction (or percentage) containing drops of smaller diameter than D
q	Rosin-Rammler drop size distribution parameter
SMD	Sauter mean diameter (D_{32})
s_g	geometric standard deviation
s_n	standard deviation
V	volume
X	characteristic diameter in Rosin-Rammler equation
δ	dispersion index
Δ	relative span factor
Δ_B	dispersion boundary factor

REFERENCES

1. Fraser, R. P., and Eisenklam, P., Liquid Atomization and the Drop Size of Sprays, *Trans. Inst. Chem. Eng.*, Vol. 34, 1956, pp. 294-319.

2. Rizk, N. K., and Lefebvre, A. H., Airblast Atomization: Studies on Drop-Size Distribution, *J. Energy*, Vol. 6, No. 5, 1982, pp. 323–327.
3. Miesse, C. C., and Putnam, A. A., Mathematical Expressions for Drop-Size Distributions, *Injection and Combustion of Liquid Fuels*, Section II, WADC Technical Report 56-344, Battelle Memorial Institute, March 1957.
4. Nukiyama, S., and Tanasawa, Y., Experiments on the Atomization of Liquids in an Air Stream, Report 3, On the Droplet-Size Distribution in an Atomized Jet, Defense Research Board, Department National Defense, Ottawa, Canada; translated from *Trans. Soc. Mech. Eng. Jpn.*, Vol. 5, No. 18, 1939, pp. 62–67.
5. Rosin, P., and Rammler, E., The Laws Governing the Fineness of Powdered Coal, *J. Inst. Fuel*, Vol. 7, No. 31, 1933, pp. 29–36.
6. Rizk, N. K., and Lefebvre, A. H., Drop-Size Distribution Characteristics of Spill-Return Atomizers, *AIAA J. Propul. Power*, Vol. 1, No. 3, 1985, pp. 16–22.
7. Rizk, N. K., Spray Characteristics of the LHX Nozzle, Allison Gas Turbine Engines Report Nos. AR 0300-90 and AR 0300-91, 1984.
8. Mugele, R., and Evans, H. D., Droplet Size Distributions in Sprays, *Ind. Eng. Chem.*, Vol. 43, No. 6, 1951, pp. 1317–1324.
9. Brodkey, R. A., *The Phenomena of Fluid Motions*, Addison-Wesley, Reading, Mass., 1967.
10. Marshall, W. R., Jr., Mathematical Representations of Drop-Size Distributions of Sprays, *Atomization and Spray Drying*, Chapter VI, Chem. Eng. Prog. Monogr. Ser., Vol. 50, No. 2, American Institute of Chemical Engineers, 1954.
11. Chin, J. S., and Lefebvre, A. H., Some Comments on the Characterization of Drop-Size Distributions in Sprays, *Proceedings of the 3rd International Conference on Liquid Atomization and Spray Systems*, London, Paper No. IVA/1, 1985.
12. Zhao, Y. H., Hou, M. H., and Chin, J. S., Dropsizes Distributions from Swirl and Airblast Atomizers, *Atomization Spray Technol.*, Vol. 2, 1986, pp. 3–15.
13. Martin, C. A., and Markham, D. L., Empirical Correlation of Drop Size/Volume Fraction Distribution in Gas Turbine Fuel Nozzle Sprays, ASME Paper ASME 79-WA/GT-12, 1979.
14. Tate, R. W., Some Problems Associated with the Accurate Representation of Drop-Size Distributions, *Proceedings of the 2nd International Conference on Liquid Atomization and Sprays (ICLASS)*, Madison, Wis., 1982, pp. 341–351.

FLOW IN ATOMIZERS

INTRODUCTION

In twin-fluid atomizers of the airblast and air-assist types, atomization and spray dispersion tend to be dominated by air momentum forces, with hydrodynamic processes playing only a secondary role. With pressure-swirl nozzles, however, the internal flow characteristics are of primary importance, because they govern the thickness and uniformity of the annular liquid film formed in the final discharge orifice as well as the relative magnitude of the axial and tangential components of velocity of this film. It is therefore of great practical interest to examine the interrelationships that exist between internal flow characteristics, nozzle design variables, and important spray features such as cone angle and mean drop size. The various equations that have been derived for nozzle discharge coefficient are discussed at some length, because this coefficient not only affects the flow rate of any given nozzle but also can be used to calculate its velocity coefficient and spray cone angle.

Consideration is also given to the complex flow situations that arise on the surface of a rotating cup or disk. These flow characteristics are of basic importance to the successful operation of these atomizers, because they exercise a controlling influence on the nature of the atomization process, the quality of atomization, and distribution of drop sizes in the spray.

FLOW NUMBER

The effective flow area of a pressure atomizer is usually described in terms of a flow number, which is expressed as the ratio of the nozzle throughput to the

square root of the fuel-injection pressure differential. Two definitions of flow number are in general use: a British version, based on the volume flow rate, and the other is an American version, based on the mass flow rate. They are

$$FN_{UK} = \frac{\text{flow rate, UK gal/h}}{(\text{injection pressure differential, psid})^{0.5}} \quad (5.1)$$

$$FN_{US} = \frac{\text{flow rate, lb/h}}{(\text{injection pressure differential, psid})^{0.5}} \quad (5.2)$$

Note that 1 UK gallon = 1.2 US gallons.

Equations (5.1) and (5.2) have the advantage of being expressed in units that are in general use. Unfortunately, they are basically unsound. For example, they do not allow a fixed and constant value of flow number to be assigned to any given nozzle. Thus, although it is customary to stamp or engrave a value of flow number on the body of a simplex atomizer, this value is correct only when the nozzle is flowing a standard calibrating fluid of density 765 kg/m^3 . In the past this has posed no problems with aircraft gas turbines because 765 kg/m^3 roughly corresponds to the density of aviation gasoline. However, for liquids of other densities, these two definitions of flow number could lead to appreciable errors when used to calculate liquid flow rates or injection pressures.

The basic deficiency in Eqs. (5.1) and (5.2) is the omission of liquid density. Inclusion of this property would not only allow these equations to be rewritten in dimensionally correct form but also enable the flow number to be defined in a much more positive and useful manner than at present, namely as the effective flow area of the nozzle. Thus the flow number of any given nozzle would have a fixed and constant value for all liquids.

By including liquid density, the flow number in square meters is obtained as

$$FN = \frac{\text{flow rate, kg/s}}{(\text{pressure differential, Pa})^{0.5} (\text{liquid density, kg/m}^3)^{0.5}} \quad (5.3)$$

The standard UK and US flow numbers may be calculated from Eq. (5.3) using the formulas:

$$FN_{UK} = 0.66 \times 10^8 \times \rho_L^{-0.5} \times FN \quad (5.4)$$

$$FN_{US} = 0.66 \times 10^6 \times \rho_L^{0.5} \times FN \quad (5.5)$$

PLAIN-ORIFICE ATOMIZER

It is generally considered that the flow in a plain-orifice atomizer is similar to that in a pipe, and consequently the nature of the flow is related to the Reynolds number. Measurements of discharge coefficient carried out by Bird [1] and Gel-lales [2] showed great differences at high as compared with low Reynolds number. Schweitzer [3] noted that the spray was more rapidly dispersed when the Reynolds

number exceeded a certain value, and he attributed this to the transition from laminar to turbulent flow, which resulted in more rapid disruption of the jet.

The critical value of Reynolds number, that is, the value at which the nature of flow changes from laminar to turbulent, usually lies between 2000 and 3000. For values below the critical Reynolds number the flow tends to be laminar, while at higher values it tends to be turbulent. One reason why the critical value of Reynolds number cannot be precisely defined is that it depends on the geometry of the atomizer and on the properties of the liquid [3, 4]. In general, laminar flow is promoted by

1. A rounded entrance to the orifice
2. Smooth passage walls
3. Absence of bends
4. High liquid viscosity
5. Low liquid velocity

Turbulent flow is promoted by

1. Large passage diameters
2. Changes in flow velocity and direction
3. Abrupt changes in cross-sectional area
4. Surface roughness
5. Imperfections in atomizer geometry
6. Mechanical vibrations
7. Low liquid viscosity
8. High liquid velocity

Turbulence is conducive to good atomization but usually at the expense of some increase in pressure loss.

For low-viscosity liquids, such as water, kerosine, and light diesel oil, the flow in the atomizer is normally turbulent. However, in intermittent injection systems there is a period at the beginning of each injection when the velocity is rising from zero and a corresponding period at the end when it falls again to zero, during which the flow is either laminar or semiturbulent.

Discharge Coefficient

The discharge coefficient of a plain-orifice atomizer is governed partly by the pressure losses incurred in the nozzle flow passages and also by the extent to which the liquid flowing through the final discharge orifice makes full use of the available flow area. Discharge coefficient is related to nozzle flow rate by the equation

$$\dot{m}_L = C_D A_o (2\rho_L \Delta P_L)^{0.5} \quad (5.6)$$

$$= 1.111 C_D d_o^2 (\rho_L \Delta P_L)^{0.5} \quad (5.7)$$

$$= 35.12 C_D d_o^2 (SG \Delta P_L)^{0.5} \quad (5.8)$$

Factors influencing discharge coefficient. Measurements of discharge coefficient carried out on various orifice configurations over wide ranges of operating conditions indicate that the most important parameters are Reynolds number, length/diameter ratio, injection pressure differential, ambient gas pressure, inlet chamfer, and cavitation.

Reynolds number. The influence of Reynolds number on discharge coefficient has been investigated by Bird [1], Gellales [2], Bergwerk [5], Spikes and Pennington [6], Lichtarowicz et al. [7], Arai et al. [8], and others. The manner in which Reynolds number influences the discharge coefficient of a plate orifice is illustrated in Fig. 5.1. This figure exhibits three distinct sections. In the first stage, corresponding to laminar flow, C_D increases almost linearly with $\sqrt{Re}$. During the second stage, corresponding to semiturbulent flow, C_D at first increases with Re up to a maximum value, beyond which further increase in Re causes C_D to decline. In the fully turbulent stage, which is of most practical interest, C_D remains sensibly constant.

Length/diameter ratio. The characteristic peak in the curve of C_D against Re for a sharp-edged orifice diminishes rapidly as l_o/d_o is increased from 0.5 to unity, as illustrated in Fig. 5.2. This figure is based on experimental data compiled from several sources by Lichtarowicz et al. [7]. It shows that for $l_o/d_o = 0.5$ the passage is similar to a simple plate orifice, as illustrated in Fig. 5.3. The discharge coefficient is low because the liquid jet forms a vena contracta, which, in the short length available, has no time to reexpand and fill the nozzle. With increase in l_o/d_o the jet expands in the passage and C_D increases, reaching a maximum at a

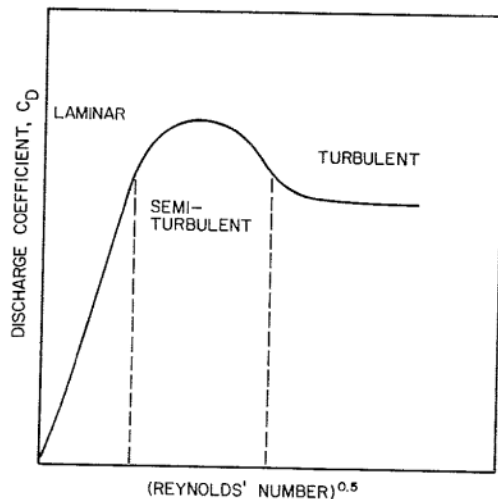


Figure 5.1 Variation in discharge coefficient with Reynolds number (Giffen and Muraszew [4]).

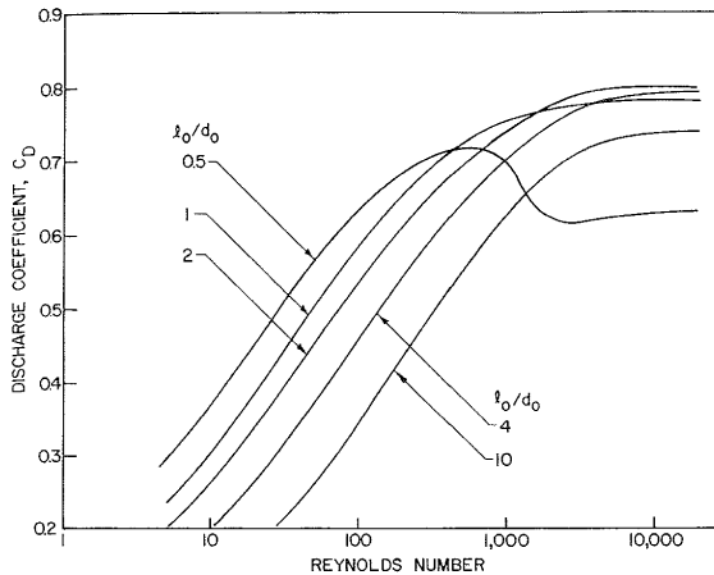


Figure 5.2 Variation of discharge coefficient with Reynolds number for various orifice length/diameter ratios. Based on experimental data compiled by Lichtarowicz et al. [7].

value of l_o/d_o of around 2. Further increase in l_o/d_o reduces C_D due to increase in frictional losses.

Figure 5.2 shows that discharge coefficients generally increase with increase in Reynolds numbers until a maximum value is attained at a Reynolds number of around 10,000. Beyond this point, the value of C_D remains sensibly constant and independent of Reynolds number. The maximum values of C_D are shown plotted against l_o/d_o in Fig. 5.4. The experimental data on which this figure is based were drawn from reference [7], but actual data points have been omitted for clarity. The figure shows $C_{D_{max}}$ rising steeply from about 0.61 to a maximum value of about 0.81 as l_o/d_o increases from 0 to 2. Further increase in l_o/d_o causes $C_{D_{max}}$ to decline slowly in a nearly linear fashion to about 0.74 at $l_o/d_o = 10$. Figure

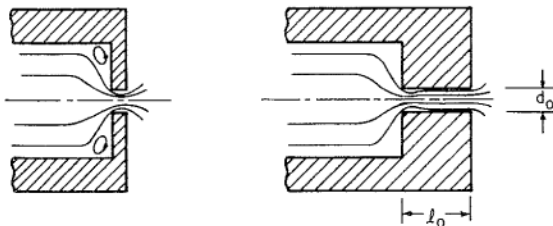


Figure 5.3 Influence of final orifice length/diameter ratio on flow pattern.

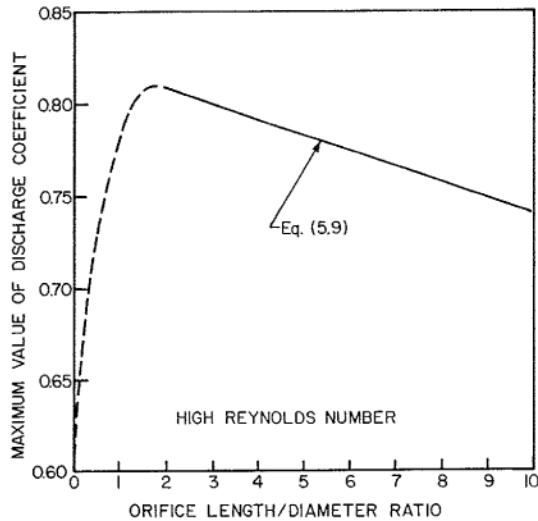


Figure 5.4 Variation of maximum value of discharge coefficient with orifice length/diameter ratio. Based on experimental data compiled by Lichtarowicz et al. [7].

5.4 indicates a linear relationship between $C_{D_{\max}}$ and l_o/d_o for the range of l_o/d_o between 2 and 10. For this region, Lichtarowicz et al. [7] proposed the following expression, which is claimed to fit the experimental data to within about 1%:

$$C_{D_{\max}} = 0.827 - 0.0085 \frac{l_o}{d_o} \quad (5.9)$$

Injection pressure. The influence of liquid injection pressure differential on C_D is quite small. For example, Gellales [2] found for diesel oil an increase in C_D from 0.91 to 0.93 for a five-fold increase in injection pressure. The result was obtained at an l_o/d_o ratio of 3. For higher values of l_o/d_o the discharge coefficient decreases with an increase in injection pressure. This is due mainly to higher frictional losses, which increase with the square of the velocity [4].

Ambient pressure. The effect of ambient air or gas pressure on discharge coefficient has been investigated by Arai et al. [8]. Their results, with data points omitted, are shown in Fig. 5.5. They were obtained when flowing water through a circular nozzle 1.2 mm long and 0.3 mm in diameter. It is of interest to observe in this figure that C_D changes not only with Re but also with ambient gas pressure. At normal atmospheric pressure, C_D reaches a maximum value of around 0.8 at $Re = 3000$. For Reynolds numbers between 3000 and 15,000 it has two values. The higher value corresponds to Re increasing from 3000 and the lower value corresponds to Re decreasing from 15,000. For Reynolds numbers larger than 15,000, C_D remains single-valued at around 0.7.

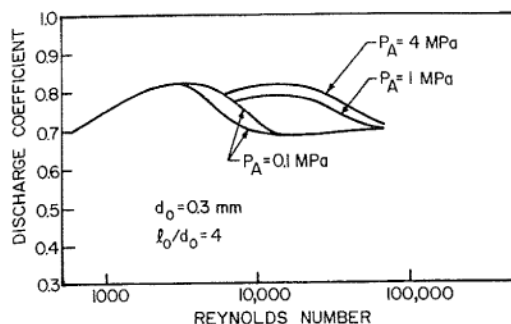


Figure 5.5 Influence of ambient air pressure on discharge coefficient. Based on experimental data of Arai et al. [8].

Double values of C_D were not observed at higher ambient pressures. Instead, C_D remained fairly constant and independent of ambient pressure at a value of about 0.8 over a range of Re from 2000 to 20,000. At higher values of Re , C_D slowly declines from 0.8 at $Re = 20,000$ to around 0.72 at $Re = 50,000$.

Inlet chamfer. During tests on diesel injection nozzles, Bergwerk [5] found that an inlet chamfer on the nozzle could raise the discharge coefficient. Similar results were obtained by Zucrow [9], who showed that the discharge coefficient reaches a maximum for submerged orifices with low pressure drop when the included angle of the chamfer is between 20° and 60° .

Detailed information on the influence of inlet chamfer on discharge coefficient has been provided by Spikes and Pennington [6]. These workers carried out a comprehensive test program on submerged orifices to determine the optimum angle and depth of chamfer for maintaining a constant discharge coefficient over the greatest possible range of operating conditions. In a first series of tests they varied the chamfer angle. An orifice 1.57 mm in diameter, having a fixed parallel throat length of 0.51 mm, was chosen for these tests and a chamfer 0.51 mm deep was cut on the upstream edge. The main feature of the results obtained using chamfered orifices is the increase in discharge coefficient given by the chamfer, as illustrated in Fig. 5.6. This figure indicates an optimum chamfer angle of about 50° , which is consistent with Zucrow's [9] findings.

Further tests were made to examine the effects of chamfer depth. Figure 5.7 shows the variation in discharge coefficient with chamfer depth for an orifice with a chamfer having an included angle of 50° . This figure demonstrates a strong dependence of discharge coefficient on depth of chamfer and shows that even slight variations in chamfer depth can have a marked effect on orifice flow rate.

Cavitation. In flow regions of low static pressure, gas or vapor may be released from the liquid to form bubbles, which can have a pronounced influence on discharge coefficient. The subsequent explosion or collapse of these bubbles can also

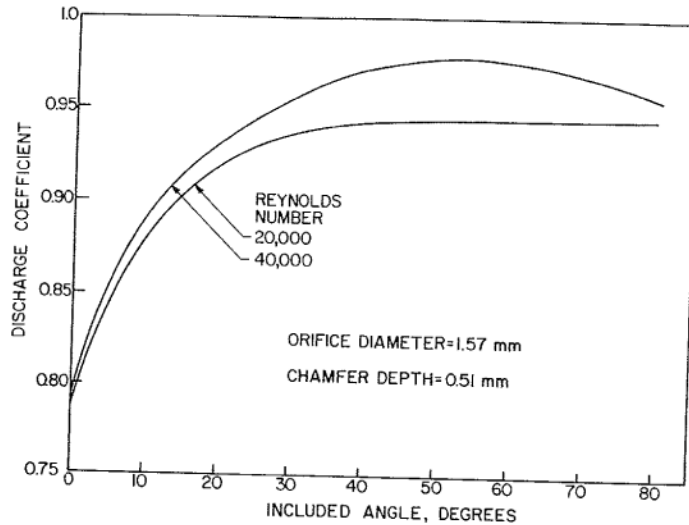


Figure 5.6 Variation of discharge coefficient with angle of upstream chamfer. Based on experimental data of Spikes and Pennington [6].

accelerate jet breakup. In addition to its adverse effect on discharge coefficient, cavitation may cause severe erosion of nozzle passages. The normal working range of diesel nozzles is such that both cavitating and noncavitating flow may occur under certain conditions.

The results of tests carried out by Spikes and Pennington [6] using square-edged orifices are shown in Fig. 5.8. In these curves the discharge coefficient is shown as a function of Reynolds number and cavitation number C , the latter being defined as

$$C = \frac{p_1 - p_2}{p_2 - p_v} \quad (5.10)$$

where p_1 is the upstream pressure, p_2 is the downstream pressure, and p_v is the vapor pressure. Cavitation occurs if the local pressure becomes equal to the vapor pressure of the liquid. The fluid used in Spikes and Pennington's tests was aviation kerosine. The vapor pressure of this liquid is sufficiently small to be neglected. Under such conditions the cavitation number may be expressed more simply as

$$C = \frac{p_1 - p_2}{p_2} = \frac{\text{pressure drop}}{\text{downstream pressure}} \quad (5.11)$$

Figure 5.8 shows that the effects of cavitation can be high and can cause changes of discharge coefficient greater than those associated with Reynolds number when the latter is high.

Many orifices show instability around the point of changeover from cavitating to noncavitating flow. With long holes this is due to the tendency of the liquid

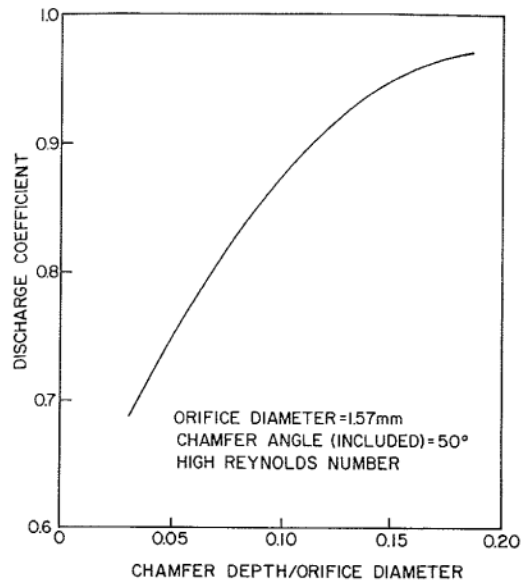


Figure 5.7 Variation of discharge coefficient with depth of upstream chamfer. Based on experimental data of Spikes and Pennington [6].

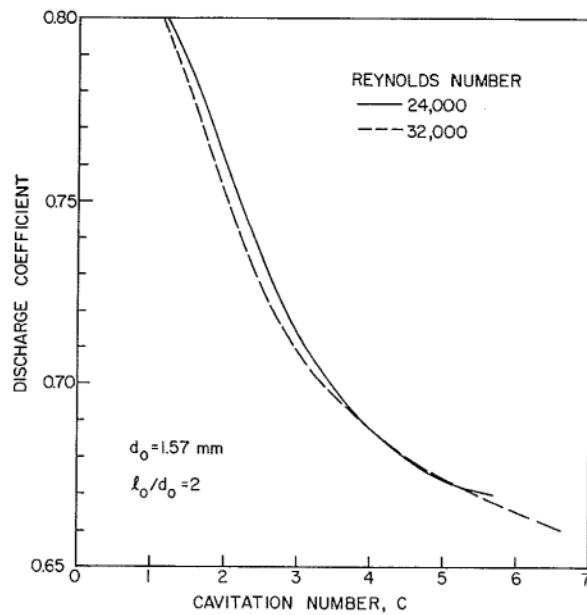


Figure 5.8 Influence of cavitation on discharge coefficient for a square-edged round orifice. Based on experimental data of Spikes and Pennington [6].

downstream of the vena contracta to expand and fill the cavity. The use of a very short length places the cavitation zone some distance downstream of the orifice, so that it cannot influence the flow pattern through the hole. Instabilities can also be eliminated by making the hole sufficiently long, but this results in large changes in discharge coefficient with both cavitation number and Reynolds number [6].

As cavitation is associated with sharp corners, which create regions of high liquid velocity and low static pressure, it can be alleviated to some extent by replacing the sharp corner at the entry of the orifice with a lead-in chamfer. An ideal chamfer would have a profile similar to that of a jet stream during contraction downstream of a sharp-edged entry, and the orifice itself would have a diameter equal to that of the vena contracta. The experiments of Spikes and Pennington [6] have shown that this ideal condition can be approached closely with a straight-sided chamfer if the included angle of the chamfer is 50° and the chamfer depth is 0.30 times the diameter of the orifice. However, it should be noted that although this chamfer will suppress cavitation over a wide range, Reynolds number effects can still cause appreciable changes in the discharge coefficient [6].

Empirical expressions. The lack of any quantitative theory for flow in orifices over wide ranges of Reynolds number has led to a number of empirical correlations for noncavitating flows. According to Nakayama [10]

$$C_D = \frac{Re^{5/6}}{17.11l_o/d_o + 1.65Re^{0.8}} \quad (5.12)$$

for l_o/d_o in the range 1.5 to 17 and Re in the range 550 to 7000, to an accuracy of within 2.8%. Nakayama also suggests

$$C_{D_{\max}} = 0.868 - 0.0425 \left(\frac{l_o}{d_o} \right)^{0.5} \quad (5.13)$$

as an expression for discharge coefficient at high Reynolds numbers, valid over the same range of l_o/d_o . This formula gives lower values than Eq. (5.9), from Lichtarowicz et al. [7], since in Nakayama's tests the range of Re was restricted at the upper end.

Asihmin et al. [11] suggest

$$C_D = \left[1.23 + \frac{58(l_o/d_o)}{Re} \right]^{-1} \quad (5.14)$$

for l_o/d_o in the range 2 to 5 and Re from 100 to 1.5×10^5 , claiming an accuracy within 1.5%. To accommodate a wider variation in l_o/d_o , Lichtarowicz et al. [7] proposed the following modification of Asihmin's equation:

$$\frac{1}{C_D} = \frac{1}{C_{D_{\max}}} + \frac{20}{Re} \left(1 + 2.25 \frac{l_o}{d_o} \right) \quad (5.15)$$

in which $C_{D_{\max}}$ is given by Eq. (5.9). This provides an excellent fit to the experimental data [7] in the range of l_o/d_o from 2 to 10 and for Re in the range from 10 to 20,000.

It is important to note that these empirical equations for C_D do not apply under conditions where cavitation is present. Tests carried out by Ruiz and Chigier [12] on a diesel electromagnetic injector showed the occurrence of cavitation in this type of injector over most of its operating range. For Reynolds numbers up to about 40,000, C_D remained sensibly constant at around 0.7, but cavitation effects caused C_D to fall to around 0.6 with increase in Reynolds number from 40,000 to 50,000.

Another complication with diesel injectors is that the value of Reynolds number varies during the injection period. Varde and Popa [13] quote values of C_D varying from 0.5 to 0.8 depending on the average Reynolds number.

PRESSURE-SWIRL ATOMIZER

In this type of nozzle the liquid is injected through tangential or helical passages into a swirl chamber, from which it emerges with both tangential and axial velocity components to form a thin conical sheet at the nozzle exit. This sheet rapidly attenuates, finally disintegrating into ligaments and then drops.

Despite the geometric simplicity of the simplex swirl atomizer, the hydrodynamic processes occurring within the nozzle are highly complex. Nevertheless, the early theories based on the assumption of frictionless flow soon led to the formulation of quantitative relationships between the main atomizer dimensions and various flow parameters, such as discharge coefficient, and initial spray cone angle [14, 15]. Subsequently, Taylor [16] showed that although the motion in the bulk of the fluid can be considered irrotational, the viscous effect of the retarded boundary layer is far from negligible. The liquid in contact with the swirl chamber end walls cannot rotate at a sufficient rate to hold it in a circular path against the radial pressure gradient, and hence a current directed toward the orifice is set up through the surface layer. Other workers [17, 18] have shown that for real fluids an outward flow may also occur through a boundary layer around the air core. However, as Dombrowski and Hassan [19] and Jones [20] have emphasized, for low-viscosity liquids simple inviscid analysis still provides a basic understanding of the flow characteristics of pressure-swirl atomizers and gives a reasonable guide to discharge coefficients and cone angle.

Discharge Coefficient

The discharge coefficient of a swirl atomizer is inevitably low, owing to the presence of the air core, which effectively blocks off the central portion of the orifice. Radcliffe [21] studied the performance of a family of injectors based on common design rules, using fluids that covered wide ranges of density and viscosity. He demonstrated the existence of a unique relationship between the discharge coefficient and the Reynolds number based on orifice diameter. At low Reynolds numbers, the effect of viscosity is to thicken the fluid film in the final orifice and thereby increase in discharge coefficient. With nozzles of small flow number this effect can be significant at low flow rates. However, for Reynolds numbers larger

than 3000, that is, over most of the normal working range, the discharge coefficient is practically independent of Reynolds number. Thus, for liquids of low viscosity, the convention is to disregard conditions at low Reynolds number and assume that any given atomizer has a constant discharge coefficient.

The following analysis, from Giffen and Muraszew [4], refers to a simplex atomizer, but the results obtained can be applied to other types of pressure-swirl nozzles, such as duplex, dual-orifice, and spill-return. The liquid flow pattern is produced by the imposition of a spiral motion on a free vortex, as illustrated in Fig. 5.9. Conservation of angular momentum provides the following relationship between tangential velocity v and radius r :

$$vr = v_i R_s \quad (5.16)$$

where v_i is the inlet velocity to the swirl chamber and R_s is its radius. Also, we have

$$v_i = \frac{\dot{m}_L}{\rho_L A_p} \quad (5.17)$$

where A_p is the total cross-sectional area of the inlet ports. This equation implies the existence of an air core at the center of the swirl chamber, which is always observed in practice, since with $r = 0$ the velocity v would otherwise be infinite.

With the assumption that no losses occur within the atomizer, the total head can be considered constant throughout the swirl chamber and equal to the injection pressure P . The total pressure at any point in the liquid flowing through the orifice is then given by Bernoulli's equation as

$$P = p + 0.5\rho_L u^2 + 0.5\rho_L v^2 = \text{constant} \quad (5.18)$$

where p is the static pressure at any point in the liquid and u is the axial velocity in the orifice. But for a free vortex alone it can be shown that $p + 0.5\rho_L v^2 = \text{constant}$, with the result that, for the steady flow of a nonviscous fluid, the axial velocity u is uniform and constant for all values of r in the liquid annulus around the air core in the orifice. At the air core the static pressure is the back pressure of the ambient atmosphere, i.e., $p = 0$, and so

$$P = 0.5\rho_L(u_{r_a}^2 + v_{r_a}^2) \quad (5.19)$$

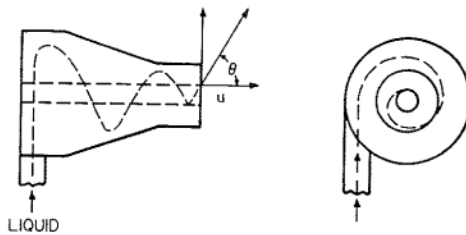


Figure 5.9 Illustration of flow path in simplex atomizer.

Since u remains constant, we have $u_{ra} = u$. Substitution of u for u_{ra} in Eq. (5.19) gives

$$P = 0.5\rho_L(v_{ra}^2 + u^2) \quad (5.20)$$

The axial component of velocity in the orifice is given by

$$u = \frac{\dot{m}_L}{\rho_L(A_o - A_a)} \quad (5.21)$$

where A_o is the orifice area and A_a is the air core area.

From Eqs. (5.16) and (5.17) we have

$$v_{ra} = \frac{\dot{m}_L R_s}{\rho_L A_p r_a} \quad (5.22)$$

Substituting into Eq. (5.20) u and v_{ra} from Eqs. (5.21) and (5.22), respectively, gives

$$P = 0.5\rho_L \left[\left(\frac{\dot{m}_L R_s}{\rho_L A_p r_a} \right)^2 + \left(\frac{\dot{m}_L}{\rho_L (A_o - A_a)} \right)^2 \right] \quad (5.23)$$

Now

$$\dot{m}_L = C_D A_o (2\rho_L P)^{0.5} \quad (5.6)$$

Substitution of $\dot{m}_L$ from Eq. (5.6) into Eq. (5.23) gives

$$\frac{1}{C_D^2} = \frac{1}{K_1^2 X} + \frac{1}{(1 - X)^2} \quad (5.24)$$

where $X = A_a/A_o$ and $K_1 = A_p/\pi r_o R_s$.

Equation (5.24) provides a relationship between the atomizer dimensions, the size of the air core, and the discharge coefficient of the nozzle. To eliminate one of these variables we may apply the condition that, for any given value of K_1 , the size of the air core will always be such as to give maximum flow; i.e., the value of C_D , expressed as a function of X , is a maximum [4].

Putting $d(1/C_D^2)/dX = 0$ leads to

$$2K_1^2 X^2 = (1 - X)^3 \quad (5.25)$$

Substituting in Eq. (5.24) the value of K_1 from Eq. (5.25) gives

$$C_D = \left[\frac{(1 - X)^3}{1 + X} \right]^{0.5} \quad (5.26)$$

Since, from Eq. (5.25) X is a unique function of K_1 , it follows from Eq. (5.26) that C_D also depends solely on K and is independent of injection pressure. Figure 5.10 illustrates the relationship of C_D with the atomizer constant K , where $K = A_p/D_s d_o = \pi K_1/4$.

Giffen and Muraszew [4] observed that Eq. (5.26) gave values of C_D that

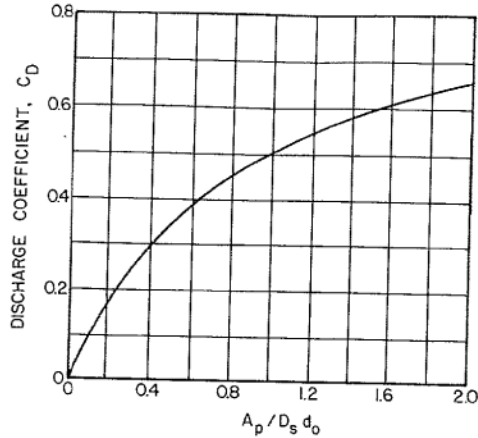


Figure 5.10 Theoretical relationship between discharge coefficient and atomizer dimensions.

were too low in comparison with the experimental data. To accommodate this they introduced a constant into Eq. (5.26), which then becomes

$$C_D = 1.17 \left[\frac{(1 - X)^3}{1 + X} \right]^{0.5} \quad (5.27)$$

Several other equations for discharge coefficient have been derived. According to Taylor [16], for inviscid flow in a swirl atomizer the discharge coefficient is given by

$$C_D^2 = 0.225 \frac{A_p}{D_s d_o} \quad (5.28)$$

This equation requires modification in the light of Carlisle's [22] evidence on the effect of D_s/d_o and L_s/D_s on discharge coefficient. The relationship between C_D and D_s/d_o is indicated by the curve drawn through the data points in Fig. 5.11, which corresponds to the relationship

$$\frac{C_{D,\text{meas}}}{C_{D,\text{theor}}} = 0.55 \left(\frac{D_s}{d_o} \right)^{0.5} \quad (5.29)$$

Thus the influence of D_s/d_o can readily be accommodated using the correction factor $0.55(D_s/d_o)^{0.5}$. With regard to L_s/D_s , inspection of Fig. 5.12 reveals that over the range of most interest (i.e., from 0.5 to 1.0) the appropriate correction factor is constant at about 0.95. Incorporating these two terms into Eq. (5.28) gives the relationship

$$C_D^2 = 0.0616 \frac{D_s}{d_o} \frac{A_p}{D_s d_o} \quad (5.30)$$

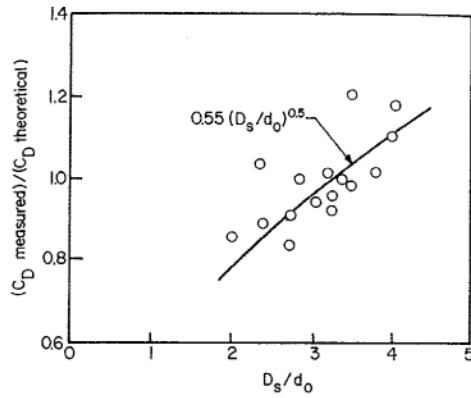


Figure 5.11 Influence of swirl chamber dimensions on discharge coefficient (Carlisle [22]).

This equation is shown plotted in Fig. 5.13 for two values of D_s/d_o , namely 3.5 and 5.0.

Eisenklam [23] and Dombrowski and Hassan [19] used the following dimensionally correct group to correlate their experimental data on discharge coefficients:

$$\frac{A_p}{D_s d_o} \left(\frac{D_s}{d_o} \right)^{1-n} \quad (5.31)$$

Eisenklam states that the value of n varies from 0.1 to 0.5, but Dombrowski and Hassan claim that a unique correlation is obtained for $n = 0.5$.

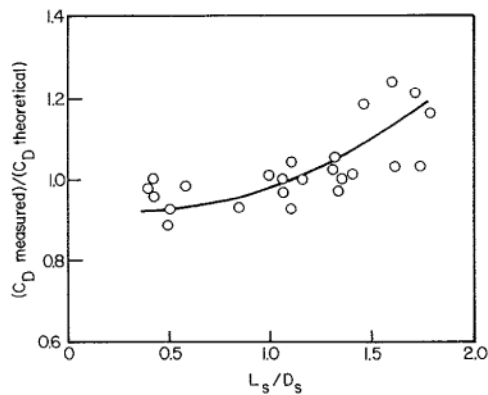


Figure 5.12 Influence of swirl chamber dimensions on discharge coefficient (Carlisle [22]).

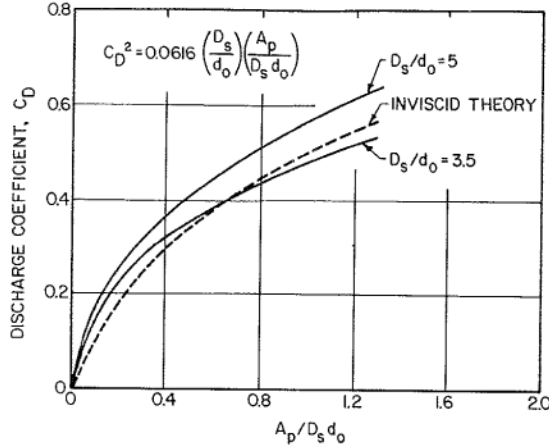


Figure 5.13 Practical relationship between discharge coefficient and atomizer dimensions.

Rizk and Lefebvre [24] derived the following relationship for C_D :

$$C_D = 0.35 \left(\frac{A_p}{D_s d_o} \right)^{0.5} \left(\frac{D_s}{d_o} \right)^{0.25} \quad (5.32)$$

This relationship is illustrated in Fig. 5.14.

Design considerations. Frictional losses are a major consideration in the design of swirl atomizers. They have two opposing effects on discharge coefficient. First, frictional losses represent a wasteful dissipation of atomization energy, which reduces the effective pressure drop across the atomizer and also the discharge coefficient. Second, by impeding the rotating flow in the swirl chamber, friction reduces the diameter of the air core and thereby increases the discharge coefficient. The relative importance of these two opposing effects depends mainly on various geometric features, which are discussed below.

Ratio of swirl chamber diameter to final orifice diameter. The effect of D_s/d_o on discharge coefficient is illustrated in Fig. 5.11, which shows that C_D increases with increase in D_s/d_o . However, Carlisle [22] pointed out that D_s/d_o should be kept small to reduce frictional losses and suggested that D_s/d_o should not exceed 5.0. A similar view was taken by Tipler and Wilson [25], who recommended a value of 2.5. However, in the absence of conflicting considerations, a D_s/d_o ratio of 3.3 has much to commend it, since it is consistent with the recommendations of Carlisle, Tipler, and Wilson and has the additional advantage, as shown in Fig. 5.11, that the minimum deviation between the theoretical and measured values of C_D occurs when $D_s/d_o = 3.3$.

Length/diameter ratio of swirl chamber. This should be kept short to minimize frictional losses. However, sufficient length must be provided for the separate jets

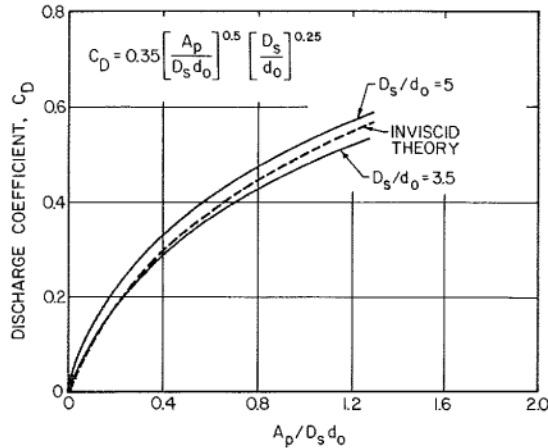


Figure 5.14 Practical relationship between discharge coefficient and atomizer dimensions (*Rizk and Lefebvre [24]*).

issuing from the swirl ports to coalesce into a uniform vortex sheet. In most current designs the L_s/D_s ratio lies between 0.5 and 1.0, although it has been suggested that higher values of L_s/D_s , up to a maximum of 2.75, would result in improved atomization [26].

Length/diameter ratio of final orifice. The high frictional losses incurred in this passage call for the shortest possible length. On large atomizers l_o/d_o can be made as small as 0.2, but with small atomizers the difficulty of manufacturing small-scale components to the required degree of accuracy usually dictates minimum values for l_o/d_o of around 0.5.

Length/diameter ratio of swirl ports. Tipler and Wilson [25] recommended that this ratio should not be less than 1.3, as short slots discharge the liquid in a diffused manner and may result in an uneven spray.

Manufacture. Good atomization and uniform drop distribution can only be obtained with an atomizer that is accurately made and well finished. The various faults that can arise in manufacture and impair atomizer performance have been fully described by Joyce [27]. It is also important to ensure that nozzle orifices do not get damaged during installation.

Comprehensive equation for discharge coefficient. From the foregoing discussion it is clear that the discharge coefficient is not solely dependent on the dimensionless parameters $A_p/D_s d_o$ and D_s/d_o . Although the simply derived equations for C_D based on these terms hold reasonably well in practice, if more precise values are needed then consideration must also be given to some of the other

geometric ratios discussed above, namely L_s/D_s and l_o/d_o , which can have a small but significant effect on C_D .

Jones [20] conducted a detailed experimental investigation on the effects of liquid properties, operating conditions, and atomizer geometry on discharge coefficient. The effect of atomizer geometry was examined by using a specially designed, three-piece, large atomizer. Several versions of these three constituent pieces were manufactured, allowing a total of 159 different atomizer configurations to be constructed. In practice, however, it was found necessary to use only a small proportion of the total number of possible atomizer configurations to cover a satisfactory range of each of the geometric dimensionless groups.

Table 5.1 illustrates the range over which each nondimensional group was investigated and compares the values with those typical of the large pressure-swirl atomizers employed in power generation by the Central Electricity Generating Board in the United Kingdom.

From analysis of the experimental data, Jones [20] obtained the following empirical equation for discharge coefficient:

$$C_D = 0.45 \left(\frac{d_o \rho_L U}{\mu_L} \right)^{-0.02} \left(\frac{l_o}{d_o} \right)^{-0.03} \left(\frac{L_s}{D_s} \right)^{0.05} \left(\frac{A_p}{D_s d_o} \right)^{0.52} \left(\frac{D_s}{d_o} \right)^{0.23} \quad (5.33)$$

In this equation it is of interest to note that the exponents for the two main

Table 5.1 Range of values of nondimensional groups covered by Jones [20]

Dimensionless group	Range covered	Typical value	
$\frac{l_o}{d_o}$	0.1–0.9	0.15	
$\frac{L_s}{D_s}$	0.31–1.26	0.7	
$\frac{L_p}{D_p}$	0.79–3.02	1.2	
$\frac{A_p}{d_o D_s}$	0.19–1.21	0.52	
$\frac{D_s}{d_o}$	1.41–8.13	2.7	
$\frac{d_o \rho_L U^2}{\sigma}$	11.5×10^3 – 3.55×10^5	Low pressure 2.4 MPa (350 psi)	High pressure 6.3 MPa (900 psi)
		1.08×10^5	3.88×10^5
$\frac{d_o \rho_L U}{\mu_L}$	1.913×10^3 – 21.14×10^3	6.45×10^3	23.64×10^3
$\frac{\mu_L}{\mu_A}$	279–2235	750	
$\frac{\rho_L}{\rho_A}$	694–964	700	

terms, $A_p/D_s d_o$ and D_s/d_o , are very close to those in Eq. (5.32). In Eq. (5.33) these exponents are 0.52 and 0.23, as compared with 0.5 and 0.25 in Eq. (5.32).

Babu et al. [28] followed Eisenklam [23] and Dombrowski and Hassan [19] in assuming a vortex flow pattern within the nozzle of the form

$$vr^n = \text{constant} \quad (5.34)$$

Their theoretical study, supplemented by analysis of experimental data, led to the following empirical expression for discharge coefficient:

$$C_D = \frac{K_{cd}}{[1/(1-X)^2 + (\pi/4B)^2/X^{0.5}]} \quad (5.35)$$

where

$$K_{cd} = 7.3423 \frac{A_o^{0.13735} A_s^{0.07782}}{A_p^{0.041066}}$$

$$B = \frac{A_p}{D_m d_o} \left(\frac{D_m}{d_o} \right)^{1-n}$$

$$n = 17.57 \frac{A_o^{0.1396} A_p^{0.2336}}{A_s^{0.1775}} \quad \text{for } \Delta P_L > 2.76 \text{ MPa (400 psi)}$$

$$n = 28 \frac{A_o^{0.14176} A_p^{0.27033}}{A_s^{0.17634}} \quad \text{for } \Delta P_L = 0.69 \text{ MPa (100 psi)}$$

The main advantage of Eq. (5.35) is that it is supported by a considerable body of experimental data obtained with several different atomizers. However, it takes no account of variations in the geometric parameters L_s/D_s and l_o/d_o . Furthermore, it demands a knowledge of the air core area A_a to calculate the value of X for insertion into Eq. (5.35). Such information is not always readily available.

Film Thickness

In pressure-swirl atomizers the liquid emerges from the nozzle as a thin conical sheet that rapidly attenuates as it spreads radially outward, finally disintegrating into ligaments and then drops. In the prefilming airblast atomizer the liquid is also spread out into a thin continuous sheet before being exposed to the high-velocity air. It is of interest, therefore, to examine the factors that govern the thickness of this liquid film.

Theory. For both pressure-swirl and airblast types of atomizers, it has long been recognized that the thickness of the annular liquid film produced at the nozzle exit has a strong influence on the mean drop size of the spray [29]. In pressure-swirl atomizers the thickness of the liquid film in the final orifice is directly related to the area of the air core. Giffen and Muraszew's analysis of the flow conditions

within a simplex nozzle, assuming a nonviscous fluid, led to the relationship between atomizer dimensions and the size of the air core expressed in Eq. (5.25). Substituting in this equation for $K_1 = 4A_p/\pi D_s d_o$ gives

$$\left(\frac{A_p}{D_s d_o}\right)^2 = \frac{\pi^2 (1-X)^3}{32 X^2} \quad (5.36)$$

where X is the ratio of the area of the air core to the area of the final discharge orifice.

After calculating X from Eq. (5.36), the corresponding value of liquid film thickness t can readily be derived, since from geometric considerations

$$X = \frac{(d_o - 2t)^2}{d_o^2} \quad (5.37)$$

In Eq. (5.27) the nozzle discharge coefficient is expressed as a unique function of X . In Eq. (5.32) the discharge coefficient is expressed in terms of nozzle dimensions. Combining these two equations yields the following expression, from which X and hence t can be calculated from the nozzle dimensions:

$$\frac{(1-X)^3}{1+X} = 0.09 \left(\frac{A_p}{D_s d_o}\right) \left(\frac{D_s}{d_o}\right)^{0.5} \quad (5.38)$$

Simmons and Harding [30] derived the following expression for t :

$$t = \frac{0.48FN}{d_o \cos \theta} \quad (5.39)$$

In Eq. (5.39) it should be noted that film thickness is expressed in micrometers, FN is the nozzle flow number in $(\text{lb/h})/(\text{psid})^{0.5}$ for standard calibrating fluid (MIL-C-7024II), d_o is the discharge orifice diameter in inches, and θ is half the included spray angle in degrees.

In SI units Eq. (5.39) becomes

$$t = \frac{0.00805 \sqrt{\rho_L} FN}{d_o \cos \theta} \quad (5.40)$$

The implication of Eqs. (5.36), (5.38), and (5.40) is that film thickness is independent of liquid viscosity and liquid injection pressure.

Rizk and Lefebvre [24] used a theoretical approach to investigate the internal flow characteristics of pressure-swirl atomizers. In particular, they examined the effects of atomizer dimensions and operating conditions on spray cone angle, velocity coefficient, and the thickness of the annular liquid film formed at the discharge orifice. A general expression for film thickness was derived in terms of atomizer dimensions, liquid properties, and liquid injection pressure as

$$t^2 = \frac{1560 \dot{m}_L \mu_L}{\rho_L d_o \Delta P_L} \frac{1+X}{(1-X)^2} \quad (5.41)$$

or, by substituting $FN = \dot{m}_L / (\Delta P_L \rho_L)^{0.5}$

$$t^2 = \frac{1560 FN \mu_L (1 + X)}{\rho_L^{0.5} d_o \Delta P_L^{0.5} (1 - X)^2} \quad (5.42)$$

Since X is dependent on t —in fact, $X = A_a/A_o = (d_o - 2t)^2/d_o^2$ —some trial-and-error procedures are involved in the solution of Eqs. (5.41) and (5.42).

Rizk and Lefebvre [24] used Eq. (5.41) to calculate film thicknesses for different nozzle dimensions and operating conditions. Their results are shown in Figs. 5.15 to 5.19 as plots of film thickness against injection pressure differential. Also shown in these figures are the corresponding measured values of Kutty et al. [31, 32]. Theory and experiment both indicate that a higher nozzle pressure drop produces a thinner liquid sheet. Thus the improvement in atomization quality that always accompanies an increase in nozzle pressure drop, which is usually attributed to increase in liquid discharge velocity, may also be due in part to the decrease in film thickness caused by the increase in nozzle pressure drop, as shown in Figs. 5.15 to 5.19.

Increase in final orifice diameter leads to a thicker film, as shown in Fig. 5.15. This is because increase in d_o lowers C_D [see, for example, Eqs. (5.28) to (5.33)]. The variation of film thickness with liquid ports area is shown in Fig. 5.16. Increasing the inlet area raises the flow rate through the nozzle, which re-

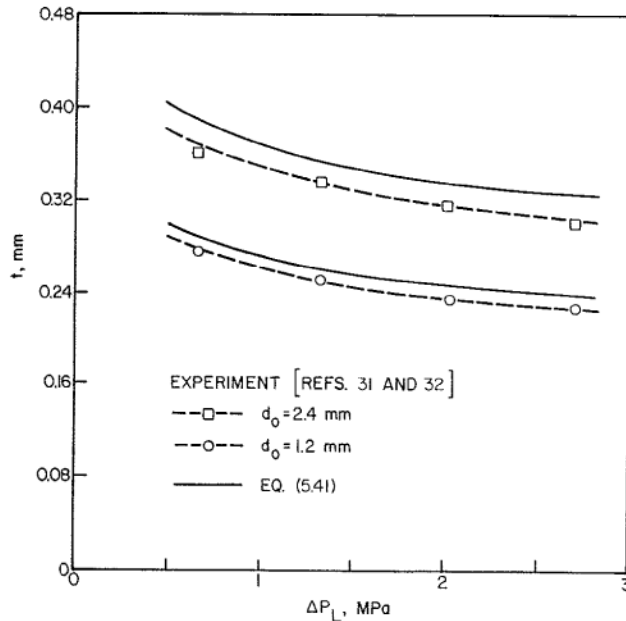


Figure 5.15 Variation of film thickness with injection pressure for different orifice diameters (Rizk and Lefebvre [24]).

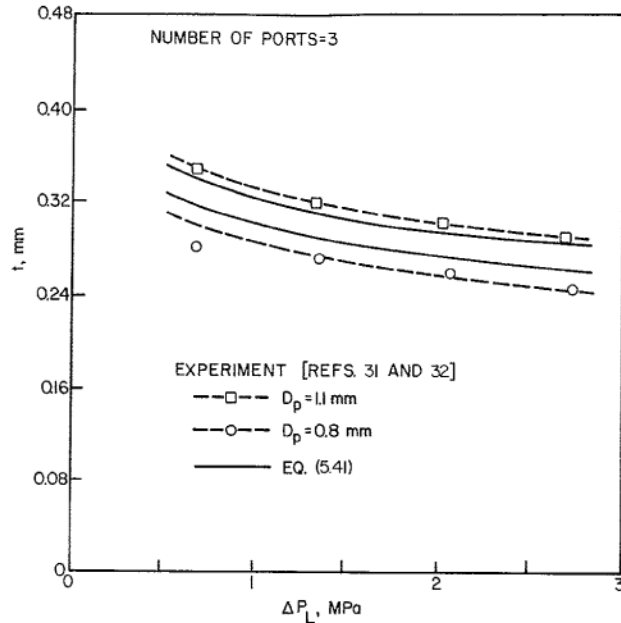


Figure 5.16 Variation of film thickness with injection pressure for different inlet port diameters (Rizk and Lefebvre [24]).

sults in a thicker film. The effect of a reduction in swirl chamber diameter is to increase the liquid film thickness, as shown in Fig. 5.17. This is attributed to the lower swirl action, which reduces the diameter of the air core within the final discharge orifice. The effects of orifice length and swirl chamber length on film thickness are quite small, as illustrated in Figs. 5.18 and 5.19.

Equation (5.41) was also used to determine values of liquid film thickness for the SMD data reported in [24, 30, 33]. The results are plotted in Fig. 5.20; they indicate that $SMD \propto t^{0.39}$. This exponent 0.39 is almost identical to that obtained by Rizk and Lefebvre [34] for prefilming airblast atomizers and also agrees closely with the conclusion reached by Simmons [35].

Rizk and Lefebvre [24] have suggested that for $t/d_o \ll 1$ Eq. (5.42) can be written more succinctly, while still retaining all its essential features, as

$$t = 3.66 \left[\frac{d_o F N \mu_L}{(\Delta P_L \rho_L)^{0.5}} \right]^{0.25} \quad (5.43)$$

The above equation provides useful guidance on the effects of atomizer characteristics and liquid flow properties on film thickness and on the mean drop size of the ensuing spray. It is of interest to note that surface tension does not appear in the above expression for film thickness although, of course, it does play a major role in the subsequent breakup of the liquid sheet into ligaments and drops. The

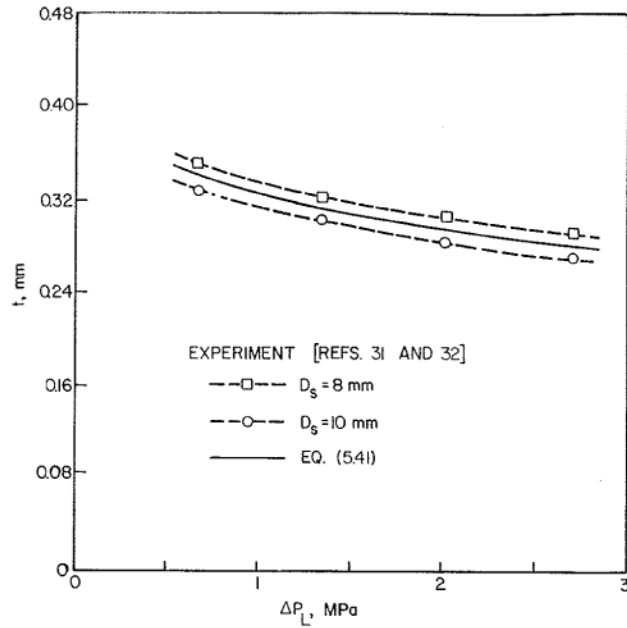


Figure 5.17 Variation of film thickness with injection pressure for different swirl chamber diameters (Rizk and Lefebvre [24]).

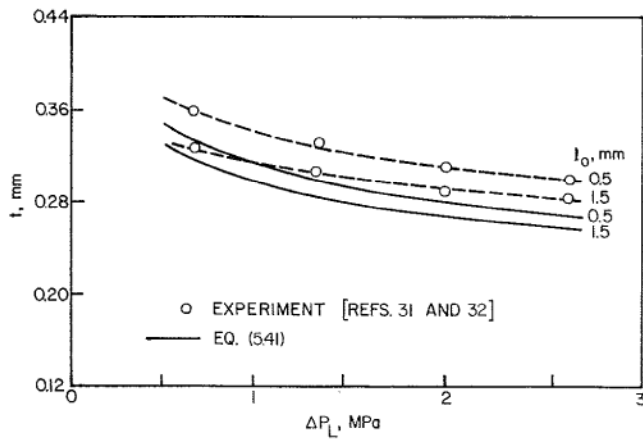


Figure 5.18 Variation of film thickness with injection pressure for different orifice lengths (Rizk and Lefebvre [24]).

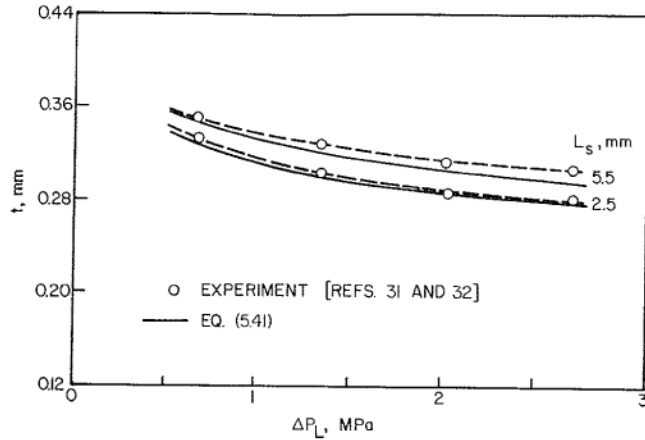


Figure 5.19 Variation of film thickness with injection pressure for different swirl chamber lengths (Rizk and Lefebvre [24]).

influence of liquid viscosity is clearly of major importance in the atomization process, because viscous forces impede atomization in two ways, by increasing the initial film thickness and by resisting the disintegration of the liquid sheet into drops. Equation (5.43) shows that the effect of liquid density on film thickness is quite small ($t \propto \rho_L^{-0.125}$). Thus its influence on atomization quality should also be small, and this is confirmed by the results of measurements of mean drop size for pressure-swirl atomizers [36].

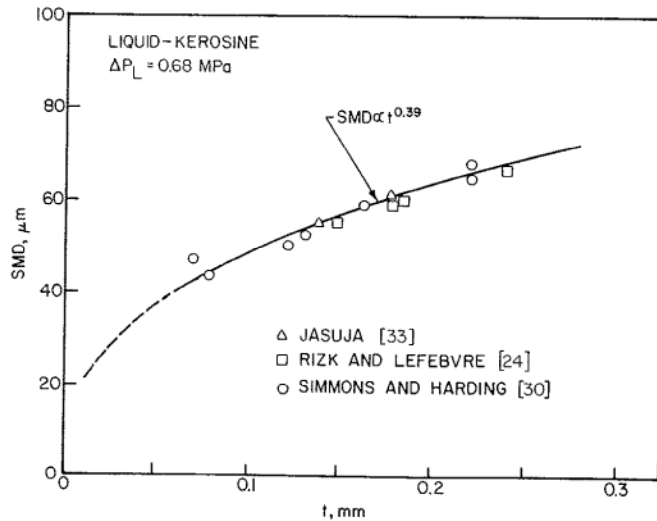


Figure 5.20 Variation of Sauter mean diameter with liquid film thickness (Rizk and Lefebvre [24]).

By making the substitution

$$FN = \frac{\dot{m}_L}{(\Delta P_L \rho_L)^{0.5}}$$

Eq. (5.43) becomes

$$t = 3.66 \left(\frac{d_o \dot{m}_L \mu_L}{\rho_L \Delta P_L} \right)^{0.25} \quad (5.44)$$

This equation indicates that liquid film thickness increases with increases in nozzle size, liquid flow rate, and liquid viscosity and diminishes with increase in liquid density and/or liquid injection pressure.

Experiment. Several attempts have been made to measure liquid film thickness in the final orifice of a pressure-swirl atomizer. In some experiments the film thickness was measured directly; in others it was obtained as half the difference between the measured air core diameter and the orifice diameter.

Kutty et al. [31] used a photographic technique to investigate the influence of liquid pressure differential on air core size. Photographs were taken with the camera pointing upstream through the nozzle outlet orifice. Illumination was achieved by fitting a transparent window at the rear of the swirl chamber. Air core diameters were measured from the negatives by enlarging them in a micro-film reader having a total magnification of 100.

Suyari and Lefebvre [37] used a method based on measurement of electrical conductance. The procedure is to flow water through the atomizer and measure the electrical conductance between two electrodes located in the discharge orifice. As the electrical conductivity of water is known, this measurement provides a direct indication of the average liquid film thickness in the flow path between the two electrodes.

The system is calibrated by flowing water through the nozzle and measuring electrical conductance with a plastic rod of low electrical conductivity inserted along the axis of the nozzle discharge orifice. By repeating this measurement with rods of different diameter, a calibration curve can be drawn to relate voltmeter reading to film thickness.

To examine the effect of variations in nozzle geometry on liquid film thickness, a number of inserts were produced to provide four different values of inlet port diameter, namely 0.749, 1.143, 1.245, and 1.346 mm. The results for liquid film thickness obtained with these different nozzle geometries are shown in Fig. 5.21, in which film thickness is plotted against water injection pressure differential for an ambient air pressure of 1 atm.

Comparison of theory and experiment. It is clearly of interest to compare the measured values of liquid film thickness with the corresponding predicted values from Eqs. (5.36), (5.38), (5.39), and (5.43). This is done in Fig. 5.22 for conditions of normal atmospheric air pressure and a liquid pressure differential of 0.69 MPA (100 psi).

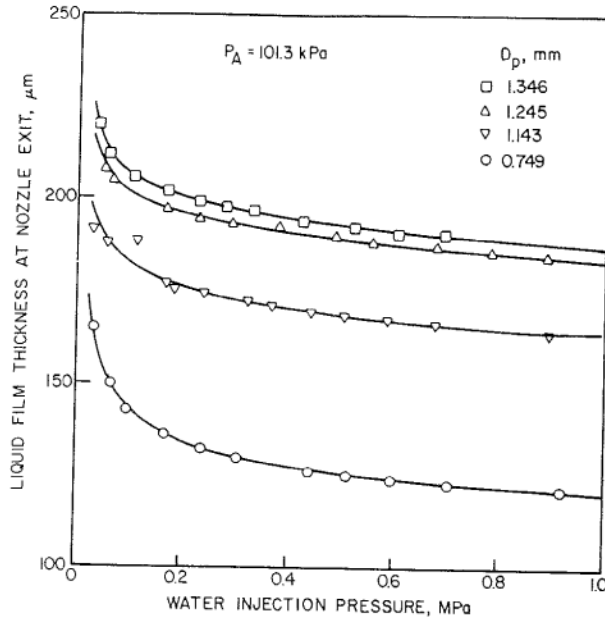


Figure 5.21 Influence of water injection pressure and inlet port diameter on liquid film thickness in final orifice (Suyari and Lefebvre [37]).

Inspection of Fig. 5.22 shows good agreement between the experimental data and values of t calculated using the simple inviscid relationship between atomizer dimensions and air core size provided by Giffen and Muraszew [4] [see Eq. (5.36)]. However, Eq. (5.38) provides the best fit to the experimental data. The equation proposed by Simmons and Harding [Eq. (5.39)] also demonstrates good agreement with the measured values.

The predictive capability of Eq. (5.43), due to Rizk and Lefebvre [24], is less satisfactory. This may be because the constant 3.66 in this equation was chosen to provide the best fit to the data of Kutty et al. [31]. Their photographic method measures the *maximum* film thickness to be found anywhere in the final orifice. The presence of ripples or undulations on the liquid surface could cause their measured values of t to be higher than those obtained by the conductance method, which measures the *average* film thickness in the final orifice. If the average values of t as determined by Suyari and Lefebvre [37] are used to obtain the constant in Eq. (5.43), it becomes

$$t = 2.7 \left[\frac{d_o F N \mu_L}{(\Delta P_L \rho_L)^{0.5}} \right] \quad (5.45)$$

This equation demonstrates a marked improvement in predictive capability, as indicated by the dashed line in Fig. 5.22, but the level of agreement with experimental data is still not as good as that obtained with Eq. (5.38).

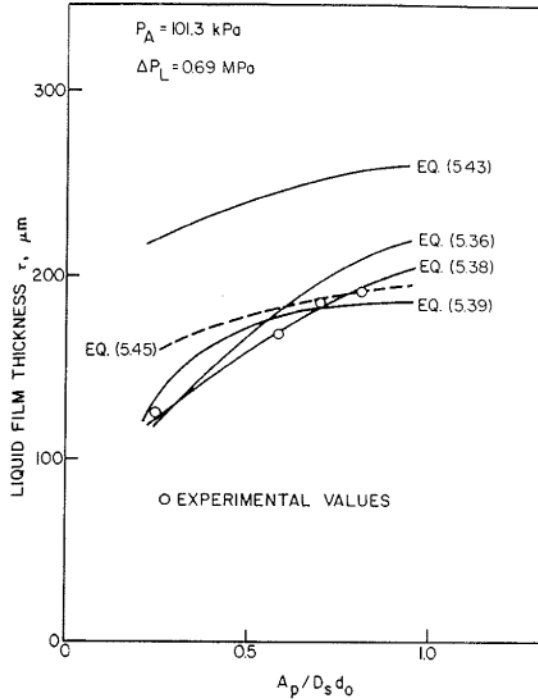


Figure 5.22 Comparison of measured values of film thickness with theoretical predictions (Suyari and Lefebvre [37]).

Flow Number

The flow number of an atomizer is easily determined. The liquid flow rate $\dot{m}_L$ is obtained by measuring the time required to pass a given volume (or mass) of liquid while maintaining the pressure differential across the nozzle at a constant value. Usually, the pressure differential is 0.69 MPa (100 psi) and the liquid is standard calibrating fluid (MIL-C-7024II), which has a density of 765 kg/m^3 at room temperature. These values of $\dot{m}_L$, ΔP_L , and ρ_L are then substituted into Eq. (5.3) to obtain the flow number.

The influence of nozzle dimensions on flow number was studied experimentally by Kutty et al. [31, 38]. They manufactured a large number of simplex nozzles, all having three equispaced inlet ports, in a manner designed to elucidate the effect of variations in each of the key dimensions on nozzle flow characteristics. Rizk and Lefebvre's [24] analysis of their data yielded the following empirical expression for nozzle flow number:

$$FN = 0.0308 \left(\frac{A_p^{0.5} d_o}{D_s^{0.45}} \right) \quad (5.46)$$

Figures 5.23 to 5.27 show the variations in flow number with changes in nozzle dimensions as given by this equation, along with corresponding experimental results for kerosine from references [31] and [38]. These figures demonstrate the ability of Eq. (5.46) to predict the effect on nozzle flow number of wide variations in atomizer dimensions.

Equation (5.46) should apply also to nozzles having other than three inlet ports, but the constant (0.0308) may require modification.

Considering the various nozzle dimensions in Eq. (5.46), it is reasonable to expect that higher values of final orifice diameter and inlet port area should increase the liquid flow rate for any given injection pressure differential, due to the increase in available flow area. The effect of increasing swirl chamber diameter in reducing flow rate, as indicated in Eq. (5.46) and demonstrated in Fig. 5.25, is due to the higher swirl action, which enlarges the air core diameter and thereby reduces the effective flow area of the final orifice. The frictional losses in the nozzle are relatively small, so that the lengths of the swirl chamber and final orifice have little effect on flow number.

Equation (5.46) can be modified to a dimensionally correct form as

$$FN = 0.395 \left(\frac{A_p^{0.5} d_o^{1.25}}{D_s^{0.25}} \right) \quad (5.47)$$

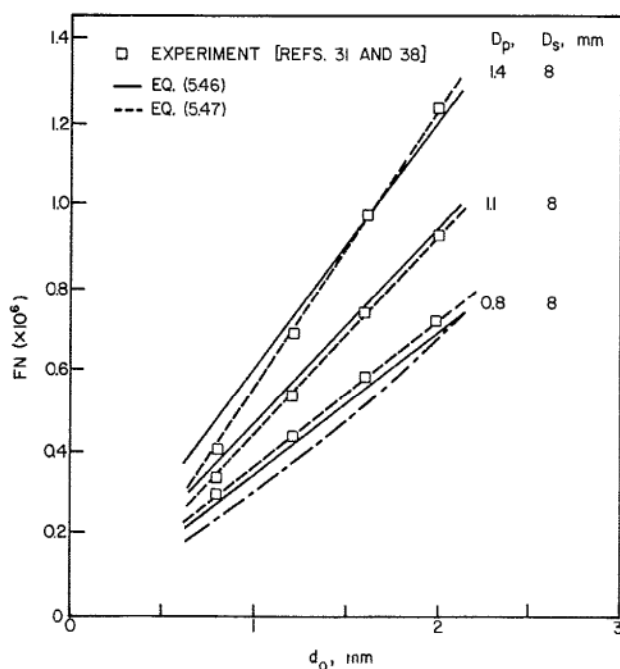


Figure 5.23 Influence of discharge orifice diameter on nozzle flow number (Rizk and Lefebvre [24]).

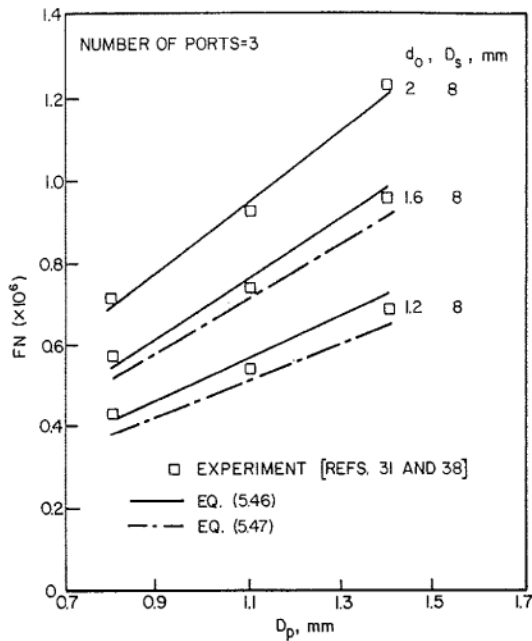


Figure 5.24 Influence of inlet port diameter on nozzle flow number (Rizk and Lefebvre [24]).

Although this equation does not predict the measured flow numbers to quite the same degree of accuracy as Eq. (5.46), the level of agreement between theory and experiment is still fairly close, as illustrated in Figs. 5.23 to 5.27.

Velocity Coefficient

This is defined as the ratio of the actual discharge velocity to the theoretical velocity corresponding to the total pressure differential across the nozzle, i.e.,

$$K_v = \frac{U}{(2\Delta P_L / \rho_L)^{0.5}} \quad (5.48)$$

Accurate values of K_v are essential for calculating the true velocity at which the liquid is discharged from the nozzle. This velocity is of prime importance to atomization, since it is the relative velocity between the initial liquid sheet and the surrounding air or gas that largely determines the mean drop size of the spray.

It has been shown [39] that K_v can be related to nozzle dimensions and spray cone angle by the expression

$$K_v = \frac{C_D}{(1 - X) \cos \theta} \quad (5.49)$$

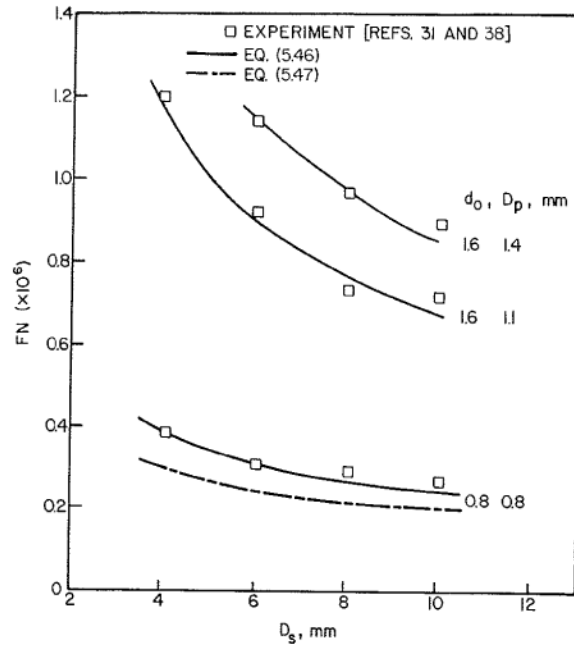


Figure 5.25 Influence of swirl chamber diameter on nozzle flow number (Rizk and Lefebvre [24]).

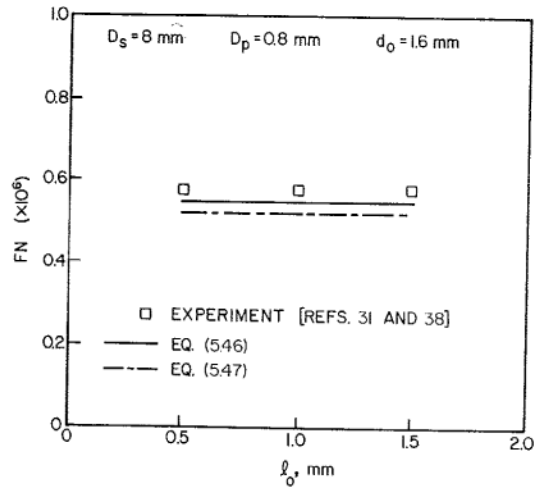


Figure 5.26 Influence of discharge orifice length on nozzle flow number (Rizk and Lefebvre [24]).

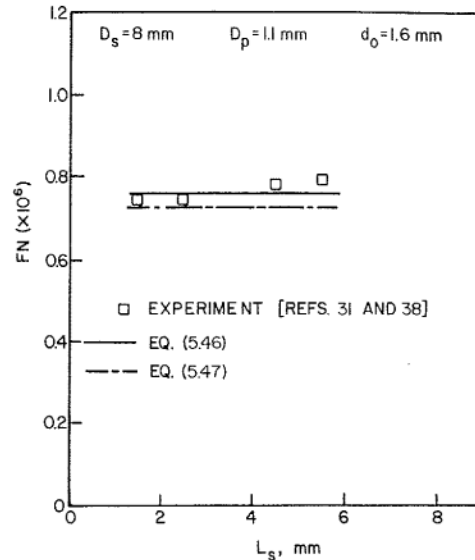


Figure 5.27 Influence of swirl chamber length on nozzle flow number (Rizk and Lefebvre [24]).

Rizk and Lefebvre [39] used the available experimental data on cone angle [31], air core size [32], and liquid flow rate [38] to calculate the velocity of the liquid discharging from the nozzle for different atomizer dimensions and different liquid pressure differentials. The ratio of the theoretical pressure drop corresponding to the discharge velocity to the actual measured pressure drop across the nozzle is plotted against final orifice diameter in Fig. 5.28. The figure shows that the ratio of "effective" injection pressure to the actual injection pressure decreases rapidly with increase in orifice diameter. This suggests that for two atomizers of similar size and geometry, differing only in final orifice diameter, the smaller one will produce a higher discharge velocity for the same pressure drop. It is also clear from Fig. 5.28 that with increase in pressure differential the atomizer becomes more efficient in converting the available pressure drop into discharge velocity.

Equation (5.49) was also used to examine the effect on K_v of variations in nozzle dimensions. Calculated values of film thickness, cone angle, and discharge coefficient from Eqs. (5.44), (7.18), and (5.32), respectively, were used in conjunction with measured values of liquid flow rate and nozzle pressure differential to produce the results shown in Figs. 5.29 to 5.33. Figure 5.29 shows that increase in orifice diameter causes a reduction in velocity coefficient. This is because the beneficial effect on K_v of increasing X is outweighed by the corresponding reduction in C_D . The improvement in velocity coefficient at higher levels of ΔP_L is due to the reduction in film thickness.

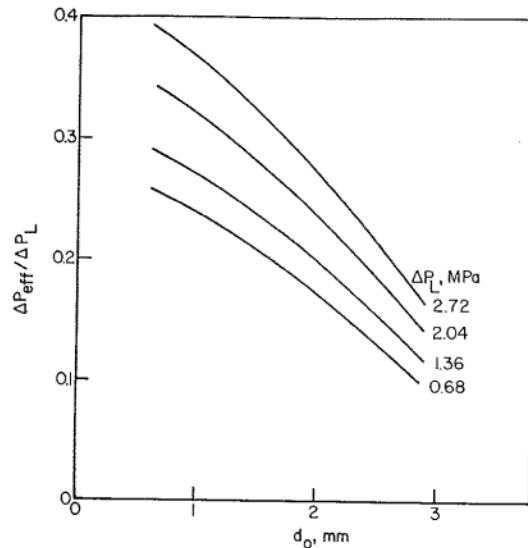


Figure 5.28 Influence of final orifice diameter on effective injection pressure (Rizk and Lefebvre [39]).

The higher values of K_v that accompany increases in inlet port diameter, as illustrated in Fig. 5.30, are attributed to higher discharge coefficients, which more than compensate for increases in liquid film thickness. Figure 5.31 illustrates the variation of K_v with swirl chamber diameter. It shows that K_v declines rapidly at first with increase in D_s , but the curves then level off to fairly constant values. These effects can also be explained in the light of Eqs. (5.32) and (5.49).

The variation of velocity coefficient with atomizer constant K is illustrated in Figs. 5.32 and 5.33. The curves drawn in Fig. 5.32 are the result of calculations, while those in Fig. 5.33 are based on actual experimental data. Both figures show that higher values of K and ΔP_L yield higher values of K_v . Comparison of the two figures helps to confirm the validity of Eq. (5.49) for calculating K_v , agreement between theory and experiment being generally satisfactory, especially for low values of K and ΔP_L .

It is well known that viscosity forces impair spray quality by opposing the disintegration of the liquid jets and sheets produced by various types of atomizers, but less attention has been paid to the other adverse effect of viscosity on atomization, which stems from its strong influence on the flow processes occurring within the nozzle itself. Calculated results on the effect of viscosity on K_v [40] show that K_v is markedly reduced by an increase in liquid viscosity, due to the combined effects of increase in film thickness and reduction in cone angle. However, when the film thickness has developed to such an extent that the air core diameter is quite small, changes in X and θ become less significant than increase in discharge coefficient. This causes the decline of K_v to slow down; in fact, K_v

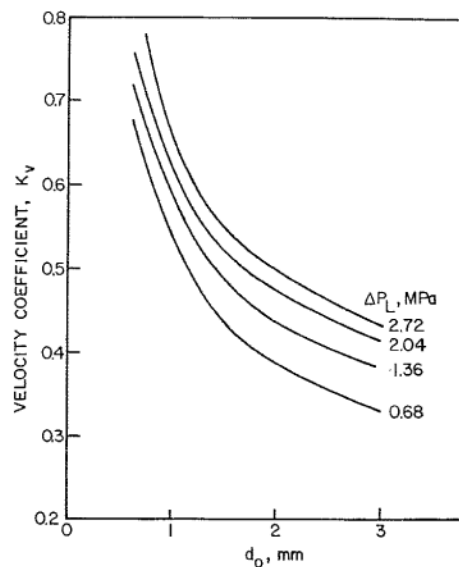


Figure 5.29 Influence of final orifice diameter on velocity coefficient (Rizk and Lefebvre [39]).

even rises a little until a level of viscosity is reached at which the discharge orifice is running full. At this condition K_v has a value of around 0.34, regardless of ΔP_L .

This study by Rizk and Lefebvre [40] clearly demonstrates that increase in liquid viscosity impedes atomization, not only by opposing the breakup of the liquid sheet into drops but also by thickening the liquid film in the final orifice

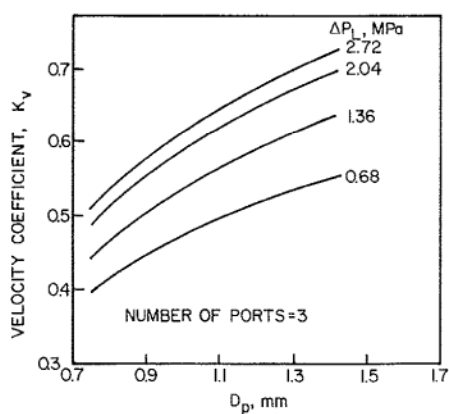


Figure 5.30 Effect of inlet port diameter on velocity coefficient (Rizk and Lefebvre [39]).

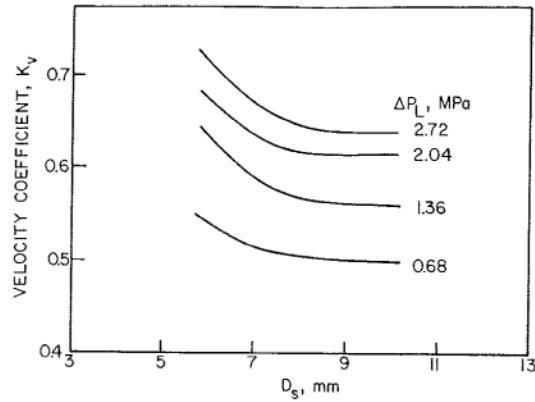


Figure 5.31 Effect of swirl chamber diameter on velocity coefficient (*Rizk and Lefebvre [39]*).

and by reducing the efficiency of the processes whereby the nozzle pressure differential is converted into kinetic energy at the nozzle exit. The results of their analysis show that the main factors governing velocity coefficient are nozzle geometry, nozzle pressure differential, and the liquid properties of density and viscosity. It is found that values of the velocity coefficient, as calculated for wide ranges of atomizer dimensions and liquid properties, conform closely to the empirical expression

$$K_v = 0.00367 K^{0.29} \left(\frac{\Delta P_L \rho_L}{\mu_L} \right)^{0.2} \quad (5.50)$$

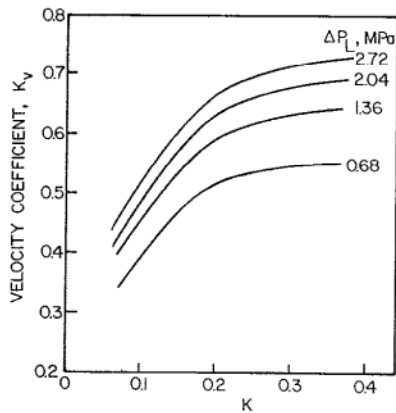


Figure 5.32 Calculated variation of velocity coefficient with atomizer coefficient (*Rizk and Lefebvre [39]*).

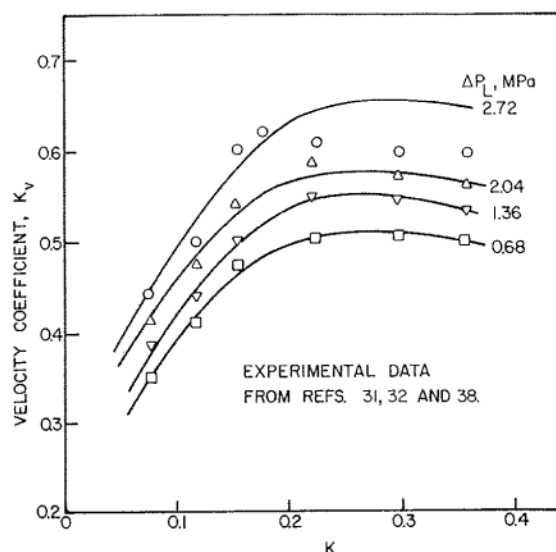


Figure 5.33 Experimental data on variation of velocity coefficient with atomizer constant (Rizk and Lefebvre [39]).

ROTARY ATOMIZER

With this device, liquid is supplied to the center of a rotating disk or cup. The friction between the liquid and the cup wall causes the liquid to rotate at roughly the same speed as the cup. This rotary motion creates centrifugal forces within the liquid that induce it to flow radially outward toward the rim of the cup. If the rotational speed of the cup is sufficiently high, the liquid will arrive at the rim in a thin continuous film. The manner in which this film disintegrates into drops will depend on the size and geometry of the cup, its rotational speed, the liquid flow rate, and the physical properties of the liquid [41–44].

It is instructive to consider what happens when the liquid flow rate to a cup rotating at constant speed is gradually increased from zero. At low flow rates the liquid spreads out across the surface and is centrifuged off in the form of drops of sensibly uniform size. This phenomenon is generally known as *direct drop formation*. If the flow rate is increased, ligaments are formed along the periphery and then disintegrate into drops. The number of ligaments increases with increasing flow rate up to a maximum value, beyond which it remains constant, regardless of flow rate. The thickness of the ligaments also increases with flow rate in this mode of atomization [45]. The ligaments themselves are unstable and disintegrate into drops at some distance from the periphery, as illustrated in Fig. 5.34. This process is usually termed *atomization by ligament formation*.

With continuing increase in flow rate a condition is finally reached where the ligaments have attained their maximum number and size and can no longer ac-

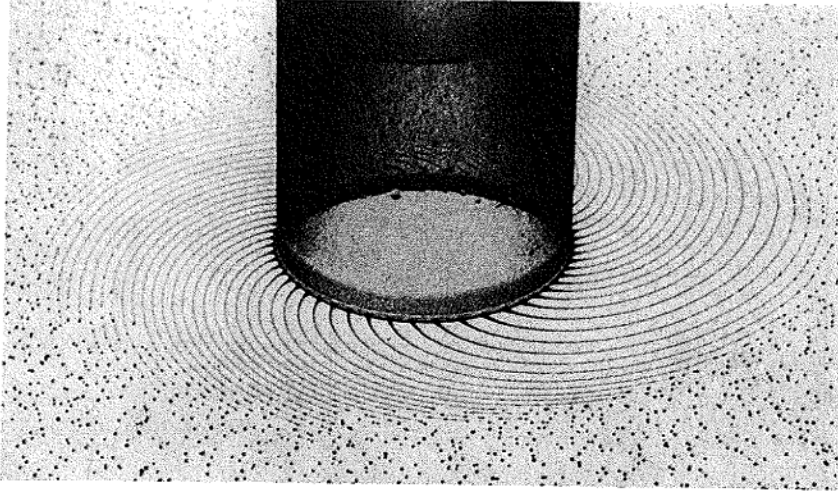


Figure 5.34 Illustration of ligament formation and disintegration into drops (Courtesy of P. Eisenklam [55]).

commodate the flow of liquid. A thick film is produced that extends outward to some distance beyond the cup rim. When this film disintegrates into ligaments, it does so in an irregular manner, which results in an appreciable variation in drop size. This process is commonly referred to as *atomization by film formation*.

Critical Flow Rates

Empirical formulas for calculating the critical flow rates corresponding to the transition from one mode of atomization to another have been provided by Tanasawa et al. [46], as follows.

Direct drop formation:

$$q \leq 2.8 \left(\frac{D}{n} \right)^{2/3} \left(\frac{\sigma}{\rho_L} \right) \left[1 + 10 \left(\frac{\mu_L}{(\rho_L \sigma D)^{0.5}} \right)^{1/3} \right]^{-1} \quad (5.51)$$

Ligament formation:

$$q = 80 \left(\frac{D}{n} \right)^{2/3} \left(\frac{\sigma}{\rho_L} \right) \left[1 + 10 \left(\frac{\mu_L}{(\rho_L \sigma D)^{0.5}} \right)^{1/3} \right]^{-1} \quad (5.52)$$

Film formation:

$$q \geq 5.3 \left(\frac{D}{n} \right)^{2/3} \left(\frac{\sigma}{\rho_L} \right) \left(\frac{\rho_L}{\mu_L} \right)^{1/3} \quad \text{for } \frac{D \rho_L}{\mu_L} < 30 \text{ s/cm} \quad (5.53)$$

$$q \cong 20(D)^{1/2} \left(\frac{1}{n} \right)^{2/3} \left(\frac{\sigma}{\rho_L} \right)^{5/6} \quad \text{for } \frac{D\rho_L}{\mu_L} > 30 \text{ s/cm} \quad (5.54)$$

where q = volume flow rate, cm^3/s
 D = disk diameter, cm
 n = rotational speed, rpm
 s = pitch of ligaments, cm
 μ_L = dynamic viscosity of liquid, dyn/cm^2
 ν_L = kinematic viscosity of liquid, cm^2/s
 ρ_L = liquid density, g/cm^3
 σ = surface tension, dyn/cm

Film Thickness

Hinze and Milborn [41] analyzed the radial velocity distribution in a thin film on a rotating cup by simply equating the viscous and centrifugal forces. Emslie et al. [47] carried out a similar analysis and included the effects of gravity. Oyama and Endou [48] derived expressions for both radial and tangential velocity distributions. Recently, more exact solutions have been developed [49]. Nikolaev et al. [50] solved the equations of motion for a liquid film on a rotating disk from considerations of the difference between the angular velocity of the film and that of the disk. Bruin [51] obtained simpler expressions for the velocity profiles by using the complex function method to solve the simplified Navier-Stokes equations. Matsumoto et al. [52] predicted the velocity profiles in a film on a rotating disk by numerical integration of nonlinear differential equations deduced from the Navier-Stokes equations.

Matsumoto et al. [49] have compared the solutions by Nikolaev et al. [50], Bruin [51], and Matsumoto et al. [52] for the radial velocity distribution for flow on flat disks. Their results, shown in Fig. 5.35, indicate only small differences between these solutions. Although the lag between the angular velocities of the liquid film and the rotating disk is usually considered quite large, these results suggest that the lag is not more than around 20%.

Theoretical solutions for the film thickness predicted by these results are compared with experimental data in Fig. 5.36. The level of agreement between these different solutions is clearly satisfactory.

Toothed Designs

It has been found that grooves cut into the wall of the cup can help to guide and regulate the flow as it is centrifugally forced to the outside. Additional benefits can be gained by placing teeth on the outer rim of the cup. Such teeth can help to maintain ligament drop formation and avoid film atomization (see Fig. 4.35). The optimum number of teeth depends on cup size, rotational velocity, and liquid

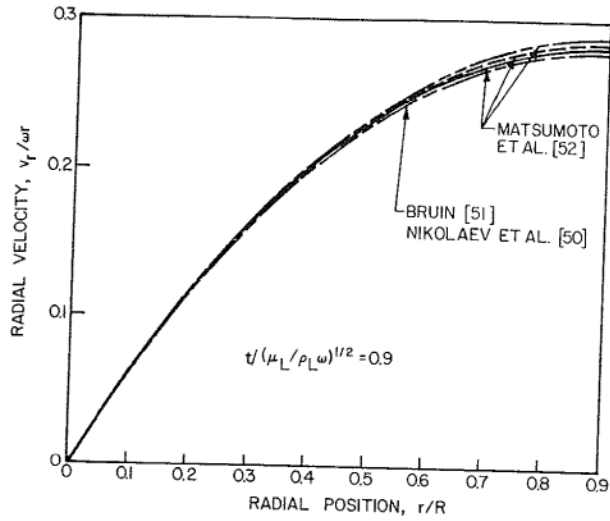


Figure 5.35 Predicted radial velocity distributions from flow on flat disks (Matsumoto et al. [49]).

properties. According to Christensen and Stealy [45], the optimum number of teeth may be calculated from the expression

$$z = 0.215 \left(\frac{\rho_L \omega^2 D^3}{\sigma} \right)^{5/12} \left(\frac{\rho_L \sigma D}{\mu_L^2} \right)^{1/6} \quad (5.55)$$

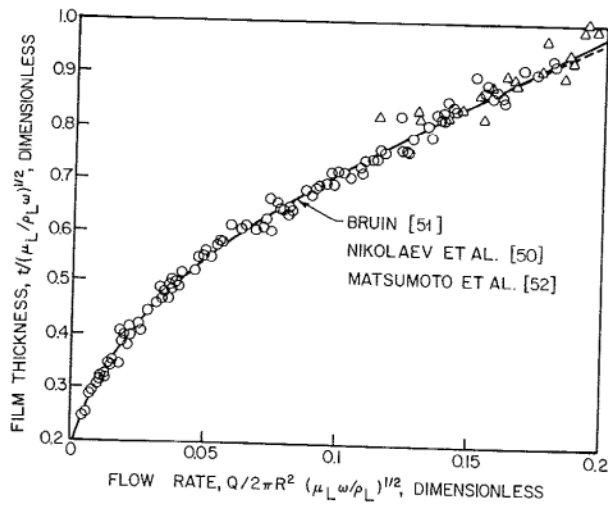


Figure 5.36 Predicted film thickness for flow on flat disks (Matsumoto et al. [49]).

This equation shows that the number of teeth needed to maintain ligament formation increases with increases in cup size and rotational velocity and is diminished by increases in surface tension and viscosity. Table 5.2 illustrates the number of teeth needed as calculated from Eq. (5.55).

Once the liquid reaches the teeth, the various modes of atomization discussed earlier can occur. These modes are illustrated in Fig. 5.37. They include direct drop formation from the teeth, combined drop and ligament formation, ligament formation, and film formation. This figure illustrates that "monodisperse" sprays can only be obtained with direct drop formation or ligament formation.

AIRBLAST ATOMIZER

For plain-jet airblast atomizers the internal hydrodynamics are essentially the same as those of the plain-orifice pressure atomizer. With prefilming airblast nozzles the situation is more complex, but the key factors involved in the attainment of good atomization have been known for some time. From detailed experimental studies carried out on several different atomizer configurations, Lefebvre and Miller [53] identified the following essential features of successful nozzle design.

1. The liquid should first be spread into a thin continuous sheet of uniform thickness.
2. Finest atomization is obtained by producing a liquid sheet of minimum thickness. In practice, this means that the atomizing lip should have the largest possible diameter.
3. The annular liquid sheet formed at the atomizing lip should be exposed on both sides to air at the highest possible velocity. Thus, the atomizer should be designed to achieve minimum loss of total pressure in the airflow passages and maximum air velocity at the atomizing lip.

In most prefilming nozzles the liquid flows through a number of equispaced

Table 5.2 Number of teeth required to maintain ligament formation as a function of rotational velocity and disk diameter [45]

Diameter (cm)	Rotational velocity (rpm)	No. teeth	No. teeth per centimeter
5	1000	37.6	2.4
5	3000	94.6	6.0
5	5000	145.4	9.3
10	1000	56.6	1.8
10	3000	142.5	4.5
10	5000	219.0	7.0
20	1000	85.2	1.4
20	3000	214.5	3.4
20	5000	330.0	5.2

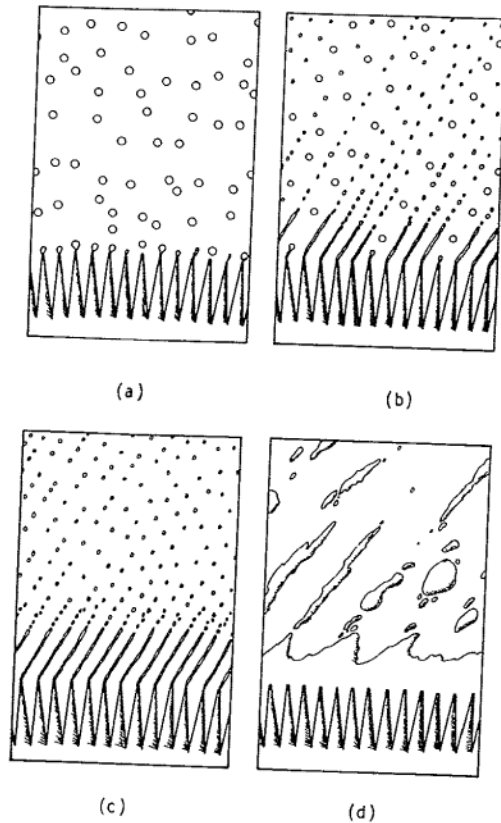


Figure 5.37 Illustration of rotary atomization, using a toothed rim. (a) Direct formation; (b) direct and ligament formation; (c) ligament formation; (d) sheet formation. (Christensen and Stealy [45].)

tangential ports into a weir, from which it spreads over a prefilming surface before being discharged at the atomizing lip. This arrangement is shown schematically in Fig. 5.38. The purpose of the weir is to protect the liquid from the high-velocity air until the discrete jets of liquid flowing into the weir have lost their separate identities and have merged to form a fully annular reservoir behind the weir. As the liquid flows out of this reservoir, it is exposed to the high-velocity airstream, which causes it to spread out over the prefilming surface to form a thin continuous film at the atomizing lip. Normally, this film retains a significant proportion of its tangential velocity component, so when it flows over the prefilming lip it is flung radially outward to form a hollow conical spray. This "natural" spray cone is, of course, observed only in the absence of atomizing air. Under normal operating conditions, the liquid film produced at the prefilmer lip is rapidly disintegrated by the atomizing air, which dictates the effective cone angle of the ensuing spray. Nevertheless, a wide natural spray angle is a desirable design feature

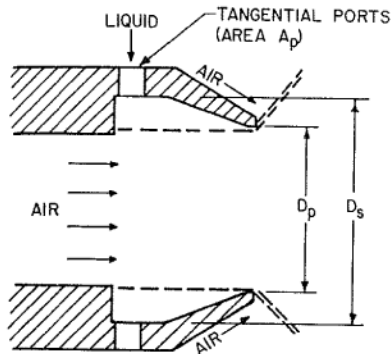


Figure 5.38 Schematic diagram of prefilmer.

because it increases the relative velocity between the liquid and the atomizing air and also increases the interaction between the liquid and the outer airstream. This is one of the reasons why it is customary, in the design of prefilming airblast atomizers, to arrange for the outer airflow rate to exceed the inner airflow rate.

Theoretical values of discharge coefficient for prefilming airblast atomizers are plotted against $A_p/D_s D_p$ in Fig. 5.39. The values shown are the results of

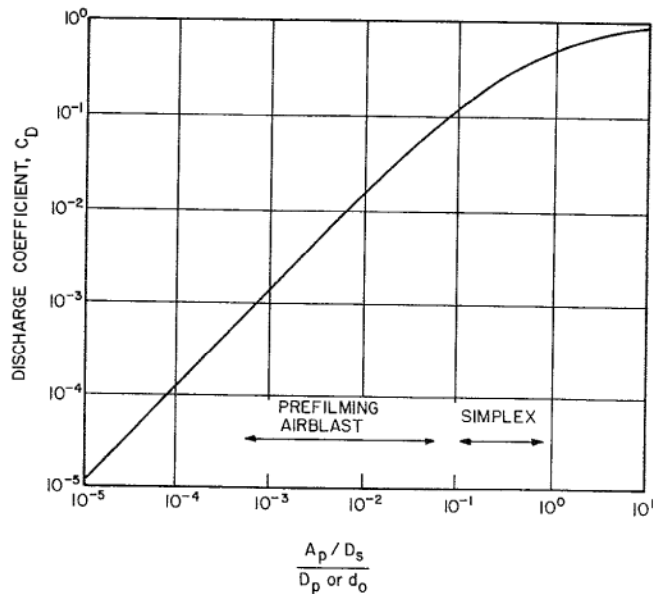


Figure 5.39 Theoretical relationship between discharge coefficient and atomizer dimensions (Simmons [54]).

calculations carried out by Simmons [54] using Eqs. (5.25) and (5.26), due to Giffen and Maraszew [4]. It is clear from this figure that typical values of the discharge coefficient for prefilming airblast nozzles are appreciably lower than those normally associated with pressure-swirl nozzles.

NOMENCLATURE

A_a	air core area, m^2
A_p	total inlet ports area, m^2
A_o	discharge orifice area, m^2
A_s	swirl chamber area, m^2
C	cavitation number
C_D	discharge coefficient
D	disk diameter, m
D_m	diametral distance between centerlines of inlet ports (see [28])
D_p	inlet port diameter, m
D_s	swirl chamber diameter, m
d_o	discharge orifice diameter, m
FN	flow number $[\dot{m}_L/(\Delta P_L \rho_L)^{0.5}]$, m^2
FN_{UK}	flow number, (UK gal/h)/(psid) ^{0.5}
FN_{US}	flow number, (lb/h)/(psid) ^{0.5}
K	atomizer constant ($A_p/d_o D_s$)
K_1	atomizer constant ($A_p/\pi r_o R_s$)
K_v	velocity coefficient, ratio of actual to theoretical velocity
L_s	length of parallel portion of swirl chamber, m
L_p	length of inlet swirl ports, m
l_o	orifice length, m
$\dot{m}_L$	liquid flow rate, kg/s
P	total pressure, Pa
p	static pressure, Pa
ΔP_L	injection pressure differential across nozzle, Pa
R	disk radius, m
R_s	radius of swirl chamber, m
r	local radius, m
r_a	radius of air core, m
r_o	radius of final discharge orifice, m
SG	specific gravity
t	film thickness, m
U	resultant velocity in orifice, m/s
u	axial velocity in orifice, m/s
v	local flow velocity, m/s
v_i	inlet velocity to swirl chamber, m/s
v_r	radial velocity of liquid, m/s

X	A_a/A_o
z	number of teeth
μ_L	liquid dynamic viscosity, kg/m s
ν_L	liquid kinematic viscosity, m ² /s
ρ_L	liquid density, kg/m ³
σ	surface tension, kg/s ²
ω	rotational speed, radian/s
θ	spray cone half angle, degrees

REFERENCES

1. Bird, A. L., Some Characteristics of Nozzles and Sprays for Oil Engines, *Trans. Second World Power Conference*, Berlin, Vol. 8, Section 29, No. 82, 1930, p. 260.
2. Gellales, A. G., Effect of Orifice Length/Diameter Ratio on Fuel Sprays for Compression Ignition Engines, NACA Report No. 402, 1931.
3. Schweitzer, P. H., Mechanism of Disintegration of Liquid Jets, *J. Appl. Phys.*, Vol. 8, 1937, pp. 513–521.
4. Giffen, E., and Muraszew, A., *Atomization of Liquid Fuels*, Chapman & Hall, London, 1953.
5. Bergwerk, W., Flow Pattern in Diesel Nozzle Spray Holes, *Proc. Inst. Mech. Eng.*, Vol. 173, No. 25, 1959, pp. 655–660.
6. Spikes, R. H., and Pennington, G. A., Discharge Coefficient of Small Submerged Orifices, *Proc. Inst. Mech. Eng.*, Vol. 173, No. 25, 1959, pp. 661–665.
7. Lichtarowicz, A., Duggins, R. K., and Markland, E., Discharge Coefficients for Incompressible Non-cavitating Flow Through Long Orifices, *J. Mech. Eng. Sci.*, Vol. 7, No. 2, 1965, pp. 210–219.
8. Arai, M., Shimizu, M., and Hiroyasu, H., Breakup Length and Spray Angle of High Speed Jet, *Proceedings of the 3rd International Conference on Liquid Atomization and Spray Systems (ICLASS)*, London, 1985, pp. IB/4/1–10.
9. Zucrow, M. J., Discharge Characteristics of Submerged Jets, Bull. No. 31, Engineering Experimental Station, Purdue University, West Lafayette, Ind., 1928.
10. Nakayama, Y., Action of the Fluid in the Air Micrometer: First Report, Characteristics of Small Diameter Nozzle and Orifice, *Bull. Jpn. Soc. Mech. Eng.*, Vol. 4, 1961, pp. 516–524.
11. Asihmin, V. I., Geller, Z. I., and Skobel'cyn, Yu. A., Discharge of a Real Fluid from Cylindrical Orifices (in Russian), *Oil Ind.*, Vol. 9, Moscow, 1961.
12. Ruiz, F., and Chigier, N., The Mechanics of High Speed Cavitation, *Proceedings of the 3rd International Conference on Liquid Atomization and Spray Systems*, London, 1985, pp. VIB/3/1–15.
13. Varde, K. S., and Popa, D. M., Diesel Fuel Spray Penetration at High Injection Pressure, SAE Paper 830448, 1984.
14. Watson, E. A., unpublished report, Joseph Lucas Ltd., London, 1947.
15. Taylor, G. I., The Mechanics of Swirl Atomizers, *Seventh International Congress of Applied Mechanics*, Vol. 2, Pt. 1, 1948, pp. 280–285.
16. Taylor, G. I., The Boundary Layer in the Converging Nozzle of a Swirl Atomizer, *Q. J. Mech. Appl. Math.*, Vol. 3, Pt. 2, 1950, pp. 129–139.
17. Binnie, A. M., and Harris, D. P., The Application of Boundary Layer Theory to Swirling Liquid Flow Through a Nozzle, *Q. J. Mech. Appl. Math.*, Vol. 3, Pt. 1, 1950, pp. 80–106.
18. Hodgekinson, T. G., Porton Technical Report No. 191, 1950.
19. Dombrowski, N., and Hassan, D., The Flow Characteristics of Swirl Centrifugal Spray Pressure Nozzles with Low Viscosity Liquids, *AIChE J.*, Vol. 15, 1969, p. 604.
20. Jones, A. R., Design Optimization of a Large Pressure-Jet Atomizer for Power Plant, *Proceedings of the 2nd International Conference on Liquid Atomization and Spray Systems*, Madison, Wis., 1982, pp. 181–185.

21. Radcliffe, A., The Performance of a Type of Swirl Atomizer, *Proc. Inst. Mech. Eng.*, Vol. 169, 1955, pp. 93–106.
22. Carlisle, D. R., Communication on The Performance of a Type of Swirl Atomizer, by A. Radcliffe, *Proc. Inst. Mech. Eng.*, Vol. 169, 1955, p. 101.
23. Eisenklam, P., Atomization of Liquid Fuels for Combustion, *J. Inst. Fuel*, Vol. 34, 1961, pp. 130–143.
24. Rizk, N. K., and Lefebvre, A. H., Internal Flow Characteristics of Simplex Swirl Atomizers, *AIAA J. Propul. Power*, Vol. 1, No. 3, 1985, pp. 193–199.
25. Tipler, W., and Wilson, A. W., Combustion in Gas Turbines, Paper B9, *Proceedings of the Congress International des Machines a Combustion (CIMAC)*, Paris, 1959, pp. 897–927.
26. Elkotb, M. M., Rafat, N. M., and Hanna, M. A., The Influence of Swirl Atomizer Geometry on the Atomization Performance, *Proceedings of the 1st International Conference on Liquid Atomization and Spray Systems*, Tokyo, 1978, pp. 109–115.
27. Joyce, J. R., Report ICT15, Shell Research Ltd., London, 1947.
28. Babu, K. R., Narasimhan, M. V., and Narayanaswamy, K., Correlations for Prediction of Discharge Rate, Cone Angle, and Air Core Diameter of Swirl Spray Atomizers, *Proceedings of the 2nd International Conference on Liquid Atomization and Spray Systems*, Madison, Wis., 1982, pp. 91–97.
29. Lefebvre, A. H., *Gas Turbine Combustion*, Hemisphere, Washington, D.C., 1983.
30. Simmons, H. C., and Harding, C. F., Some Effects of Using Water as a Test Fluid in Fuel Nozzle Spray Analysis, ASME Paper 80-GT-90, presented at ASME Gas Turbine Conference, New Orleans, 1980.
31. Sankaran Kutty, P., Narasimhan, M. V., and Narayanaswamy, K., Design and Prediction of Discharge Rate, Cone Angle, and Air Core Diameter of Swirl Chamber Atomizers, *Proceedings of the 1st International Conference on Liquid Atomization and Spray Systems*, Tokyo, 1978, pp. 93–100.
32. Narasimhan, M. V., Sankaran Kutty, P., and Narayanaswamy, K., Prediction of the Air Core Diameter in Swirl Chamber Atomizers, unpublished report, 1978.
33. Jasuja, A. K., Atomization of Crude and Residual Fuel Oils, *ASME J. Eng. Power*, Vol. 101, No. 2, 1979, pp. 250–258.
34. Rizk, N. K., and Lefebvre, A. H., Influence of Liquid Film Thickness on Airblast Atomization, *ASME J. Eng. Power*, Vol. 102, No. 3, 1980, pp. 706–710.
35. Simmons, H. C., The Prediction of Sauter Mean Diameter for Gas Turbine Fuel Nozzles of Different Types, *ASME J. Eng. Power*, Vol. 102, No. 3, 1980, pp. 646–652.
36. Rizk, N. K., and Lefebvre, A. H., Spray Characteristics of Simplex Swirl Atomizers, *Prog. Aeronaut. Astronaut.*, Vol. 95, 1985, pp. 563–580.
37. Suyari, M., and Lefebvre, A. H., Film Thickness Measurements in a Simplex Swirl Atomizer, *AIAA J. Propul. Power*, Vol. 2, No. 6, 1986, pp. 528–533.
38. Sankaran Kutty, P., Narasimhan, M. V., and Narayanaswamy, K., Prediction of the Coefficient of Discharge of Swirl Chamber Atomizers, unpublished work, 1978.
39. Rizk, N. K., and Lefebvre, A. H., Prediction of Velocity Coefficient and Spray Cone Angle for Simplex Swirl Atomizers, *Proceedings of the 3rd International Conference on Liquid Atomization and Spray Systems*, London, 1985, pp. III C/2/1–16.
40. Rizk, N. K., and Lefebvre, A. H., Influence of Liquid Properties on the Internal Flow Characteristics of Simplex Swirl Atomizers, *Atomization Spray Technol.*, Vol. 2, No. 3, 1986, pp. 219–233.
41. Hinze, J. O., and Milborn, H., Atomization of Liquids by Means of a Rotating Cup, *J. Appl. Mech.*, Vol. 17, No. 2, 1950, pp. 145–153.
42. Adler, C. R., and Marshall, W. R., Performance of Spinning Disk Atomizers, *Chem. Eng. Prog.*, Vol. 47, 1951, pp. 515–601.
43. Fraser, R. P., Dombrowski, N., and Routley, J. H., The Production of Uniform Liquid Sheets from Spinning Cups; The Filming by Spinning Cups; The Atomization of a Liquid Sheet by an Impinging Air Stream, *Chem. Eng. Sci.*, Vol. 18, 1963, pp. 315–321, 323–337, 339–353.
44. Walton, W. H., and Prewitt, W. C., The Production of Sprays and Mists of Uniform Drop Size

- by Means of Spinning Disc Type Sprayers, *Proc. Phys. Soc.*, Vol. 62, Pt. 6, June 1949, pp. 341–350.
45. Christensen, L. S., and Steely, S. L., Monodisperse Atomizers for Agricultural Aviation Applications, NASA CR-159777, February 1980.
 46. Tanasawa, Y., Miyasaka, Y., and Umehara, M., Effect of Shape of Rotating Disks and Cups on Liquid Atomization, *Proceedings of the 1st International Conference on Liquid Atomization and Spray Systems*, Tokyo, 1978, pp. 165–172.
 47. Emslie, A. G., Benner, F. T., and Peck, L. G., *J. Appl. Phys.*, Vol. 29, 1958, p. 858.
 48. Oyama, Y., and Endou, K., On the Centrifugal Disk Atomization and Studies on the Atomization of Water Droplets, *Kagaku Kogaku*, Vol. 17, 1953, pp. 256–260, 269–275 (in Japanese, English summary).
 49. Matsumoto, S., Belcher, D. W., and Crosby, E. J., Rotary Atomizers: Performance Understanding and Prediction, *Proceedings of the 3rd International Conference on Liquid Atomization and Spray Systems*, London, 1985, pp. IA/1/1–21.
 50. Nikolaev, V. S., Vachagin, K. D., and Baryshev, Yu. N., *Int. Chem. Eng.*, Vol. 7, 1967, p. 595.
 51. Bruin, S., Velocity Distributions in a Liquid Film Flowing Over a Rotating Conical Surface, *Chem. Eng. Sci.*, Vol. 24, No. 11, 1969, p. 1647–1654.
 52. Matsumoto, S., Saito, K., and Takashima, Y., *Bull Tokyo Inst. Technol.*, Vol. 116, 1973, p. 85.
 53. Lefebvre, A. H., and Miller, D., The Development of an Air Blast Atomizer for Gas Turbine Application, CoA-Report-AERO-193, College of Aeronautics, Cranfield, Bedford, England, June 1966.
 54. Simmons, H. C., Parker Hannifin Report, BTA 136, 1981.
 55. Eisenklam, P., On Ligament Formation from Spinning Disks and Cups, *Chem. Eng. Sci.*, Vol. 19, 1964, pp. 693–694.

ATOMIZER PERFORMANCE

INTRODUCTION

The spray properties of importance include mean drop size, drop size distribution, radial and circumferential patternation, droplet number density, cone angle, and penetration. The *quality* or *fineness* of an atomization process is usually described in terms of a mean drop size. Various definitions of mean drop size are available (see Chapter 3), of which the most widely used is the Sauter mean diameter, which represents the volume/surface ratio of the spray. This definition of mean drop size has special significance for heat and mass transfer applications, such as spray drying and the combustion of liquid fuel sprays.

Unfortunately, the physical processes involved in atomization are not yet sufficiently well understood for mean diameters to be expressed in terms of equations derived from basic principles. The simplest case of the breakup of a liquid jet has been studied theoretically for more than a hundred years, but the results of these studies have failed to predict the spray characteristics to a satisfactory level of accuracy. The situation in regard to the complex sprays produced by more sophisticated types of atomizers is, understandably, even worse. As the physical structure and dynamics of a spray are the result of many interwoven complex mechanisms, none of which are fully understood, it is hardly surprising that mathematical treatments of atomization have so far defied successful development. In consequence, the majority of investigations into the drop size distributions produced in atomization have, of necessity, been empirical in nature. Nevertheless, they have yielded a considerable body of useful information from which a number of general conclusions on the effects of liquid properties, gas properties, and injector dimensions on mean drop size can be drawn.

The properties of a liquid most relevant to atomization are surface tension, viscosity, and density. For a liquid injected into a gaseous medium, the only thermodynamic property generally considered of importance is the gas density. The liquid flow variables of importance are the velocity of the liquid jet or sheet and the turbulence in the liquid stream. The gas flow variables to be considered are the absolute velocity and the relative gas-to-liquid velocity. The importance of this parameter is unfortunate, since the actual velocity field in the atomization region cannot be determined accurately. Even when liquid is sprayed into so-called quiescent air, the air velocity adjacent to the spray can attain quite high values due to the momentum transfer from the liquid to the surrounding air [1]. The turbulence characteristics of the air or gas may also influence atomization, but no systematic study of this effect has yet been undertaken.

For plain-orifice injectors, the key geometric variables are the orifice length and diameter. Final orifice diameter is also of prime importance for pressure-swirl atomizers. For prefilming-type airblast atomizers, the dimensions that have most influence on mean drop size are the prefilmer diameter and the hydraulic mean diameter of the atomizer air duct at the exit plane. For rotary atomizers, the only dimension of significance is the diameter of the rotating disk or cup. For all types of pneumatic or twin-fluid injectors, another variable affecting atomization is the mass ratio of liquid to air, or steam.

The absence of any general theoretical treatment of the atomization process has led to the evolution of empirical equations to express the relationship between the mean drop size in a spray and the variables of liquid properties, gas properties, flow conditions, and atomizer dimensions. Many of the drop size equations published in the literature are of somewhat dubious value. In some cases they are based on data obtained using experimental techniques that are no longer considered valid. With other equations, the insertion of appropriate values of gas and liquid properties yields improbable values of SMD. The equations selected for inclusion in this chapter are considered to be the best available for engineering calculations of mean drop sizes for the types of atomizers described in Chapter 4.

PLAIN-ORIFICE ATOMIZER

In plain-orifice atomizers, disintegration of the jet into drops is promoted by an increase in flow velocity, which increases both the level of turbulence in the issuing jet and the aerodynamic drag forces exerted by the surrounding medium; it is opposed by an increase in liquid viscosity, which delays the onset of atomization by resisting breakup of the ligaments. Merrington and Richardson's [2] experiments on sprays injected from a plain circular orifice into stagnant air yielded the following relationship for mean drop size.

$$\text{SMD} = \frac{500d_o^{1.2}v_L^{0.2}}{U_L} \quad (6.1)$$

Most of the research carried out on plain-orifice atomizers has been directed toward the types of injectors employed in compression ignition (diesel) engines. With these injectors, jet breakup is due mainly to aerodynamic interaction with a highly turbulent jet. In an early study, Panasenkov [3] examined the influence of turbulence on the breakup of a liquid jet and determined mean drop sizes for jet Reynolds numbers ranging from 1000 to 12,000. Drop sizes were correlated in terms of discharge orifice diameter and liquid Reynolds number as

$$\text{MMD} = 6d_o \text{Re}_L^{-0.15} \quad (6.2)$$

Harmon's [4] equation for SMD takes account of ambient gas properties as well as liquid properties. We have

$$\text{SMD} = 3330 d_o^{0.3} \mu_L^{0.07} \rho_L^{-0.648} \sigma^{-0.15} U_L^{-0.55} \mu_G^{0.78} \rho_G^{-0.052} \quad (6.3)$$

An unusual feature of this equation is that an increase in surface tension is predicted to give finer atomization.

From their study on the effects of ambient gas density on drop size distributions, Giffen and Lamb [5] drew the following conclusions:

1. The fineness and uniformity of a spray are improved by an increase in gas density, but the rate of improvement decreases at high gas densities.
2. Gas density appears to exert little effect on the minimum drop size in the spray.
3. The maximum drop size is reduced at higher gas densities according to the relationship $D_{\max} \propto \rho_G^{-0.2}$.
4. The improvement in atomization with increase in gas density is due mainly to a reduction in the number of large drops and is probably caused by subdivision of these big drops.

In 1955 Miesse [6] published his analysis of the results of previous theoretical and experimental studies on the disintegration of liquid jets. His equation for the "best-fit" correlation of the available experimental data is

$$D_{0.999} = d_o \text{We}_L^{-0.333} (23.5 + 0.000395 \text{Re}_L) \quad (6.4)$$

At about the same time, Tanasawa and Toyoda [7] proposed the following equation:

$$\text{SMD} = 47 d_o U_L^{-1} \left(\frac{\sigma}{\rho_G} \right)^{0.25} \left[1 + 331 \frac{\mu_L}{(\rho_L \sigma d_o)^{0.5}} \right] \quad (6.5)$$

More recent equations for the mean drop sizes produced by the diesel-type injectors are that of Hiroyasu and Katoda [8], which is of the form

$$\text{SMD} = 2330 \rho_A^{0.121} Q^{0.131} \Delta P_L^{-0.135} \quad (6.6)$$

and that of Elkotb [9]

$$\text{SMD} = 3.08 \nu_L^{0.385} (\sigma \rho_L)^{0.737} \rho_A^{0.06} \Delta P_L^{-0.54} \quad (6.7)$$

The above equations for plain-orifice atomizers, as listed in Table 6.1, apply strictly to the injection of liquids into quiescent air. Two other cases of practical importance are (1) injection into a coflowing or contraflowing stream of air and (2) transverse injection across a flowing stream of air.

When a plain-orifice atomizer is oriented normal to the airflow, the larger drops penetrate farther into the airstream and the spectrum of drop sizes produced in the flowing airstream is skewed radially, as illustrated in Fig. 6.1. This distortion of the spray is not necessarily disadvantageous; for example, the ignition of a turbojet combustor is facilitated by locating the igniter in a region of small drops.

PRESSURE-SWIRL ATOMIZERS

Because of its wide range of applications, the pressure-swirl atomizer has attracted the attention of many research workers and has been the subject of considerable theoretical and experimental studies. However, despite these efforts, our knowledge of pressure-swirl atomization is still unsatisfactory. The physics is not well understood, the available data and correlations are of questionable validity, and there is little agreement between the various investigators as to the exact relationships between liquid properties, nozzle dimensions, and mean drop size. This unsatisfactory situation has several causes: the great complexity of the atomization process; differences in the design, size, and operating conditions of the nozzles tested; and the inaccuracies and limitations associated with drop size measurement techniques.

Most of the reported studies have concentrated on small-scale pressure-swirl atomizers of the type used in aircraft gas turbines, the only exception of note being the comprehensive experimental study carried out by Jones [10] on pressure-swirl nozzles of the large scale used in electrical power generation. In these var-

Table 6.1 Drop size equations for plain-orifice atomizers

Investigators	Equation
Merrington and Richardson [2]	$\text{SMD} = \frac{500d_o^{1.2}v_o^{0.2}}{U_L}$
Panasenkov [3]	$\text{MMD} = 6d_o\text{Re}_L^{-0.15}$
Harmon [4]	$\text{SMD} = 3330d_o^{0.3}\mu_L^{0.07}\rho_L^{-0.648}\sigma^{-0.15}U_L^{-0.15}\mu_G^{0.78}\rho_G^{-0.052}$
Miesse [6]	$D_{0.999} = d_o\text{We}_L^{-0.333}(23.5 + 0.000395\text{Re}_L)$
Tanasawa and Toyoda [7]	$\text{SMD} = 47d_oU_L^{-1}\left(\frac{\sigma}{\rho_G}\right)^{0.25}\left[1 + 331\frac{\mu_L}{(\rho_L\sigma d_o)^{0.5}}\right]$
Hiroyasu and Katoda [8]	$\text{SMD} = 2330\rho_A^{0.121}\mathcal{Q}^{0.131}\Delta P^{-0.135}$
Elkobt [9]	$\text{SMD} = 3.08v_L^{0.385}(\sigma\rho_L)^{0.737}\rho_A^{0.06}\Delta P_L^{-0.54}$

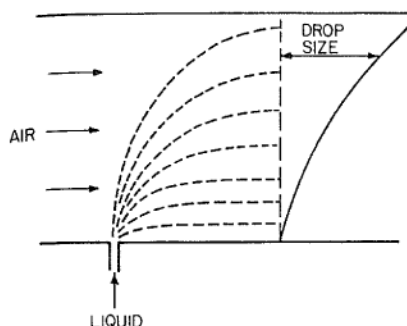


Figure 6.1 Influence of cross-stream airflow on drop size distribution.

ious studies emphasis has been placed on the variation of drop size with liquid properties and operating conditions.

Effect of Variables on Mean Drop Size

The main factors governing the atomization quality of pressure-swirl atomizers are liquid properties, the physical properties of the air or gas into which the liquid is injected, the liquid injection pressure, and the size of the atomizer as indicated by its flow number.

Liquid properties. The liquid properties of importance are surface tension and viscosity. In practice, the significance of surface tension is diminished by the fact that most commercial fuels exhibit only slight differences in this property. This is also true for density. However, viscosity varies by almost two orders of magnitude in some applications, so its effect on atomization quality can be quite large.

Surface tension. Simmons and Harding [11] studied the difference in atomizing performance between water and kerosine for six simplex fuel nozzles. These two liquids have virtually the same viscosity and a 30% difference in density, but the surface tension of water is higher by a factor of 3. Thus, according to Simmons and Harding, any significant differences in SMD must be due to differences in surface tension rather than density. For the important practical case of Weber number less than unity, it was found that $SMD \propto \sigma^a$, where a has a value of 0.19. This result is not supported by the findings of Kennedy [12], who used simplex nozzles featuring much higher flow numbers and observed a far stronger dependence of SMD on surface tension, namely $SMD \propto \sigma$. However, Jones's [10] studies on nozzles of high flow number led to a value for a of 0.25. This is fairly close to Simmons's result and is identical to the value derived by Lefebvre [13] from dimensional analysis of published data on SMD.

Wang and Lefebvre [14] used two simplex nozzles of different flow numbers to examine the influence of surface tension on SMD. They compared the results

obtained with water and diesel oil and made a small adjustment to the diesel data, using Eq. (6.18) to compensate for the diesel oil's higher viscosity. Thus in Figs. 6.2 and 6.3 the higher SMD values exhibited by water, in comparison with diesel oil, are due entirely to the higher surface tension of water.

Analysis of the data contained in Figs. 6.2 and 6.3 indicates a value for a of around 0.25. Thus, if Kennedy's [12] exceptionally high value of a is neglected, the evidence acquired in the most recent experimental studies suggests a "best" value for a of around 0.25.

Viscosity. Published data on the effect of viscosity on mean drop size are usually expressed in the form

$$\text{SMD} \propto \mu_L^b \quad (6.8)$$

Values for b of 0.16, 0.20, 0.118, 0.16, 0.06, and 0.215 have been reported by Jasuja [15], Radcliffe [16], Dodge and Biaglow [17], Jones [10], Simmons [18], and Knight [19], respectively. Wang and Lefebvre [14] blended diesel oil (DF-2) with polybutene in different concentrations to achieve a wide range of viscosities. Some of their results, showing the adverse effect of increase in liquid viscosity on atomization quality, are presented in Figs. 6.4 and 6.5. Of special interest in these figures is that viscosity appears to have less effect on SMD for nozzles of lower spray cone angle.

The influence of liquid viscosity on SMD is shown more directly in Fig. 6.6, which illustrates the adverse effects of increases in viscosity and flow number on mean drop size. This figure also indicates that the dependence of SMD on viscosity diminishes with increase in flow number.

Liquid flow rate. All the reported data show that mean drop sizes increase at higher flow rates [10, 15, 16, 20]. Some of the results obtained by Wang and

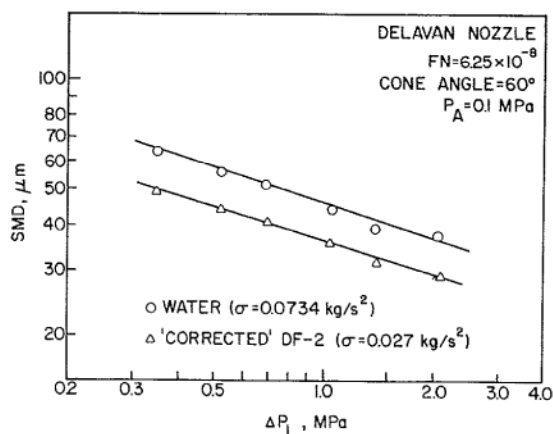


Figure 6.2 Influence of surface tension on mean drop size (Wang and Lefebvre [14]).

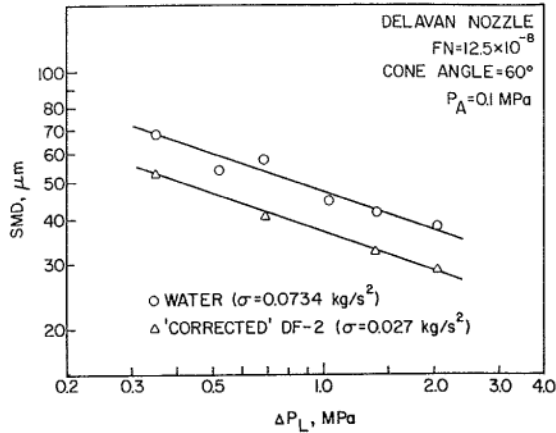


Figure 6.3 Influence of surface tension on mean drop size (Wang and Lefebvre [14]).

Lefebvre [14] for water are shown in Figs. 6.7 and 6.8 for 90° and 60° spray cone angles, respectively. By using three similar atomizers with different flow numbers, it was possible to study the influence of flow rate on mean drop size while maintaining either flow number or nozzle pressure differential constant. For any given flow number, both figures show that mean drop size declines with increase in flow rate. This is due to the increase in injection pressure differential that accompanies an increase in flow rate. If the pressure differential is kept constant, the flow rate can be raised only by increasing the flow number, i.e., by switching to a larger nozzle. This impairs atomization quality, especially at low injection pressures, as shown in Figs 6.7 and 6.8.

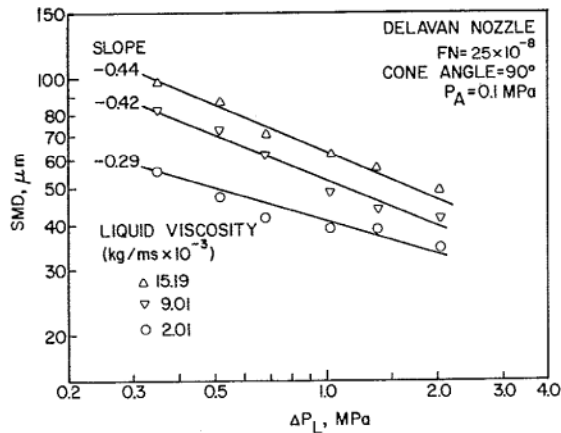


Figure 6.4 Influence of viscosity on SMD for a cone angle of 90° (Wang and Lefebvre [14]).

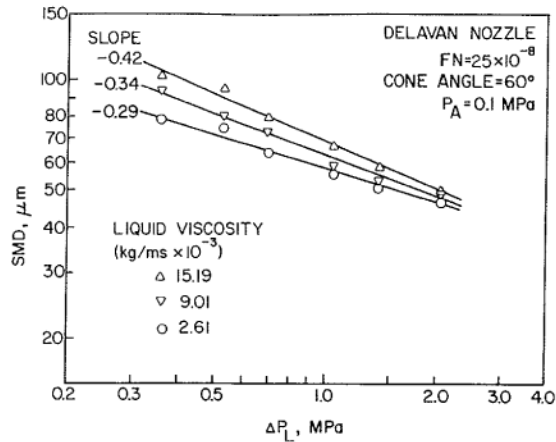


Figure 6.5 Influence of viscosity on SMD for a cone angle of 60° (Wang and Lefebvre [14]).

Flow number. Figures 6.7 and 6.8 also serve to demonstrate the influence of flow number on mean drop size. They show that an increase in flow number produces a coarser spray. However, the influence of flow number is seen to diminish with increase in liquid pressure differential, becoming negligibly small at pressure differentials of around 2 MPa (290 psid). Further evidence on the effect of flow number on mean drop size is provided in Figs. 6.9 and 6.10. Figure 6.9 was obtained using a diesel oil (DF-2), and Fig. 6.10 shows similar results for diesel oil after blending with polybutene to produce a large increase in viscosity. Comparison of Figs. 6.9 and 6.10 reveals that the effect of a reduction in flow

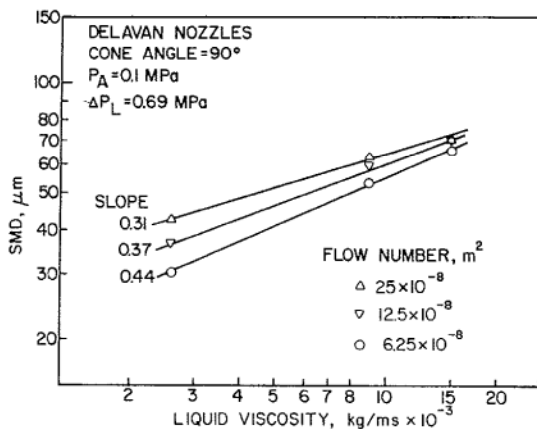


Figure 6.6 Influence of viscosity on SMD for various flow numbers (Wang and Lefebvre [14]).

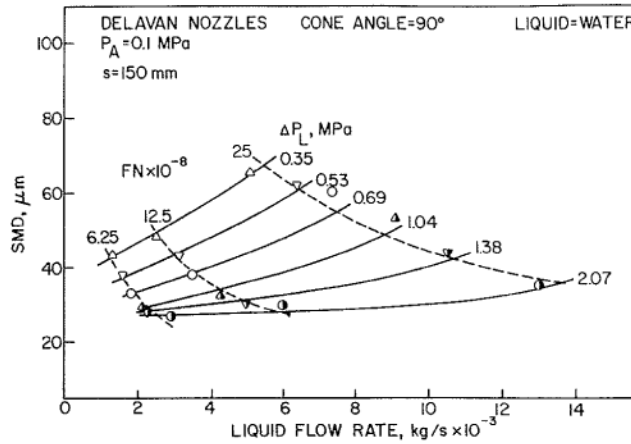


Figure 6.7 Relationship between SMD, flow number, and injection pressure for water; spray cone angle = 90° (Wang and Lefebvre [14]).

number, which is always conducive to better atomization, is much less for the liquid of higher viscosity.

Nozzle pressure differential. Increase in liquid pressure differential causes the liquid to be discharged from the nozzle at a higher velocity, which promotes a finer spray. The beneficial effect of raising the nozzle pressure differential on

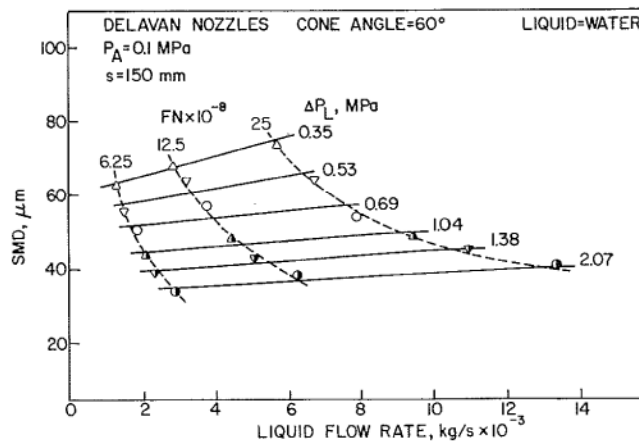


Figure 6.8 Relationship between SMD, flow number, and injection pressure for water; spray cone angle = 60° (Wang and Lefebvre [14]).

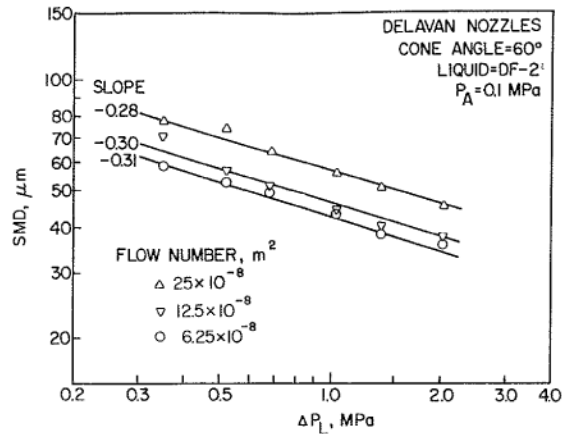


Figure 6.9 Influence of flow number on SMD for low-viscosity liquid (Wang and Lefebvre [14]).

atomization quality is clearly illustrated in Figs. 6.2 to 6.10. The effect may be expressed quantitatively as

$$\text{SMD} \propto \Delta P_L^d \quad (6.9)$$

Figure 6.9 illustrates the effects of liquid pressure differential on mean drop size for DF-2, while Fig. 6.10 shows similar results for a liquid of higher viscosity. These figures, are plotted in logarithmic form to facilitate data analysis. The slopes indicated in these and other figures denote the exponent d in the expression $\text{SMD} \propto \Delta P_L^d$. Figures 6.4, 6.5, 6.9, and 6.10 show values of d ranging

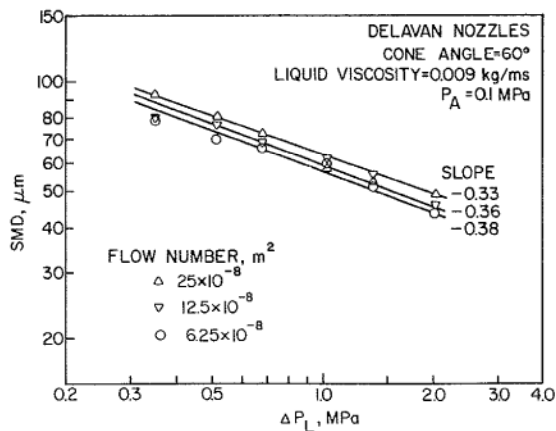


Figure 6.10 Influence of flow number on SMD for high-viscosity liquid (Wang and Lefebvre [14]).

from -0.28 to -0.44 . Other reported values of d are -0.275 , -0.35 , -0.4 , and -0.43 , from Simmons [18], Abou-Ellail et al. [21], Radcliffe [16], and Jasuja [15], respectively.

Air properties. The two properties of interest are air pressure and air temperature. From a fundamental standpoint, they are usually regarded only as components of density. The results of various experimental studies on the influence of air density on SMD have been reviewed by Lefebvre [13] and Dodge and Biaglow [17].

De Corso [22] was among the first to investigate the influence of ambient air pressure on the spray characteristics of simplex swirl atomizers. Using a fuel of similar physical properties to diesel oil (DF-2), he measured an increase in drop size in going from 0.1 MPa (14.5 psia) to 0.79 MPa (114.5 psia). According to De Corso, "a continuing decrease in drop size with increase in ambient pressure would be expected, since the drag force on a drop increases with increasing density. Thus as ambient pressure rises, a decrease would be expected in the critical drop size, i.e., the maximum size that can withstand breakup, as indicated by Hinze [23] and Lane [24]." De Corso attributed his actual observed increase in drop size to increased coalescence of the spray drops as the ambient pressure is increased. Previous work by De Corso and Kemeny [25] had shown a reduction in spray cone angle with increase in ambient pressure, due to the action of induced gas currents, which tend to collapse the spray into a small volume and thereby provide more opportunity for coalescence.

Neya and Sato [26] also examined the influence of ambient air pressure on mean drop size and spray cone angle. Using water as the test fluid, they obtained results similar to those of De Corso and Kemeny [25] in regard to the contraction of spray cone angle with increase in air pressure. Over a pressure range of 0.1 to 0.5 MPa (1 to 5 atm) they observed a marked rise in the SMD with increase in P_A ($SMD \propto P_A^{0.27}$).

Neya and Sato also invoked droplet coalescence to explain this increase of SMD with P_A , but they asserted that under certain conditions an additional factor that should be considered is a change in the atomization process. Based on instantaneous snapshots of the spray, they concluded that the waviness of the initial liquid sheet is intensified with increasing P_A and the sheet breakup length is shortened.

Rizk and Lefebvre [27] also observed a decline in spray quality with increase in P_A . Their explanation for this result was that contraction of the spray angle reduces the volume of air that interacts with the spray. In consequence, the aerodynamic drag forces created by the spray induce a more rapid acceleration of this smaller air volume in the direction of spray motion, thereby reducing the relative velocity between the drops and the air surrounding them. As mean drop size is inversely proportional to this relative velocity, the effect of a reduction of spray angle is to increase the mean drop size.

In a recent study of the influence of air pressure on mean drop size, Wang and Lefebvre [28] flowed liquids of different viscosities through several Delavan

simplex nozzles having various values of spray cone angle and flow number. Some of the results acquired during this investigation are shown in Figs. 6.11 to 6.13.

Figure 6.11 shows the influence of ambient air pressure on mean drop size for three nozzles having 90° cone angles and flow numbers of 6.25×10^{-8} , 12.5×10^{-8} , and $25 \times 10^{-8} \text{ m}^2$. The liquid employed is diesel oil (DF-2), and the data correspond to a liquid injection pressure differential of 0.69 MPa (100 psi). For the nozzle of highest flow number, the SMD rises fairly steeply with increase in P_A up to a maximum value around 0.4 MPa (4 atm), beyond which any further increase in P_A causes the SMD to decline. For the nozzle of lowest flow number, the initial increase of SMD with P_A is also quite steep up to an air pressure of around 0.4 MPa. However, further increase in P_A above this pressure level appears to have little influence on SMD.

Figures 6.12 and 6.13 shows similar data for sprays of higher viscosity and lower cone angle. The general SMD levels are much higher, but in all cases the SMD again rises with P_A , reaching maximum values at around 0.3 MPa. Further increase in P_A above 0.3 MPa causes atomization quality to improve appreciably.

A characteristic feature of Figs. 6.11 to 6.13 is that SMD increases with ambient pressure up to a maximum value and then declines with any further increase in pressure. Possible explanations for the observed initial rise in SMD with increase in P_A include: (1) reduction in spray angle [25, 26, 29], which promotes drop coalescence and also increases the proportion of large drops in the spray sampling beam, and (2) reduction in sheet breakup length, so that drops are formed from a thicker sheet initially and consequently are larger.

If possible errors arising from measurement techniques are neglected, then Figs. 6.11 to 6.13 show that the forces inhibiting atomization are predominant at low pressures and the SMD rises with increase in P_A . With continuing increase

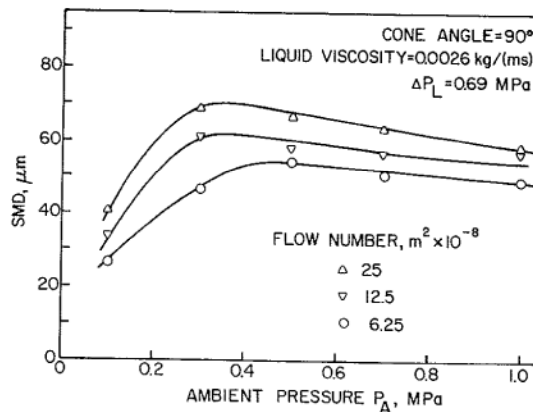


Figure 6.11 Influence of ambient air pressure and nozzle flow number on mean drop size (Wang and Lefebvre [28]).

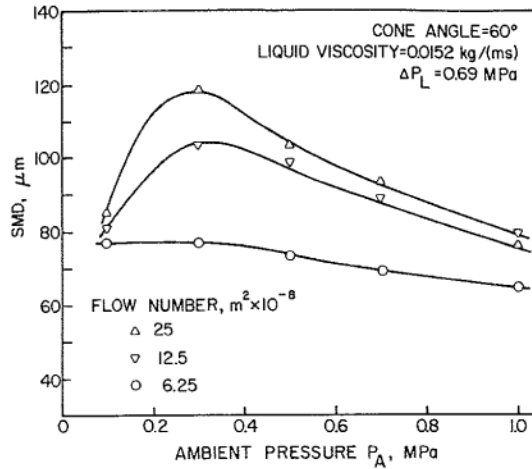


Figure 6.12 Influence of ambient air pressure and nozzle flow number on mean drop size (Wang and Lefebvre [28]).

in P_A , the contraction in spray angle, which is initially quite steep, starts to diminish and eventually becomes zero. This allows the disintegration forces to become dominant, so that any further increase in P_A above the critical value causes the SMD to decline. Thus, when SMD values are plotted against P_A , their characteristic shape is one that shows SMD rising up to a maximum value and then falling with further increase in P_A . During this latter stage the variation of SMD with P_A roughly corresponds to the relationship $\text{SMD} \propto P_A^{-0.25}$.

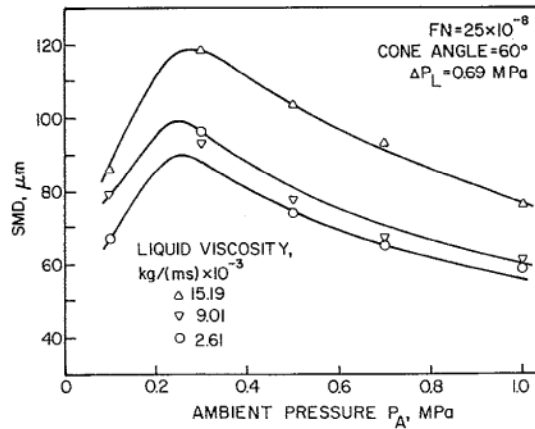


Figure 6.13 Influence of ambient air pressure and liquid viscosity on mean drop size (Wang and Lefebvre [28]).

Abou-Ellail et al. [21] have studied the effect of ambient air temperature on mean drop size. Their results are interesting because they are contrary to what might be expected from purely theoretical considerations. Since an increase in air temperature lowers the Weber number, it would be expected to impede disintegration and give rise to larger drops. However, these workers found that drop sizes diminished with increasing air temperature according to the relationship $SMD \propto T_A^{-0.56}$. Abou-Ellail et al. do not indicate whether the supply lines and spray nozzle were cooled to maintain a constant liquid temperature. As Dodge and Biaglow pointed out, if the liquid temperature is allowed to increase with air temperature, the drop size distribution will be shifted to smaller sizes. The results obtained by Dodge and Biaglow [17] using Jet-A and DF-2 fuels showed no effect of air temperature on SMD.

Atomizer dimensions. It is now generally accepted that the dimension of most importance for atomization is the thickness of the liquid sheet as it leaves the final orifice [13]. Theory [30–32] predicts, and experiment confirms, that mean drop size is roughly proportional to the square root of liquid sheet thickness. Thus, provided the other key parameters that are known to affect atomization are maintained constant, an increase in atomizer size will reduce atomization quality according to the relationship $SMD \propto d_o^{0.5}$. Although this relationship is generally supported by the results of dimensional analysis, the influence of atomizer size per se has not yet been subjected to systematic investigation. However, nozzle size is generally closely related to nozzle flow number, and it is well established that increase in flow number produces coarser atomization, as illustrated in Figs. 6.7 to 6.10.

Length/diameter ratio of swirl chamber. The effect of this ratio was studied by Elkotb et al. [33] as part of a wide investigation on the effects of swirl atomizer geometry on spray drop size. Results on the effect of L_s/D_s on drop size distribution are shown in Fig. 6.14. They show that atomization quality improves initially with an increase in the L_s/D_s ratio, owing to the contribution of extra length toward eliminating the flow striations caused by the finite number of swirl ports. However, once these flow striations have been damped out, the effect of additional length is to raise the energy losses and thereby impair atomization. Thus, one would expect an optimal L_s/D_s ratio, and the results show this optimal value to be around 2.75 [33].

Jones [10] examined the effect of L_s/D_s ratio on MMD over a range of values from 0.31 to 1.26. For the large-scale nozzles employed in this study, he found that $MMD \propto (L_s/D_s)^{0.07}$.

Length/diameter ratio of final orifice. The results obtained by Elkotb et al. on the effect of the l_o/d_o ratio on spray drop size are summarized in Fig. 6.15. Drop size distributions were measured for various values of l_o/d_o while the other parameters were maintained constant at the baseline values. It was found that the SMD diminished continuously over the range of l_o/d_o from 2.82 to 0.4. An op-

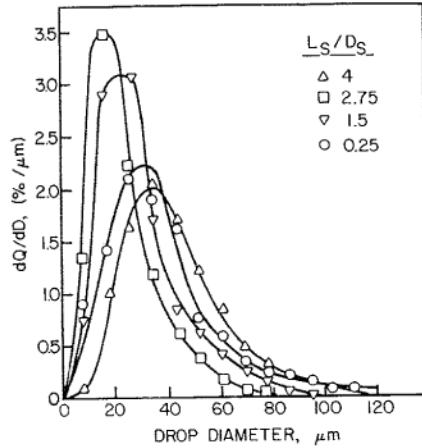


Figure 6.14 Effect of swirl chamber length/diameter ratio on drop size distribution (Elkott et al. [33]).

timal value of l_o/d_o was not encountered, so presumably further improvement in atomization quality could be gained by further reduction in l_o/d_o below 0.4. This conclusion is supported by the results of Jones [10], which showed, over a range of l_o/d_o ratios from 0.1 to 0.9, that $MMD \propto (l_o/d_o)^{0.03}$.

Drop Size Relationships

Owing to the complexity of the various physical phenomena involved in pressure-swirl nozzles, the study of atomization has been pursued principally by empirical methods, yielding correlations for mean drop size of the form

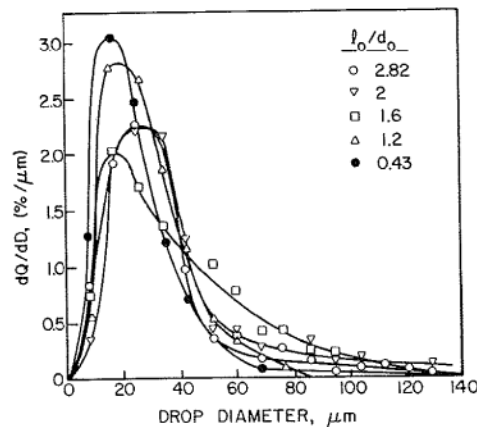


Figure 6.15 Effect of final orifice length/diameter ratio on drop size (Elkott et al. [33]).

$$\text{SMD} \propto \sigma^a \nu^b \dot{m}_L^c \Delta P_L^d \quad (6.10)$$

One of the earliest and most widely quoted expressions is that of Radcliffe [16]:

$$\text{SMD} = 7.3 \sigma^{0.6} \nu_L^{0.2} \dot{m}_L^{0.25} \Delta P_L^{-0.4} \quad (6.11)$$

This equation was derived from analysis of experimental data obtained by Needham [34], Joyce [35], and Turner and Moulton [36]. Subsequent work by Jasuja [15] yielded the expression

$$\text{SMD} = 4.4 \sigma^{0.6} \nu_L^{0.16} \dot{m}_L^{0.22} \Delta P_L^{-0.43} \quad (6.12)$$

However, the variation of surface tension in these experiments was very small and was accompanied by wide variations in viscosity. Thus the exponent 0.6 has no special significance in Eqs. (6.11) and (6.12). Babu et al. [37] used regression analysis to determine the following equations for kerosine-type fuels.

For ΔP_L less than 2.8 MPa (400 psi)

$$\text{SMD} = 133 \frac{FN^{0.64291}}{\Delta P_L^{0.22565} \rho_L^{0.3215}} \quad (6.13)$$

For ΔP_L above 2.8 MPa (400 psi)

$$\text{SMD} = 607 \frac{FN^{0.75344}}{\Delta P_L^{0.19936} \rho_L^{0.3767}} \quad (6.14)$$

The high degree of precision implied in these equations should be regarded with caution in view of the well-known uncertainties in all methods of drop size measurement.

From a series of tests conducted on 25 different fuels, using six different simplex nozzles of large flow number, Kennedy [12] derived the following correlating parameter for nozzles operating at Weber numbers larger than 10:

$$\begin{aligned} \text{SMD} = & 10^{-3} \sigma_L (6.11 + 0.32 \times 10^5 FN \sqrt{\rho_L}) \\ & - 6.973 \times 10^{-3} \sqrt{\Delta P_L} + 1.89 \times 10^{-6} \Delta P_L \end{aligned} \quad (6.15)$$

In estimating Weber number, Kennedy [12] used the film thickness in the final orifice, as given by Simmons's equation (5.39), as the characteristic dimension.

Equation (6.15) implies a very strong dependence of mean drop size on surface tension, while viscosity appears to have no effect at all. Kennedy attributes this, and other differences between his results and those of other workers, to the larger Weber numbers resulting from his use of nozzles of exceptionally high flow number. According to Kennedy, "for Weber numbers greater than 10, a different atomization process occurs, i.e., shear-type breakup, which results in much finer atomization than predicted by previously-reported correlations." However, Jones [10], using large industrial nozzles of much higher flow numbers than those employed by Kennedy, found the effects of surface tension and viscosity on mean

drop size to be fully consistent with all previous observations on small-scale nozzles.

Jones [10] used a high-speed photographic technique to investigate the effects of changing liquid properties, operational variables, and geometric parameters on the drop sizes produced by large pressure-swirl atomizers. Variations in atomizer geometry were achieved by using a series of specially designed nozzle pieces that enabled 50 different geometric configurations to be studied. Analysis of the experimental data yielded the following equation for mean drop size:

$$\begin{aligned} \text{MMD} = & 2.47 m_L^{0.315} \Delta P_L^{-0.47} \mu_L^{0.16} \mu_A^{-0.04} \sigma^{0.25} \rho_L^{-0.22} \\ & \times \left(\frac{l_o}{d_o} \right)^{0.03} \left(\frac{L_s}{D_s} \right)^{0.07} \left(\frac{A_p}{D_s d_o} \right)^{-0.13} \left(\frac{D_s}{d_o} \right)^{0.21} \end{aligned} \quad (6.16)$$

The ranges of variables covered in this investigation are listed in Table 5.1.

Lefebvre's [13] analysis of the flow processes in the final orifice of a simplex nozzle led to the following equation for SMD:

$$\text{SMD} = A \sigma^{0.25} \mu_L^{0.25} \rho_L^{0.125} d_o^{0.5} \rho_A^{-0.25} \Delta P_L^{-0.375} \quad (6.17)$$

The value of 0.25 for the surface tension exponent is reasonably close to Simmons's estimates (0.16 to 0.19) and is in exact agreement with the experimental findings of Jones [10] and Wang and Lefebvre [14]. The exponent of 0.25 for viscosity agrees fairly well with all reported values. No reliable experimental data exist to test the exponent for liquid density, but all the evidence suggests that the effect of this property on SMD is quite small. The d_o exponent of 0.5 confirms the theoretical value. The exponents for injection pressure differential and air density of -0.375 and -0.25 , respectively, are identical to the experimental values obtained in the investigation by Abou-Ellail et al. [21]. Substituting $d_o \propto \dot{m}_L^{0.5} / (\Delta P_L / \rho_L)^{0.25}$ into Eq. (6.17) and using Jasuja's [15] data to determine the value of A gives

$$\text{SMD} = 2.25 \sigma^{0.25} \mu_L^{0.25} \dot{m}_L^{0.25} \Delta P_L^{-0.5} \rho_A^{-0.25} \quad (6.18)$$

Recent developments. It is clearly unreasonable to expect any single-term parameter for SMD of the types described above, all of which are essentially derivatives or modifications of Eq. (6.10), to describe adequately the complex fluid dynamic processes involved in atomization. Further objections to this type of formula can be made on purely practical grounds. For example, it takes no account of spray cone angle, which is known to affect mean drop size. Moreover, it implies that a nonviscous liquid should yield a spray of infinitely small drops, which seems highly unlikely from both theoretical and practical considerations.

In an attempt to overcome some of the deficiencies of existing SMD formulas and to explain some of the apparent anomalies that careful measurements often reveal, Lefebvre [38] has proposed an alternative form of equation for the mean drop sizes produced by pressure-swirl atomizers. This equation is not the result of a mathematical treatment of the subject but is based on considerations of the

basic mechanisms involved in pressure-swirl atomization. It starts from the notion that the disintegration of a liquid jet or sheet issuing from a nozzle is not caused solely by aerodynamic forces but must be the result, at least in part, of turbulence or other disruptive forces within the liquid itself. These disturbances within the flow have a strong influence on sheet disintegration, especially in the first stage of atomization. Subsequently, and to some extent simultaneously, the relative velocity between the liquid and the surrounding gas plays an important role in atomization through its influence on the development of waves on the initially smooth surface and the production of unstable ligaments. Any increase in this relative velocity causes a reduction in the size of the ligaments, so that when they disintegrate much smaller drops are produced.

As the process of atomization in pressure-swirl nozzles is highly complex, it is convenient to subdivide it into two main stages. The first stage represents the generation of surface instabilities due to the combined effects of hydrodynamic and aerodynamic forces. The second stage is the conversion of surface protuberances into ligaments and then drops. It is recognized that this subdivision of the total atomization process into two separate and distinct stages represents an oversimplification of the mechanisms involved. However, it allows the formulation of an equation for SMD as

$$\text{SMD} = \text{SMD}_1 + \text{SMD}_2 \quad (6.19)$$

Here, SMD_1 represents the first stage of the atomization process. Its magnitude depends partly on the Reynolds number, which provides a measure of the disruptive forces present within the liquid sheet. These forces are enhanced by increases in liquid velocity and sheet thickness and are diminished by an increase in liquid viscosity. SMD_1 is also influenced by the Weber number, which governs the development of capillary waves (ripples) on the liquid surface. The rate of growth of these perturbations into projections large enough to break off and form ligaments is dependent on the ratio of the aerodynamic forces at the liquid/air interface to the consolidating surface tension forces in the liquid, i.e., on Weber number.

It appears, therefore, that a suitable form of SMD_1 might be one that combines both Reynolds and Weber numbers into a single parameter. Analysis of the experimental data leads to the following expression for SMD_1 :

$$\frac{\text{SMD}_1}{t_s} \propto (\text{Re} \times \sqrt{\text{We}})^{-x} \quad (6.20)$$

where Re is $\rho_L U_L t_s / \mu_L$, We is $\rho_A U_R^2 t_s / \sigma$, and t_s is the initial sheet thickness at the nozzle exit.

SMD_1 should not be confused with the Z or Ohnesorge number [39], which is denoted as $\sqrt{\text{We}}/\text{Re}$ and represents the *ratio* of surface tension to viscous forces operating on the liquid. In contrast, SMD_1 is intended to represent the manner in which surface tension and viscous forces *act together* in opposing the disruptive actions of the hydrodynamic and aerodynamic momentum forces [38].

For both pressure and airblast nozzles the relative velocity between the liquid

and the surrounding gas has a profound effect on atomization. It generates the projections on the liquid surface that are a prerequisite to atomization and also furnishes the energy needed to convert these projections into ligaments and then drops. However, another important factor in atomization, as discussed above, is the contribution made to sheet or jet disintegration by the instabilities created within the liquid itself, which are very dependent on liquid velocity. In airblast atomization, where high-velocity air impacts on a slow-moving liquid, the only factor promoting atomization is the relative velocity between the air and the liquid. This is also vitally important in pressure atomization but, by achieving this relative velocity through liquid motion instead of air motion, an important advantage is gained in that the liquid now makes an additional and independent contribution to its own disintegration, an effect that is either absent or negligibly small in airblast and air-assist atomization.

These arguments highlight the special importance of velocity in pressure atomization. The velocity at which the liquid is discharged from the nozzle has two separate effects on atomization. One important effect, which is dependent on the *absolute* velocity U_L , is in generating the turbulence and instabilities within the bulk liquid that contribute to the first stage of the atomization process. The other effect, which depends on the *relative* velocity U_R , is in promoting the atomization mechanisms that occur on the liquid surface and in the adjacent ambient gas. This means that Reynolds number, which is related to the bulk liquid, should be based on U_L , while Weber number, which is associated with events occurring through the action of the surrounding gas on the liquid surface, should be based on U_R .

Dividing $\sqrt{\text{We}}$ by Re to produce the Z number has the effect of eliminating velocity from both We and Re . This severely curtails its useful application to pressure atomization, where velocity is of paramount importance. In contrast, *multiplying* $\sqrt{\text{We}}$ by Re not only enhances the role of velocity but also provides a more accurate description of how SMD is affected by changes in liquid properties as well as nozzle size and geometry.

From simple geometric considerations, the initial thickness of the liquid sheet, t_s , after it is discharged from the nozzle can be related to the film thickness within the final orifice, t , by the equation

$$t_s = t \cos \theta \quad (6.21)$$

where θ is half the spray cone angle.

Substituting for t_s from Eq. (6.21) into Eq. (6.20), along with the appropriate terms for Re and We , gives

$$\text{SMD}_1 \propto \left(\frac{\sigma^{0.5} \mu_L}{\rho_A^{0.5} \rho_L U_R U_L} \right)^x (t \cos \theta)^{1-1.5x} \quad (6.22)$$

For the normal case of a nozzle spraying fluid into stagnant or slow-moving air, $U_R \approx U_L$. Also, $0.5 \rho_L U_L^2 = \Delta P_L$. Hence Eq. (6.22) can be simplified to

$$\text{SMD}_1 \propto \left(\frac{\sigma^{0.5} \mu_L}{\rho_A^{0.5} \Delta P_L} \right)^x (t \cos \theta)^{1-1.5x} \quad (6.23)$$

The term SMD_2 represents the final stage of the atomization process, in which the high relative velocity induced at the liquid/air interface by the rapidly evolving conical sheet causes the surface protuberances generated in the first stage to become detached and break down into ligaments and then drops. This final disintegration is opposed by surface tension forces, but Reynolds number is no longer relevant. Thus, we have

$$\frac{SMD_2}{t_s} \propto We^{-y} \propto \left(\frac{\sigma}{\rho_A U_R^2 t_s} \right)^y \quad (6.24)$$

Substituting for t_s from Eq. (6.21) converts Eq. (6.24) into

$$SMD_2 \propto \left(\frac{\sigma}{\rho_A U_R^2} \right)^y (t \cos \theta)^{1-y} \quad (6.25)$$

or, since $U_R \approx U_L$ and $\Delta P_L = 0.5 \rho_L U_L^2$,

$$SMD_2 \propto \left(\frac{\sigma \rho_L}{\rho_A \Delta P_L} \right)^y (t \cos \theta)^{1-y} \quad (6.26)$$

Equation (6.19) thus becomes

$$SMD = A \left(\frac{\sigma^{0.5} \mu_L}{\rho_A^{0.5} \Delta P_L} \right)^x (t \cos \theta)^{1-1.5x} + B \left(\frac{\sigma \rho_L}{\rho_A \Delta P_L} \right)^y (t \cos \theta)^{1-y} \quad (6.27)$$

where A and B are constants whose values depend on nozzle design.

Wang and Lefebvre [14] carried out a detailed experimental study of the factors governing the mean drop sizes produced by pressure-swirl atomizers. Extensive measurements of SMD were made using six simplex nozzles of different sizes and spray cone angles. Several different liquids were employed to provide a range of viscosity from 10^{-6} to 18×10^{-6} kg/m s (1 to 18 cS) and a range of surface tension from 0.027 to 0.0734 kg/s² (27 to 73.4 dyn/cm). The large amount of SMD data acquired in this investigation indicated values for x and y of 0.5 and 0.25, respectively. Moreover, the best fit to the experimental data was obtained using values of A and B of 4.52 and 0.39, respectively.

Substituting these values of x , y , A , and B into Eq. (6.27) gives

$$SMD = 4.52 \left(\frac{\sigma \mu_L^2}{\rho_A \Delta P_L^2} \right)^{0.25} (t \cos \theta)^{0.25} + 0.39 \left(\frac{\sigma \rho_L}{\rho_A \Delta P_L} \right)^{0.25} (t \cos \theta)^{0.75} \quad (6.28)$$

Inspection of this equation reveals several interesting features of pressure-swirl atomization:

1. Increase in spray cone angle reduces mean drop size. This result is fully consistent with the experimental data shown in Figs. 6.7 and 6.8 for water and in Figs. 6.4 and 6.5 for special liquids of high viscosity. In both sets of figures, the comparison is made between spray angles of 90° and 60°.

Equation (6.28) also suggests that an increase in spray angle should cause a

stronger dependence of SMD on both viscosity and pressure differential ΔP_L . It does this by causing SMD_2 to decline more than SMD_1 , thereby increasing the relative importance of the contribution made by SMD_1 to the overall mean drop size. Both these effects are evident in the data shown in Figs. 6.4 and 6.5. Comparison of these two figures shows that widening the spray angle not only improves atomization quality but also increases the dependence of SMD on both viscosity and liquid injection pressure. Also worthy of note in these figures is that the exponent of ΔP_L increases with increase in viscosity. This again stems directly from the added importance imparted to SMD_1 by increases in liquid viscosity.

2. Equation (6.28) indicates a surface tension exponent of 0.25. This agrees exactly with the findings of Jones [10] and is fairly close to Simmons and Harding's [11] value of 0.19. The data shown in Figs. 6.2 and 6.3 for water ($\sigma = 0.0734 \text{ kg/s}^2$) and "corrected" DF-2 ($\sigma = 0.027 \text{ kg/s}^2$) also suggest a surface tension exponent of 0.25.

3. As liquid density appears only in SMD_2 , its influence on mean drop size should generally be quite small, depending on the contribution of SMD_2 to SMD. The density of commercial hydrocarbon fuels is normally between 760 and 900 kg/m^3 , which means that changes in fuel type are not usually accompanied by large changes in density. This fact, combined with the small dependence of SMD on ρ_L shown in Eq. (6.28), implies that for most practical purposes the influence of density can usually be neglected when considering the effect of a change in fuel type on SMD.

4. As t is directly proportional to flow number [see Eq. (5.39)], Eq. (6.28) shows that the impact of a change in flow number on SMD should be less for liquids of high viscosity. This prediction is borne out by the experimental data plotted in Figs. 6.9 and 6.10 for liquid viscosities of 0.0026 and 0.009 kg/m s , respectively.

The same figures also show a stronger dependence of SMD on injection pressure for the higher-viscosity liquid. This is consistent with Eq. (6.28), which indicates exponents for ΔP_L varying from -0.25 for nonviscous liquids to -0.5 for liquids of high viscosity.

5. The influence of viscosity on mean drop size is demonstrated directly in Fig. 6.6. This figure illustrates the diminishing effect of flow number on SMD with increase in viscosity, as discussed above, and also shows that SMD becomes less dependent on viscosity with increase in flow number, as predicted by Eq. (6.28).

6. A further conclusion to be drawn from Eq. (6.28) is that the impact on SMD of changes in nozzle dimensions and operating conditions will vary depending on the level of liquid viscosity. From inspection of the equation it is evident that for liquids of high viscosity SMD will be more dependent on injection pressure differential ΔP_L and less dependent on nozzle flow number (via t) and spray cone angle than liquids of low viscosity.

Values of film thickness for insertion into Eq. (6.28) may be obtained from Eqs. (5.37) or (5.40) or from Eq. (5.44), with the value of the constant reduced from 3.66 to 2.7 following the recommendation of Suyari and Lefebvre [40], to

give

$$t = 2.7 \left[\frac{d_o F N \mu_L}{(\Delta P_L \rho_L)^{0.5}} \right]^{0.25} \quad (6.29)$$

Drop size equations for pressure-swirl atomizers are given in Table 6.2.

ROTARY ATOMIZERS

The process of centrifugal atomization has been studied by several workers, including Hinze and Milborn [41], Dombrowski and Fraser [42], and Tanasawa et al. [43]. It is described in reviews of atomization methods by Dombrowski and Munday [44], Christensen and Steely [45], Eisenklam [46], Matsumoto et al. [47], and Fraser et al. [48].

From the mechanisms for rotary atomization as outlined in Chapter 5, one would expect the mean drop size to diminish with increasing rotational speed and increase with flow rate and liquid viscosity. This expectation is confirmed by various theoretical and experimental investigations of the factors governing the

Table 6.2 Drop size relationships for pressure-swirl atomizers

Investigators	Equation	Remarks
Radcliffe [16]	$SMD = 7.3 \sigma^{0.6} \nu_L^{0.2} \dot{m}_L^{0.25} \Delta P_L^{-0.4}$	No effect of nozzle dimensions or air properties
Jasuja [15]	$SMD = 4.4 \sigma^{0.6} \nu_L^{0.16} \dot{m}_L^{0.22} \Delta P_L^{-0.43}$	No effect of nozzle dimensions or air properties
Babu et al. [37]	$SMD = 133 \frac{FN^{0.64291}}{\Delta P_L^{0.22565} \rho_L^{0.3215}}$	For $\Delta P_L < 2.8$ MPa
Babu et al. [37]	$SMD = 607 \frac{FN^{0.75344}}{\Delta P_L^{0.19936} \rho_L^{0.3767}}$	For $\Delta P_L > 2.8$ MPa
Jones [10]	$MMD = 2.47 \dot{m}_L^{0.315} \Delta P_L^{-0.47} \mu_L^{0.16} \mu_A^{-0.04} \sigma^{0.25} \rho_L^{-0.22} S$ $\times \left(\frac{l_o}{d_o} \right)^{0.03} \left(\frac{L_r}{D_s} \right)^{0.07} \left(\frac{A_p}{D_s d_o} \right)^{-0.13} \left(\frac{D_s}{d_o} \right)^{0.21}$	Suitable for large-capacity nozzles
Lefebvre [13]	$SMD = 2.25 \sigma^{2.25} \mu_L^{0.25} \dot{m}_L^{0.25} \Delta P_L^{-0.5} \rho_A^{-0.25}$	
Wang and Lefebvre [14]	$SMD = 4.52 \left(\frac{\sigma \mu_L^2}{\rho_A \Delta P_L^2} \right)^{0.25} (t \cos \theta)^{0.25}$ $+ 0.39 \left(\frac{\sigma \rho_L}{\rho_A \Delta P_L} \right)^{0.25} (t \cos \theta)^{0.75}$	Includes effect of spray cone angle Values of film thickness obtained from Eq. (5.37), (5.40), or (6.29)

atomization properties of rotary atomizers, which have resulted in a number of drop size relationships.

Karim and Kumar [49] used a photographic technique to measure the drop sizes produced by rotating cup atomizers. Their experiments were carried out with gasoline, using cups having exit diameters of 16, 19, and 20 mm and semicone angles of 10° , 20° , and 30° . Figure 6.16 is typical of the results obtained. It shows that, for a given rotational speed, the mean diameter of the drops formed increases almost linearly as the feed rate of the liquid is increased.

Figure 6.17 shows the influence of rotational speed on mean drop size. The data plotted in this figure were obtained with a 16-mm-diameter cup and semicone angle of 10° for various liquid temperatures. It can be seen that the mean diameter decreases by over an order of magnitude, down to about $10\text{ }\mu\text{m}$, as the rotational speed is increased. Similar trends are usually observed for other liquids and other cup geometries.

The adverse effect on atomization quality of lowering the liquid temperature is attributed by Karim and Kumar to the associated increase in viscosity, which decreases both the turbulence level and the rate of disintegration of the liquid into drops. Moreover, the surface tension of the liquid increases as the temperature goes down, which opposes the formation of any distortion or irregularity on the liquid surface, thereby delaying the formation of ligaments and drops.

In many practical applications of rotary atomizers, the drops produced are exposed to appreciable air movement. Figure 6.18 shows how the mean drop diameter varies as the air velocity parallel to the axis of the atomizer increases

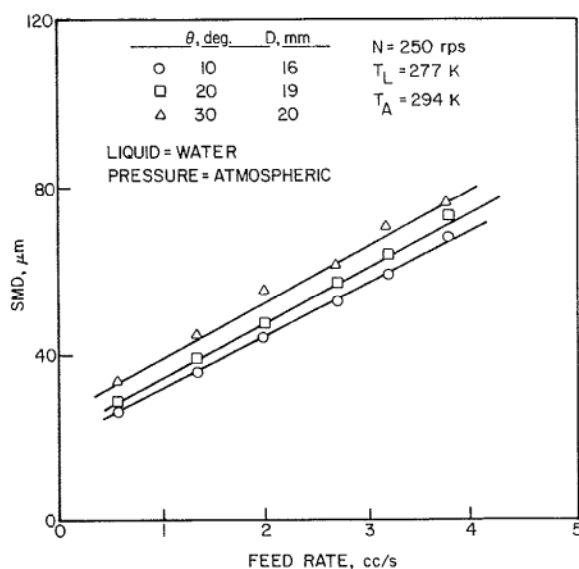


Figure 6.16 Typical variation of mean drop size with liquid feed rate (Karim and Kumar [49]).

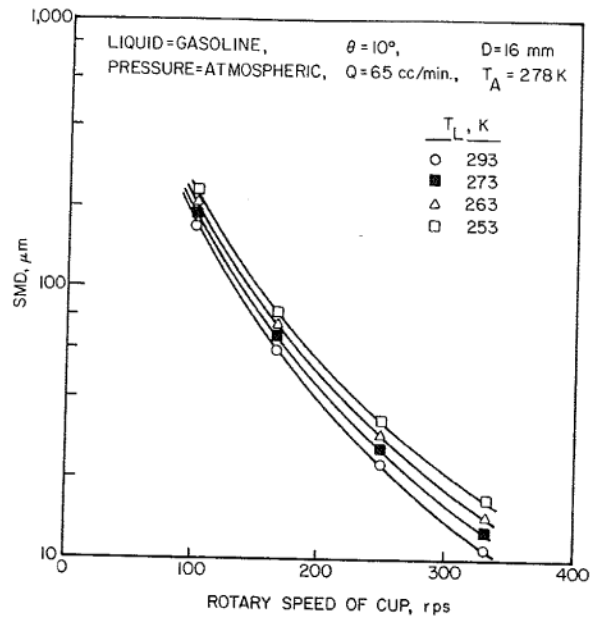


Figure 6.17 Typical variation of mean drop size with rotational speed (Karim and Kumar [49]).

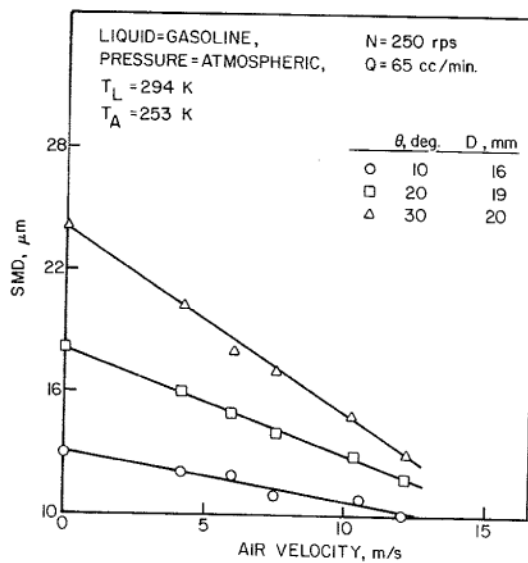


Figure 6.18 Typical variation of mean drop size with cross-flowing air velocity (Karim and Kumar [49]).

for a constant value of liquid flow rate. It can be seen that the mean diameter decreases with increase of air velocity. The extent of this decrease appears to be very dependent on the geometry of the atomizing cup. Lowering the airstream temperature, as shown in Fig. 6.19, tends to decrease the mean drop diameter almost linearly. This decrease in drop diameter is attributed to the increased density of the air, which increases the frictional forces on the liquid surface, thereby increasing the tendency for splitting up of the liquid into finer drops.

Drop Size Equations for Smooth Flat Vaneless Disks

The various modes whereby drops are produced by rotary atomization are discussed in Chapter 5. Several equations have been proposed for predicting mean drop sizes in each mode.

Direct drop formation. One of the earliest relationships for correlating the drop sizes produced by a flat disk rotary atomizer is the following due to Bar [50]:

$$D_{0.999} = \frac{1.07}{N} \left(\frac{\sigma}{d\rho_L} \right)^{0.5} \quad (6.30)$$

According to this equation the maximum drop size produced by direct drop formation increases with increase in surface tension and/or reduction in liquid

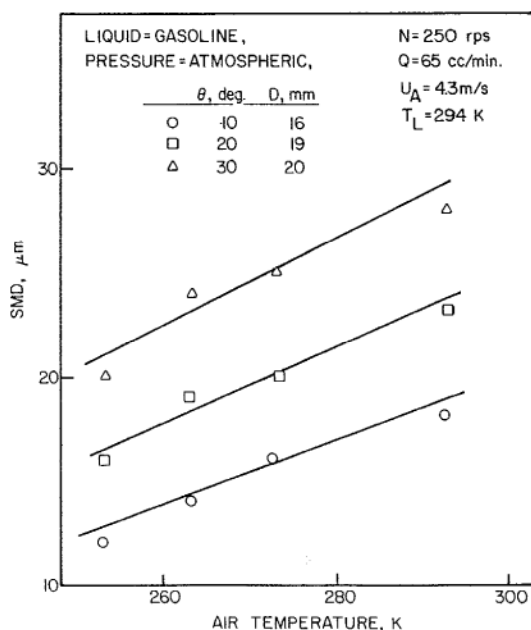


Figure 6.19 Typical variation of mean drop size with air temperature (Karim and Kumar [49]).

density and is inversely proportional to the speed of rotation. A notable feature of Eq. (6.30) is the absence of any term to account for the influence of liquid viscosity on drop size. This omission was remedied some 40 years later by Tanasawa et al. [51]. Their modified equation for Sauter mean diameter is

$$\text{SMD} = \frac{27}{N} \left(\frac{\sigma}{d\rho_L} \right)^{0.5} \left(1 + 0.003 \frac{Q}{dv_L} \right) \quad (6.31)$$

According to Matsumoto et al. [52], the dimensionless mean drop size produced in this mode of drop formation is solely dependent on Weber number. We have

$$\frac{\text{SMD}}{d} = 1.6 \text{ We}^{-0.523} \quad (6.32)$$

where $\text{We} = \rho_L d^3 \omega^2 / 8\sigma$.

Atomization by ligament formation. Walton and Prewett [53] were among the first to investigate the production of drops by ligament formation at the periphery of a rotating disk. Their equation for maximum drop size is almost identical to Bar's Eq. (6.30) for direct drop formation. We have

$$D_{0.999} = \frac{0.87}{N} \left(\frac{\sigma}{d\rho_L} \right)^{0.5} \quad (6.33)$$

Another well-known expression for the drop sizes produced via ligament formation is the following due to Oyama and Endou [54]:

$$\text{SMD} = \frac{0.177 Q^{0.2}}{Nd^{0.3}} \quad (6.34)$$

This equation was derived from analysis of drop size measurements carried out with water as the working fluid. It contains no terms to denote liquid properties, so its range of useful application is limited to water. However, it does have the merit of indicating that drop sizes should increase with increase in liquid flow rate, which is consistent with experimental observations. This effect of higher flow rate may be attributed to thickening of the liquid sheet, which on disintegration forms ligaments of larger diameter.

The relationship between ligament diameter and mean drop size has been expressed by Matsumoto and Takashima [55] as

$$\frac{\text{SMD}}{d_1} = (1.5\pi)^{1/3} \left(1 + 3 \frac{\text{We}}{\text{Re}^2} \right)^{1/6} \quad (6.35)$$

where d_1 is the ligament diameter, We (Weber number) is $\rho_L d^3 \omega^2 / 8\sigma$, and Re (Reynolds number) is $\rho_L d^2 \omega / 4\mu$.

Tanasawa et al. [51] proposed an equation for predicting the mean drop sizes produced via ligament formation that takes full account of variations in disk di-

ameter, rotational speed, liquid flow rate, and liquid properties. We have

$$\text{SMD} = 0.119 \frac{Q^{0.1} \sigma^{0.5}}{Nd^{0.5} \rho_L^{0.4} \mu_L^{0.1}} \quad (6.36)$$

An interesting feature of this equation is that drop sizes are predicted to increase slightly with reduction in liquid viscosity. Usually, of course, reduction in liquid viscosity leads to finer atomization. An equation for mean drop size that is more consistent with this general rule is the following, proposed by Kayano and Kamiya [56]:

$$\text{SMD} = 0.26N^{-0.79} Q^{0.32} d^{-0.69} \rho_L^{-0.29} \sigma^{0.26} (1 + 1.027\mu_L^{0.65}) \quad (6.37)$$

Atomization by sheet disintegration. According to Fraser and Eisenklam [57], an appropriate drop size relationship for this mode of atomization is the following:

$$D_{10} = \frac{0.76}{N} \left(\frac{\sigma}{d\rho_L} \right)^{0.5} \quad (6.38)$$

It is of interest to note the close similarity between this equation for sheet disintegration and Eqs. (6.30) and (6.33), which relate to direct drop formation and atomization via ligament formation, respectively.

A similar relationship, with an additional term to include the influence of liquid flow rate on mean drop size, is the following, due to Tanasawa et al. [51]:

$$\text{SMD} = 15.6 \frac{Q^{0.5}}{N} \left(\frac{\sigma}{d^2 \rho_L} \right)^{0.4} \quad (6.39)$$

Drop Size Equations for Vaneless Disks

For vaneless disks that do not fall in the category of smooth and flat, the drop size equations corresponding to the three modes of atomization are the following:

Direct drop formation [90]:

$$D_{10} = \left[\frac{3Q\mu_L}{2\pi\rho_L(\pi dN)^2 \sin \theta} \right]^{1/3} \quad (6.40)$$

Atomization by ligament formation [41]:

$$D_{10} = \left[0.77 \frac{Q}{Nd} \left(\frac{\rho_L N^2 d^3}{\sigma} \right)^{-5/12} \left(\frac{\rho_L \sigma d}{\mu_L^2} \right)^{-1/16} \right]^{0.5} \quad (6.41)$$

Atomization by sheet disintegration [108]:

$$D_{10} = 4.42 \left(\frac{\sigma}{\omega^2 \rho_L d} \right)^{0.5} \quad (6.42)$$

Drop Size Equations for Atomizer Wheels

Some commercial rotary atomizers consist of disks or wheels fitted with vanes. For these devices, additional geometric variables to be considered are the number of vanes n and the vane height h . According to Masters [58], suitable drop size equations for atomizer wheels are the following.

Friedman et al. [59]:

$$\text{SMD} = 0.44d \left(\frac{\dot{m}_p}{\rho_L N d^2} \right)^{0.6} \left(\frac{\mu_L}{\dot{m}_p} \right)^{0.2} \left(\frac{\sigma \rho_L n h}{\dot{m}_p^2} \right)^{0.1} \quad (6.43)$$

Herring and Marshall [60]:

$$\text{SMD} = \frac{3.3 \times 10^{-9} K \dot{m}_L^{0.24}}{(Nd)^{0.83} (nh)^{0.12}} \quad (6.44)$$

where $K = 8.5 \times 10^5$ to 9.5×10^5 . See Masters' Table 6.11 [58].

Fraser et al. [61]:

$$\text{SMD} = 0.483N^{-0.6} \rho_L^{-0.5} \left(\frac{\mu_L \dot{m}_L}{d} \right)^{0.2} \left(\frac{\sigma}{nh} \right)^{0.1} \quad (6.45)$$

Scott et al. [62]:

$$\text{SMD} = 6.3 \times 10^{-4} \dot{m}_p^{0.171} (ndN)^{-0.537} \mu_L^{-0.017} \quad (6.46)$$

Drop size equations for smooth flat vaneless disks, other types of vaneless disks, and atomizer wheels are listed in Tables 6.3, 6.4, and 6.5, respectively. Other equations for the mean drop sizes produced by rotary atomizers may be found in the literature, but not all of them can be recommended for prediction purposes, because they were derived from data sources that include more than one mechanism of drop formation.

AIR-ASSIST ATOMIZERS

In attempting to review published work on air-assist atomizer performance, a major difficulty involved is that of differentiating between air-assist and airblast atomizers, since both types of nozzle employ high-velocity air to achieve atomization and their mean geometric features are always roughly the same. For gas turbine applications, the most common method of differentiating between the two systems is on the basis of air velocity. Air-assist nozzles employ very high air velocities that usually necessitate an external supply of high-pressure air, while the lower air velocity requirements of airblast systems can usually be met by utilizing the pressure differential across the combustor liner. However, as most of the reported studies on air atomization have covered fairly wide ranges of air

Table 6.3 Drop size equations for smooth flat vaneless disks

Investigators	Equation	Remarks
Bar [50]	Direct drop formation $D_{0.999} = \frac{1.07}{N} \left(\frac{\sigma}{d\rho_L} \right)^{0.5}$	No effect of liquid viscosity
Tanasawa et al. [51]	$\text{SMD} = \frac{27}{N} \left(\frac{\sigma}{d\rho_L} \right)^{0.5} \left(1 + 0.003 \frac{Q}{dv_L} \right)$	Includes all liquid properties
Matsumoto et al. [52]	$\text{SMD} = 1.6d \left(\frac{8\sigma}{\rho_L d^3 \omega^2} \right)^{0.523}$	Mean drop size dependent on Weber number
Walton and Prewett [53]	Atomization by ligament formation $D_{0.999} = \frac{0.87}{N} \left(\frac{\sigma}{d\rho_L} \right)^{0.5}$	No effect of liquid viscosity
Oyama and Endou [54]	$\text{SMD} = \frac{0.177 Q^{0.2}}{Nd^{0.3}}$	Limited to water. Includes liquid flow rate
Matsumoto and Takashima [55]	$\frac{\text{SMD}}{d_l} = (1.5\pi)^{1/3} \left(1 + \frac{3\text{We}}{\text{Re}^2} \right)^{1/6}$	d_l = ligament diameter We = Weber number = $\rho_L d^3 \omega^2 / 8\sigma$ Re = Reynolds number = $\rho_L d^2 \omega / 4\mu$
Tanasawa et al. [51]	$\text{SMD} = 0.119 \frac{Q^{0.1} \sigma^{0.5}}{Nd^{0.5} \rho_L^{0.4} \mu_L^{0.1}}$	Includes all relevant parameters
Kayano and Kamiya [56]	$\text{SMD} = 0.26N^{-0.79} Q^{0.32} d^{-0.69} \rho_L^{-0.29} \sigma^{0.26} (1 + 1.027 \mu_L^{0.65})$	Includes all relevant parameters
Fraser and Eisenklam [57]	Atomization by sheet disintegration $D_{10} = \frac{0.76}{N} \left(\frac{\sigma}{d\rho_L} \right)^{0.5}$	
Tanasawa et al. [51]	$\text{SMD} = 15.6 \frac{Q^{0.5}}{N} \left(\frac{\sigma}{d^2 \rho_L} \right)$	Includes effect of liquid flow rate

velocity, the decision to describe any given atomizer as “air-assist” or “air-blast” is sometimes almost arbitrary.

In general, air-assist nozzles require less air than airblast nozzles, which makes them especially attractive for use in combustion systems that are required to operate with minimum excess air over a wide turndown ratio. They are usually classified into *internal-mixing* and *external-mixing* types. Internal-mixing nozzles have the advantage of wide flexibility and are particularly useful for atomizing highly viscous liquids and liquid slurries. However, their aerodynamic and fluid dynamic flow patterns are highly complex, due to the intense mixing of gas and liquid within the mixing chamber.

Table 6.4 Drop size equations for vaneless disks

Investigators	Equation
Fraser et al. [90]	Direct drop formation $D_{10} = \left[\frac{3Q\mu_L}{2\pi\rho_L(\pi dN)^2 \sin \theta} \right]^{1/3}$
Hinze and Milbourn [41]	Atomization by ligament formation $D_{10} = \left[0.77 \frac{Q}{Nd} \left(\frac{\rho_L N^2 d^3}{\sigma} \right)^{-5/12} \left(\frac{\rho_L \sigma d}{\mu_L^2} \right)^{-1/16} \right]^{0.5}$
Hege [108]	Atomization by sheet disintegration $D_{10} = 4.42 \left(\frac{\sigma}{\omega^2 \rho_L d} \right)^{0.5}$

External-mixing atomizers usually employ a high-velocity stream of air or steam that flows from an annular nozzle and is arranged to impinge at some angle onto a jet or conical sheet of liquid that is created at the center of the atomizer. External-mixing air-assist nozzles have many features in common with airblast atomizers, and the drop size equations that have been established for one type of nozzle are generally relevant to the other.

Internal-Mixing Nozzles

Wigg's [63] analysis of the mechanism of airblast atomization highlighted the importance of the kinetic energy of the atomizing air and indicated that the energy difference between the inlet air and the emerging spray was the dominant factor affecting mean drop size. In a later work [64], Wigg used the wax spray data of Clare and Radcliffe [65] and Wood [66], obtained on the NGTE atomizer shown in Fig. 4.42, to derive the following dimensionless expression for MMD:

Table 6.5 Drop size equations for atomizer wheels

Investigators	Equation
Friedman et al. [59]	$\text{SMD} = 0.44d \left(\frac{\dot{m}_L}{\rho_L N d^2} \right)^{0.6} \left(\frac{\mu_L}{\dot{m}_L} \right)^{0.2} \left(\frac{\sigma \rho_L n h}{\dot{m}_L^2} \right)^{0.1}$
Herring and Marshall [60]	$\text{SMD} = \frac{3.3 \times 10^{-9} K \dot{m}_L^{0.24}}{(Nd)^{0.83} (nh)^{0.12}}$
Fraser et al. [61]	$\text{SMD} = 0.483N^{-0.6} \rho_L^{-0.5} \left(\frac{\mu_L \dot{m}_L}{d} \right)^{0.2} \left(\frac{\sigma}{nh} \right)^{0.1}$
Scott et al. [62]	$\text{SMD} = 6.3 \times 10^{-4} \dot{m}_L^{0.171} (ndN)^{-0.537} \mu_L^{-0.017}$

$$\text{MMD} = 20v_L^{0.5}\dot{m}_L^{0.1}\left(1 + \frac{\dot{m}_L}{\dot{m}_A}\right)^{0.5} h^{0.1}\sigma^{0.2}p_A^{-0.3}U_R^{-1.0} \quad (6.47)$$

where h is the height of the air annulus in meters.

Mullinger and Chigier [67] studied the performance of an “internal-mixing” twin-fluid atomizer of the type illustrated in Fig. 4.43, in which liquid fuel is injected into a mixing chamber along with compressed air or steam. Some atomization occurs within the mixing chamber, but most of the liquid emerges from the atomizer in the form of a sheet that is then shattered into drops by the atomizing fluid. They found good agreement between their experimental data and predictions based on Eq. (6.47), even at air/liquid mass ratios as low as 0.005, which is one-tenth the lowest value tested by Wigg [63].

Application of Eq. (6.47) to water spray data [68, 69] gave a poor correlation; this was attributed by Wigg to recombination or coalescence of drops in the spray. To account for this effect, he suggested that whenever the conditions are conducive to drop coalescence, the value of MMD calculated from Eq. (6.47) should be multiplied by the empirical expression

$$1 + 5.0 \left(\frac{\dot{m}_L}{\dot{m}_A}\right)^{0.6} \dot{m}_L^{0.1} \quad (6.48)$$

From studies carried out on the atomization of water, with air as the atomizing fluid, Sakai et al. [70] derived the following empirical formula for mean drop size.

$$\text{SMD} = 14 \times 10^{-6} d_o^{0.75} \left(\frac{\dot{m}_L}{\dot{m}_A}\right)^{0.75} \quad (6.49)$$

This equation applies over a range of water flow rates from 30 to 100 kg/h and values of $\dot{m}_L/\dot{m}_A$ from 5 to 100.

Inamura and Nagai [71] flowed air at uniform velocity through a vertically mounted cylindrical nozzle and used a thin annular slot, 1 mm in width, to inject liquid along the inside wall of the nozzle. Water, ethanol, and glycerin solutions were used to investigate the effects of liquid properties on atomization. Drop sizes were measured from spray samples collected on oil-coated glass slides.

For low liquid flow rates and low air velocities, Inamura and Nagai observed disturbances of large wavelength in the liquid flowing along the wall, causing the liquid film to break down into drops through the formation of unstable ligaments at the end of the wall. They called this process “atomization by ligament formation.” Increase in air velocity and liquid flow rate caused the wavelength of the disturbances to diminish, so that the liquid emerged at the nozzle edge in the form of a thin continuous film. They termed the resulting atomization of this film by the high-velocity air “atomization by film formation.” Some of the results obtained for the effects on SMD of variations in air velocity and air/liquid mass ratio are shown in Fig. 6.20. The curves drawn in this figure exhibit a change in slope, which is attributed by Inamura and Nagai to the transition from one mode

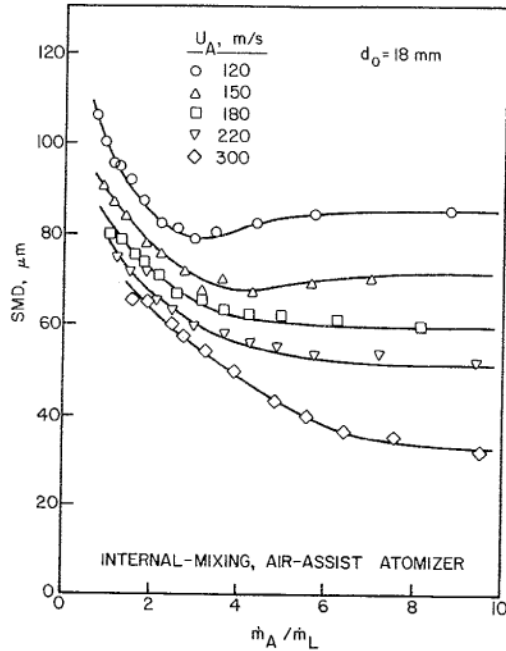


Figure 6.20 Variation of mean drop size with air/liquid mass ratio (Inamura and Nagai [71]).

of atomization to the other. It is of interest to note that the value of $\dot{m}_A/\dot{m}_L$ at which this transition occurs increases with increase in air velocity.

The influence of viscosity on SMD is illustrated in Fig. 6.21. The data plotted in this figure confirm that this type of atomizer is no different from all other types of twin-fluid atomizer in demonstrating that increase in liquid viscosity leads to coarser atomization, regardless of air velocity.

External-Mixing Nozzles

Inamura and Nagai [71] also investigated the performance of an external-mixing atomizer. The nozzle they used is designed to produce a flat circular sheet of liquid whose thickness can be varied from 0 to 0.7 mm by rotation of a screw. This liquid sheet is then deflected downward and atomized by an annular air jet, whose angle of impingement, relative to the nozzle axis, can also be varied. By using various ethanol and glycerin solutions, Inamura and Nagai were able to examine the effects of surface tension and viscosity on mean drop size. Analysis of the experimental data yielded the following dimensionally correct empirical equation for mean drop size:

$$\frac{SMD}{t} = \left[1 + \frac{16850 \text{ Oh}^{0.5}}{\text{We}(\rho_L/\rho_A)} \right] \left[1 + \frac{0.065}{(\dot{m}_A/\dot{m}_L)^2} \right] \quad (6.50)$$

where t = initial film thickness = $D_o h / D_{an}$

D_o = outer diameter of pressure nozzle

D_{an} = diameter of annular gas nozzle

h = slot width of pressure nozzle

Oh = stability number = $(\mu_L^2 / \rho_L t \sigma)^{0.5}$

We = Weber number = $\rho_A U_A^2 t / \sigma$

Elkottb et al. [72] used 40 different nozzle configurations to examine the effects of nozzle geometry and operating variables on the characteristics of kerosine sprays. The kerosine employed had the following specifications: $\rho_L = 800 \text{ kg/m}^3$, $\sigma = 0.0304 \text{ kg/s}^2$, $\mu_L = 0.00335 \text{ kg/m s}$. Mean drop sizes were measured by the slide-sampling method. Up to 4000 drops were used to determine the SMD of any given spray.

Of special interest in this study are the data obtained on the influence of ambient air pressure on mean drop size. These results are shown in Fig. 6.22. Correlation of the experimental data led to the following dimensionally correct equation for mean drop size.

$$\text{SMD} = 51 d_o \text{Re}^{-0.39} \text{We}^{-0.18} \left(\frac{\dot{m}_L}{\dot{m}_A} \right)^{0.29} \quad (6.51)$$

where $\text{Re} = \rho_L U_R d_o / \mu_L$ and $\text{We} = \rho_L d_o U_R^2 / \sigma$.

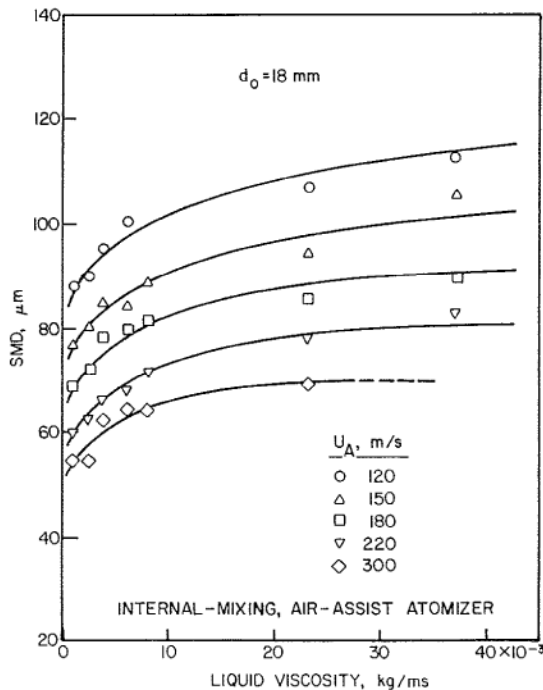


Figure 6.21 Variation of mean drop size with liquid viscosity (Inamura and Nagai [71]).

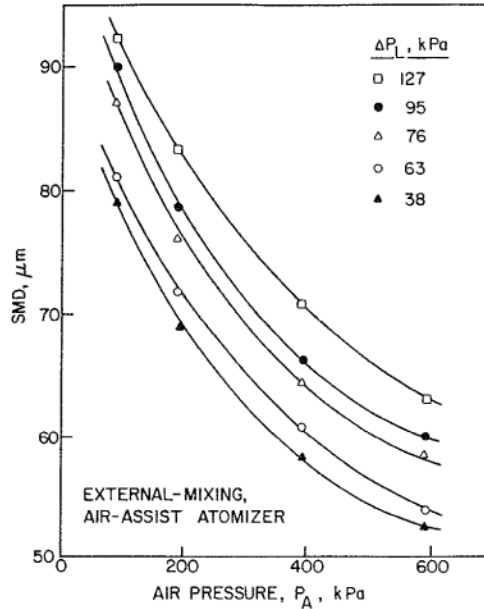


Figure 6.22 Influence of ambient air pressure on mean drop size (Elkoth et al. [72]).

In an attempt to bridge the gap between pressure atomizers and air atomizers, Simmons [18] carried out a large number of measurements of SMD on both types of atomizer. His analysis of these data led to the following equation for SMD.

$$\text{SMD} = C \left(\frac{\rho_L^{0.25} \mu_L^{0.06} \sigma^{0.375}}{\rho_A^{0.375}} \right) \left(\frac{\dot{m}_L}{\dot{m}_L U_L + \dot{m}_A U_A} \right)^{0.55} \quad (6.52)$$

In the above equation C is a constant whose value depends on nozzle design. The liquid properties ρ_L , μ_L , and σ are expressed relative to those of standard calibrating fluid [MIL-C-7024II] at room temperature, while air density is expressed as the value relative to sea level, room temperature conditions.

Suyari and Lefebvre [73] investigated the atomizing performance of an external-mixing, air-assist nozzle of the type shown in Fig. 6.23. Essentially it comprises a simplex pressure-swirl nozzle surrounded by a coflowing stream of swirling air. Drop sizes were measured using a Malvern particle sizing instrument. The liquids employed included water, gasoline, kerosene, and diesel oil.

Figure 6.24 exhibits some interesting features that appear to be characteristic of this type of atomizer. First, it is clear that increase in ΔP_A , i.e., increase in atomizing air velocity, has a very beneficial effect on atomization quality, especially at low liquid flow rates. Of more interest, however, is the observation that SMD decreases with increase in liquid flow rate when ΔP_A is low but increases with increase in liquid flow rate when ΔP_A is high. This is attributed to the fact that when atomizing air velocities are low ($\Delta P_A/P_A < 3\%$) the system

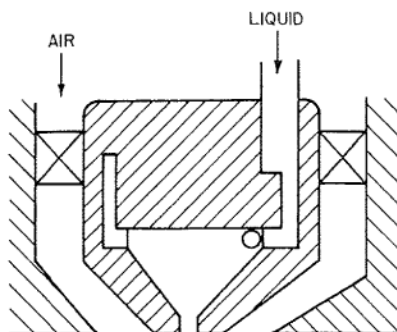


Figure 6.23 Schematic diagram of external-mixing air-assist nozzle (Suyari and Lefebvre [73]).

operates primarily as a simplex pressure-swirl nozzle. Thus increase in liquid flow rate reduces SMD by increasing the injection pressure. However, when the same nozzle is operating in an airblast mode, i.e., when $\Delta P_A/P_A$ is high ($>4\%$), increase in liquid flow rate impairs atomization quality by lowering both the atomizing air/liquid ratio and the relative velocity between the air and the liquid.

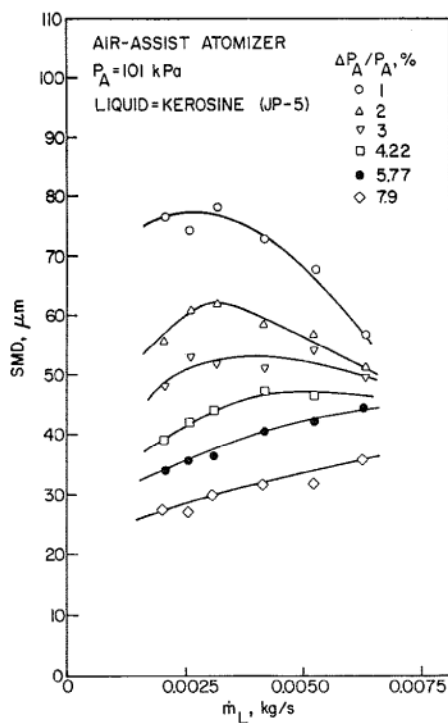


Figure 6.24 Influence of liquid flow rate and atomizing air velocity on mean drop size (Suyari and Lefebvre [73]).

These considerations help to explain the shape of the curve drawn in Fig. 6.24 for a value of $\Delta P_A/P_A$ of 2%. This curve shows SMD increasing with increase in liquid flow rate until a maximum value is reached, beyond which any further increase in $\dot{m}_L$ causes SMD to decline. The unusual shape of this curve can be explained in terms of the relative velocity between the conical sheet of liquid produced by the simplex atomizer and the surrounding coflowing air. At low liquid flow rates the air velocity is higher than the liquid velocity. Thus, in this regime any increase in liquid flow rate, i.e., any increase in liquid velocity, will impair atomization quality by reducing the relative velocity between the air and the liquid. At the peak of the curve, where atomization quality is poorest, the relative velocity is zero and the interaction between the liquid sheet and the surrounding air is minimal. Beyond this point the liquid velocity exceeds the air velocity, and any further increase in liquid flow rate serves to improve atomization by increasing the relative velocity between the liquid and the air.

These arguments suggest that for this type of nozzle there will always be one value of liquid flow rate for which SMD is a maximum. The particular value of $\dot{m}_L$ at which this occurs will be that for which the liquid velocity just equals the air velocity. Now in most practical forms of air-assist atomizer, the air and the liquid issuing from the nozzle both have radial and tangential as well as axial components of velocity. In this complex two-phase flow field the term "relative velocity" is difficult to define. Nevertheless, for any given air velocity there will always be one particular value of liquid velocity at which the level of interaction between the fuel and the air is a minimum. With increases in atomizing air velocity, the condition of zero relative velocity will obviously be reached at higher levels of liquid velocity. Thus one would expect the value of $\dot{m}_L$ at which the SMD attains its maximum value to increase with increase in $\Delta P_A/P_A$, and this is evident in the changing shapes of the curves drawn in Fig. 6.24.

Figure 6.24 demonstrates that atomization quality is improved by an increase in liquid injection pressure when the system is operating in the pressure-swirl mode and by an increase in atomizing air velocity and/or air/liquid ratio when it is operating as an air atomizer. This is fully consistent with all previous experience on pressure-swirl and airblast nozzles. However, it also shows that the relative velocity between the air and the liquid has an additional independent effect over and above the effects of air velocity and liquid velocity.

The results obtained when operating with JP 4 fuel at atmospheric pressure are shown in Fig. 6.25, in which measured values of SMD are plotted against liquid injection pressure differential, ΔP_L . One interesting feature of this figure is that except at the lowest liquid flows, where liquid pressure alone is too low to achieve good atomization, the atomization quality is actually higher without atomizing air than when air is supplied at a pressure differential of 2%. In fact, over a wide range of liquid pressures, an atomizing air pressure differential of at least 3% is needed to provide the same atomization quality that is achieved with no air assistance at all. Nevertheless, the very substantial improvements in atomization quality that are derived from air assistance at low liquid injection pressures are also apparent in Fig. 6.25.

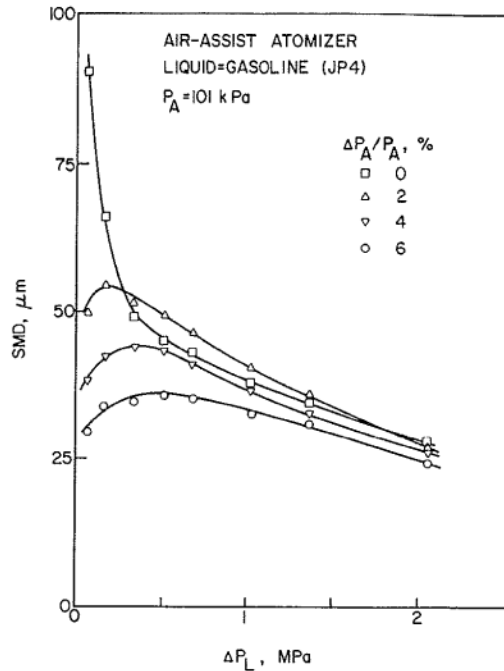


Figure 6.25 Influence of liquid pressure differential and atomizing air velocity on mean drop size (Suyari and Lefebvre [73]).

From analysis of their experimental data, Suyari and Lefebvre [73] concluded that the two key factors governing the atomization quality achieved with external-mixing air-assist atomizers, when operating with low-viscosity liquids, are air dynamic pressure, $0.5\rho_A U_A^2$ (as indicated by ΔP_A), and the relative velocity between the liquid sheet produced by the pressure-swirl nozzle and the surrounding air. Thus relative velocity appears to play an important and independent role, presumably through its influence on the development of capillary waves (ripples) on the initially smooth liquid surface and the production of unstable ligaments. As the relative velocity increases, the size of the ligaments decreases, their life becomes shorter, and, on their collapse when exposed to the high-velocity airstream, much smaller drops are formed, in accordance with Rayleigh theory [74].

These findings raise doubts concerning the validity of some existing equations for SMD. For example, Eq. (6.52) suggests that increase in either U_L or U_A should be beneficial to atomization quality. However, the results show that whereas increase in U_A is usually conducive to better atomization, especially at low liquid injection pressures, increase in U_L can either raise or lower SMD, depending on the initial values of U_L and U_A . If the initial value of U_L is less than U_A , then an increase in U_L will impair atomization by reducing the relative velocity between

the liquid and the air. Only if the initial value of U_L exceeds U_A will further increase in U_L lead to better atomization.

From the results of their study on external-mixing air-assist atomizers, Suyari and Lefebvre [73] drew the following conclusions.

1. For any given liquid, the key factors governing atomization quality are the dynamic pressure of the atomizing air and the relative velocity between the liquid and the surrounding air.
2. For any given value of air velocity, continual increase in liquid injection pressure from an initial value of zero produces an increase in SMD up to a maximum value, beyond which further increase in injection pressure causes SMD to decline.
3. The liquid injection pressure at which the SMD attains its maximum value increases with increase in atomizing air velocity.
4. Whereas increase in air velocity is usually beneficial to atomization quality, increase in liquid velocity may help or hinder atomization, depending on whether it increases or decreases the relative velocity between the liquid and the surrounding air.

The expressions for mean drop size presented above are summarized in Table 6.6.

AIRBLAST ATOMIZERS

Although the principle of airblast atomization has been known for a long time, strong interest in the design and performance of airblast nozzles was not evoked until the mid-1960s, when their potential for achieving significant reductions in soot formation and exhaust smoke in gas turbine engines of high pressure ratio started to become recognized.

Most of the systems now in service are of the *prefilming* type, in which the fuel is first spread out into a thin continuous sheet and then subjected to the atomizing action of high-velocity air. In other designs, the fuel is injected into the high-velocity airstream in the form of one or more discrete jets. The atomization performance of prefilming nozzles is generally superior to that of plain-jet nozzles [75, 76], but they are fully effective only when both sides of the liquid sheet are exposed to the air. This requirement introduces a complication in design, since it usually means arranging for two separate airflows through the atomizer. For this reason, the plain-jet type of airblast atomizer, in which the fuel is not transformed into a thin sheet but instead is injected into the high-velocity airstream in the form of discrete jets, is sometimes preferred.

Plain Jet

The first major study of airblast atomization was conducted over 40 years ago by Nukiyama and Tanasawa [68] on a plain-jet airblast atomizer, as illustrated in

Table 6.6 Drop size equations for air-assist atomizers

Investigators	Atomizer type	Equations	Remarks
Wigg [64]	Internal mixing	$MMD = 20v_L^{0.5} \dot{m}_L^{0.1} \left(1 + \frac{\dot{m}_L}{\dot{m}_A}\right)^{0.5} h^{0.1} \sigma^{0.2} \rho_A^{-0.3} U_R^{-1.0}$	h = height of air annulus, m
Sakai et al. [70]	Internal mixing	$SMD = 14 \times 10^{-6} d_o^{0.75} \left(\frac{\dot{m}_L}{\dot{m}_A}\right)^{0.75}$	d_o = nozzle diameter
Inamura and Nagai [71]	External mixing	$\frac{SMD}{t} = \left[1 + \frac{16850 \text{ Oh}^{0.5}}{\text{We}(\rho_L/\rho_A)}\right] \left[1 + \frac{0.065}{(\dot{m}_A/\dot{m}_L)^2}\right]$	$\text{Oh} = \left(\frac{\mu_L^2}{\rho_L t \sigma}\right)^{0.5}$
Elkoth et al. [72]	External mixing	$SMD = 51 d_o \text{ Re}^{-0.39} \text{We}^{-0.18} \left(\frac{\dot{m}_L}{\dot{m}_A}\right)^{0.29}$	$\text{Re} = \frac{\rho_L U_R d_o}{\mu_L}$ $\text{We} = \frac{\rho_L d_o U_R^2}{\sigma}$
Simmons [18]	General	$SMD = C \left(\frac{\rho_L^{0.25} \mu_L^{0.06} \sigma^{0.375}}{\rho_A^{0.375}}\right) \left(\frac{\dot{m}_L}{\dot{m}_L U_L + \dot{m}_A U_A}\right)^{0.55}$	

Fig. 6.26. The drop sizes were measured by collecting samples of the spray on oil-coated glass slides. Drop size data were correlated by the following empirical equation for the SMD:

$$SMD = \frac{0.585}{U_R} \left(\frac{\sigma}{\rho_L}\right)^{0.5} + 53 \left(\frac{\mu_L^2}{\sigma \rho_L}\right)^{0.225} \left(\frac{Q_L}{Q_A}\right)^{1.5} \quad (6.53)$$

It is of interest to note that the right-hand side of Eq. (6.53) is expressed as the sum of two separate terms, one of which is dominated by relative velocity and surface tension and the other by viscosity.

Although Eq. (6.53) is not dimensionally correct, it nevertheless allows some useful conclusions to be drawn. For example, for liquids of low viscosity it shows that mean drop size is inversely proportional to the relative velocity between the air and the liquid, while for large values of air/liquid ratio the influence of viscosity on SMD becomes negligibly small.

Equation (6.53) can be made dimensionally correct by introducing a term to denote length, raised to the 0.5 power. One obvious choice for this length is the diameter of the liquid orifice or air nozzle. However, from tests carried out with different sizes and shapes of nozzles and orifices, Nukiyama and Tanasawa concluded that these factors have virtually no effect on mean drop size. Thus, the absence of atomizer dimensions is a notable feature of Eq. (6.53).

Another significant omission is air density, which was kept constant (at the normal atmospheric value) in all experiments. This represents a serious limitation, since it prohibits the application of Eq. (6.53) to the many types of atomizers that are required to operate over wide ranges of air pressure and temperature.

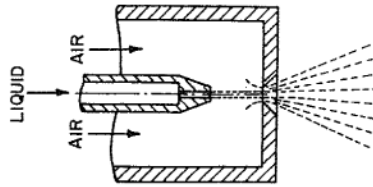


Figure 6.26 Nukiyama-Tanasawa plain-jet airblast atomizer [68].

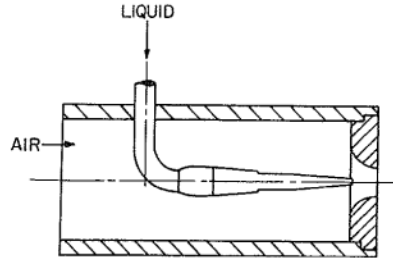


Figure 6.27 Lorenzetto-Lefebvre plain-jet airblast atomizer [75].

The performance of plain-jet atomizers was investigated in detail by Lorenzetto and Lefebvre [75], using a specially designed system in which liquid physical properties, air/liquid ratio, air velocity, and atomizer dimensions could be varied independently over wide ranges and examined for their separate effects on spray quality. A cross-sectional drawing of the atomizer is shown in Fig. 6.27.

Some of the results of this study are shown in Figs. 6.28 to 6.32. They show that atomization quality deteriorates with increases in viscosity and surface tension, in the same manner as with pressure-swirl and rotary atomizers. For liquids of low viscosity, it was found that mean drop size is inversely proportional to the relative velocity between the air and the liquid at the nozzle exit. It was also observed, in agreement with Nukiyama and Tanasawa [68], that nozzle dimensions appear to have little influence on mean drop size for liquids of low viscosity. For liquids of high viscosity, SMD varies roughly in proportion to $d_o^{0.5}$.

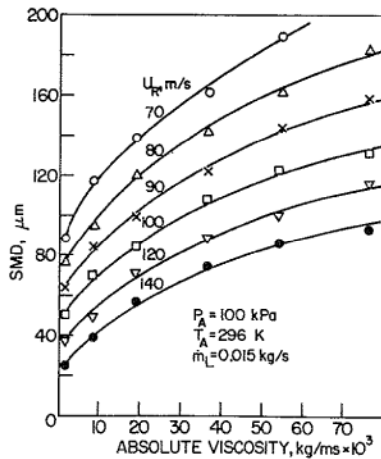


Figure 6.28 Variation of mean drop size with liquid viscosity for a plain-jet airblast atomizer (Lorenzetto and Lefebvre [75]).

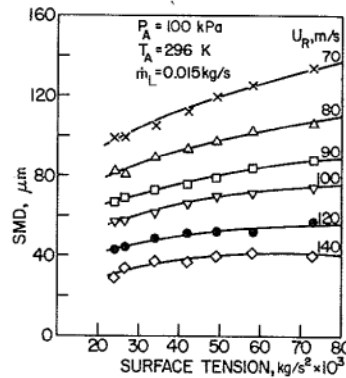


Figure 6.29 Variation of mean drop size with surface tension for a plain-jet airblast atomizer (Lorenzetto and Lefebvre [75]).

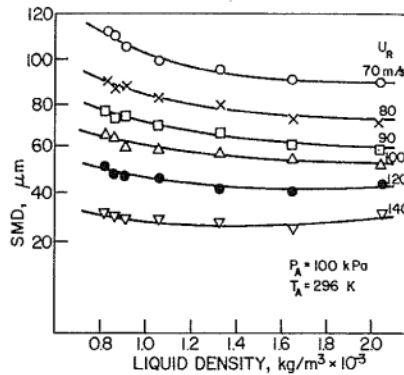


Figure 6.30 Variation of mean drop size with liquid density for a plain-jet airblast atomizer (Lorenzetto and Lefebvre [75]).

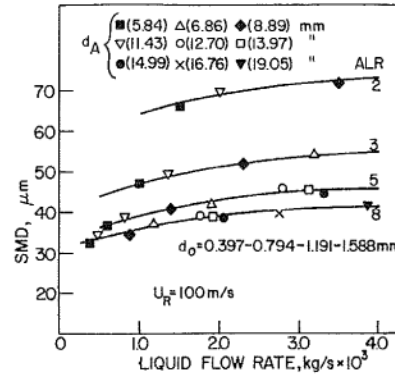


Figure 6.31 Variation of mean drop size with liquid flow rate for a plain-jet airblast atomizer (Lorenzetto and Lefebvre [75]).

The atomizing performance of prefilming and plain-jet airblast nozzles was compared by plotting some of the measured values of SMD along with the experimental data obtained by Rizkalla and Lefebvre [77] under comparable operating conditions. The results of such a comparison are presented in Fig. 6.32. It is clear that the plain-jet atomizer performs less satisfactorily than the prefilming type, especially under the adverse condition of low air/liquid ratio (ALR) and/or low air velocity.

The following dimensionally correct expression for the SMD was derived from an analysis of the experimental data:

$$\text{SMD} = 0.95 \frac{(\sigma \dot{m}_L)^{0.33}}{U_R \rho_L^{0.37} \rho_A^{0.30}} \left(1 + \frac{1}{\text{ALR}}\right)^{1.70} + 0.13 \left(\frac{\mu_L^2 d_o}{\sigma \rho_L}\right)^{0.5} \left(1 + \frac{1}{\text{ALR}}\right)^{1.70} \quad (6.54)$$

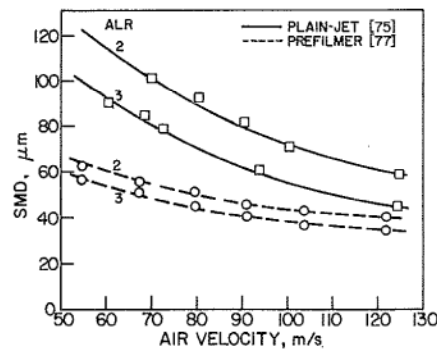


Figure 6.32 Comparison of atomizing performance of plain-jet and prefilming types of airblast atomizer (Lorenzetto and Lefebvre [75]).

Table 6.7 Test conditions employed in experimental studies

Type of atomizer	Investigators	Liquids employed	Liquid properties			Air properties			Air/liquid mass ratio	SMD $\times 10^6$	Method of measurement
			$\sigma \times 10^3$	ρ_L	$\mu_L \times 10^3$	$P_A \times 10^{-2}$	T_A	U_A			
Prefilmer	Lefebvre and Miller [80]	Water, kerosine	27.7-73.5	784-1000	1.0-1.29	1.0	295	167-122	3-9	42-90	Collected on coated slide
	Rizkalla and Lefebvre [77]	Water, kerosine, special solutions	24-73	780-1500	1.0-44	1.0-8.5	296-424	70-125	2-11	30-120	Light-scattering technique
	Jasuja [15]	Residual fuel oil	27-74	784-1000	1.0-8.6	1.0	295	55-135	1-8	30-140	Light-scattering technique
	El-Shanawany and Lefebvre [82]	Water, kerosine, special solutions	26-74	784-1000	1.0-44	1.0-8.5	295	60-190	0.5-5.0	25-125	Light-scattering technique
Spinning cup, prefilm	Fraser et al. [90]	Various oils	29-35	810-830	5-165	1.0	295	29-198	0.17-4.0	20-320	Light absorption
Plain jet	Nukiyama and Tanasawa [68]	Water, gasoline, alcohol, heavy fuel oil	19-73	700-1200	1.0-5.0	1.0	295	60-340	1-14	15-19	Coated slide
	Lorenzetto and Lefebvre [75]	Water, kerosine, special solutions	26-76	794-2180	1.0-76	1.0	295	60-180	1-16	20-130	Light-scattering technique
	Jasuja [78]	Residual fuel oils	21-74	784-1000	1.0-53	1.0	295	70-135	2-18	35-120	Light-scattering technique
	Rizk and Lefebvre [79]	Kerosine, gas oil, blended oils	27-29	780-840	1.3-3.0	1-7.7	298	10-120	2-8	15-110	Light-scattering technique

Air-assist	Wigg [64]	Water	73.5	1000	1.0	1.0	295	300-340	1-3	50-115	Immersion technique Collected on oil- coated slide Collection on coated slide Parker Hannifin spray analyzer Microscope counting
	Sakai et al. [70]	Water	73.5	1000	1.0	1.0	298		0.001-0.1	90-500	
	Inamura and Nagai [71]	Water	73.5	1000	1.0	1.0	298	0-300	1-10	35-140	
	Elkorts et al. [72]	Kerosine	30	800	3.35	1.0-6.0	295		0.5-2	20-65	
	Simmons [18]	Calibrating fluid MIL-F-70411	25	763	1.2	1.0	295		0.5-6	20-300	
Miscellaneous	Greizinger and Marshall [88]	Aqueous solutions	50		1.0	1.6-4	295	Up to sonic	1-16	5-30	High-speed photography Microscope counting Micromerograph and sifting Radiometer
	Ingebo and Foster [84]	Water, iso-octane, benzene, JP 5	16-71		0.45-1.93	0.4-1.64	298	30-210	5-100	25-80	
	Kim and Marshall [89]	Wax melts	30-50	800-960	1.0-5.0	0.76-2.0	298	75-393	0.06-40	10-160	
	Weiss and Worsham [87]	Wax melts	18-22	806-828	3.2-11.3	1.0-5.0	298	60-300	4-25	17-100	
	Ingebo [86]	Water	73.5	1000	1.0	1.0-21	295	45-220		50-200	

This expression was shown to be accurate to within 8% over the broad range of air and liquid properties indicated in Table 6.7.

Further studies at Cranfield on plain-jet atomization were made by Jasuja [78] using a single-nozzle configuration, as illustrated in Fig. 4.49. In this nozzle the liquid flows through a number of radially drilled plain circular holes, from which it emerges in the form of discrete jets that enter a swirling airstream. These jets then undergo in-flight disintegration without any further preparation such as pre-filming. Jasuja derived the following equation for drop size correlations:

$$\begin{aligned} \text{SMD} = 0.022 \left(\frac{\sigma}{\rho_A U_A^2} \right)^{0.45} \left(1 + \frac{1}{\text{ALR}} \right)^{0.5} \\ + 0.00143 \left(\frac{\mu_L^2}{\sigma \rho_L} \right)^{0.4} \left(1 + \frac{1}{\text{ALR}} \right)^{0.8} \end{aligned} \quad (6.55)$$

It should be noted that this equation contains no dimensions and is not dimensionally correct.

Rizk and Lefebvre [79] have also examined the effects of air and liquid properties and atomizer dimensions on mean drop size for the type of nozzle shown in Fig. 6.27. Tests were conducted using two geometrically similar atomizers having liquid orifice diameters of 0.55 and 0.75 mm. The liquids employed and the ranges of test conditions covered are shown in Table 6.7.

Figure 6.33, showing the effects of air velocity, air pressure, air/liquid ratio, and nozzle dimensions on SMD, is typical of the results obtained, all of which generally confirm the results of previous studies on airblast atomizers. They show that increases in air pressure, air velocity, and air/liquid ratio all tend to lower the mean drop size. An empirical, dimensionally correct equation for mean drop size was derived as follows:

$$\frac{\text{SMD}}{d_o} = 0.48 \left(\frac{\sigma}{\rho_A U_R^2 d_o} \right)^{0.4} \left(1 + \frac{1}{\text{ALR}} \right)^{0.4} + 0.15 \left(\frac{\mu_L^2}{\sigma \rho_L d_o} \right)^{0.5} \left(1 + \frac{1}{\text{ALR}} \right) \quad (6.56)$$

This equation was shown to provide an excellent data correlation, especially for low-viscosity liquids. Comparison with the results obtained by other workers is hindered by the different methods adopted for presenting the data. However, a comparison is provided for liquids of low viscosity, as illustrated in Table 6.8. The values for the exponents of air pressure, air velocity, and surface tension listed in this table are reasonably consistent with each other, but there are appreciable differences in regard to the effect of nozzle dimensions on SMD.

Prefilming

Plain-jet airblast atomizers are not very suitable for many combustion applications because they tend to concentrate the fuel drops within a small region just downstream of the nozzle. Much more preferable is the prefilming type of airblast atomizer, which can be designed to disperse the fuel drops throughout the entire combustion zone.

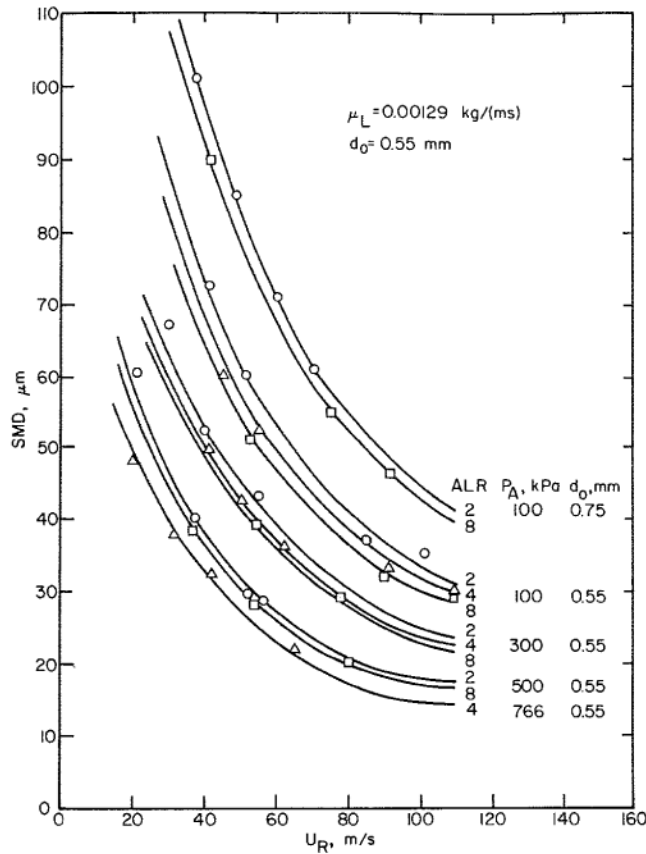


Figure 6.33 Variation of mean drop size with air pressure, air/liquid ratio, and fuel nozzle diameter for a plain-jet airblast atomizer (Rizk and Lefebvre [79]).

The prefiling concept for airblast atomization evolved from the experimental studies of Lefebvre and Miller in the early 1960s [80]. These workers carried out a large number of tests, using both water and kerosine, on several different atomizer configurations. Drop sizes were measured by collecting the drops on magnesium oxide-coated slides. Their experimental data generally confirmed the results obtained with other types of twin-fluid atomizer in regard to the important influence of air velocity on mean drop size. They also showed that variation in air/liquid ratio over the range from 3 to 9 had little effect on atomization quality. However, the main conclusion from this study was that "Minimum drop sizes are obtained by using atomizers designed to provide maximum physical contact between the air and the liquid. In particular it is important to arrange for the liquid sheet formed at the atomizing lip to be subjected to high velocity air on *both* sides. This also ensures that the drops remain airborne and avoids deposition of liquid on solid surfaces."

Table 6.8 Summary of data on effect of variables on mean drop size for liquids of low viscosity

Type of twin-fluid atomizer	Investigators	Power dependence on mean drop size of variable indicated						Dimensions
		Velocity U_a	Air density ρ_a	Liquid density ρ_L	Surface tension σ	Air/liquid ratio $(1 + \dot{m}_L/\dot{m}_A)$	Liquid/air ratio $(\dot{m}_L/\dot{m}_A)$	
Air assist	Wigg [64]	-1.0	-0.3	0	0.2	0.5	—	$R^{0.1}$
	Sakai et al. [70]	—	—	—	—	—	0.75	$d_o^{0.75}$
	Elkoth et al. [72]	-0.75	-0.39	-0.18	0.18	—	0.29	$d_o^{0.43}$
	Simmons [18]	-0.55	-0.375	.25	0.375	—	0.55	—
Plain jet	Nukiyama and Tanasawa [68]	-1.0	0	-0.5	0.5	—	0	0
	Lorenzetto and Lefebvre [75]	-1.0	-0.3	-0.37	0.33	1.70	—	$d_o^{0.55}$
	Jasuja [78]	-0.9	-0.45	0	0.45	0.5	—	$d_o^{0.6}$
	Rizk and Lefebvre [79]	-0.8	-0.4	0	0.4	0.4	—	$D_p^{0.5}$
Prefilmer	Rizkalla and Lefebvre [77]	-1.0	-1.0	0.5	0.5	1.0	—	$D_p^{0.5}$
	Jasuja [15]	-1.0	-1.0	0.5	0.5	0.5	—	0
	Lefebvre [91]	-1.0	-0.5	0	0.5	1.0	—	$D_p^{0.5}$
	El-Shanawany and Lefebvre [82]	-1.2	-0.7	0.1	0.6	1.0	—	$D_p^{0.4}$
Miscellaneous	Ingebo and Foster [84]	-0.75	-0.25	-0.25	0.25	0	0	$D_p^{0.5}$
	Weiss and Worsham [87]	-1.33	-0.30	—	—	—	—	$d_o^{0.16}$
	Gretzinger and Marshall [88]	-0.15	-0.15	0	0	—	—	$L^{-0.15}$
	Kim and Marshall [89]	-1.44	0.72	-0.16	0.41	0	0.6	0
	Fraser et al. [90]	—	-0.5	0	0.5	—	—	—

Lefebvre and Miller also stressed the importance of spreading the liquid into the thinnest possible sheet before subjecting it to airblast action, on the grounds that "Any increase in the thickness of the liquid sheet flowing over the atomizing lip will tend to increase the thickness of the ligaments which, upon disintegration, will then yield drops of larger size."

The main value of this study is not so much in the experimental data obtained, which are necessarily suspect due to the fairly primitive measuring techniques employed, but in its identification of the essential features of successful prefiling atomizer design.

Rizkalla and Lefebvre [77] used the light-scattering technique in a detailed investigation of prefiling airblast atomization. A cross-sectional drawing of the atomizer employed is shown in Fig. 6.34. In this design the liquid flows through six equispaced tangential ports and into a weir, from which it spills over the prefiling surface before being discharged at the atomizing lip. To subject both sides of the liquid to high-velocity air, two separate airflow paths are provided. One airstream flows through a central circular passage and is deflected radially outward by a pintle before striking the inner surface of the liquid sheet. The other airstream flows through an annular passage surrounding the main body of the atomizer. This passage has its minimum flow area in the plane of the atomizing lip, to impart a high velocity to the air where it meets the outer surface of the liquid sheet.

From inspection of all the data obtained on the effects of air and liquid properties on atomization quality [77], some of which are shown in Figs. 6.35 to 6.40, Rizkalla and Lefebvre drew certain general conclusions concerning the main factors governing mean drop size. For liquids of low viscosity, the key factors are surface tension, air velocity, and air density. From results obtained over a wide test range, it was concluded that liquid viscosity has an effect that is quite separate and independent from that of surface tension. This suggested a form of equation in which size is expressed as the sum of two terms—the first dominated by surface tension and air momentum and the second by liquid viscosity. Dimensional analysis was used to derive the following equation, the various constants and indices being deduced from the experimental data.

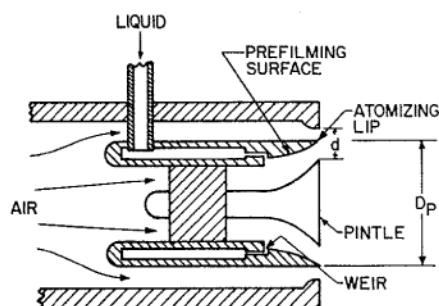


Figure 6.34 Prefiling type of airblast atomizer [77, 82]. Note that $D_h = 2d$.