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Progress Report

FLOW CHARACTERISTICS OF THE CARTER TYPE 120-166 CARBURETOR METERING JET

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ABSTRACT

An equation for predicting the flow rates through a Carter type 120-166 is presented. The equation was obtained by a dimensional analysis and considers variations in Reynolds number, ratio of jet diameter to metering rod diameter, and eccentricity of the metering rod. Accuracies of better than ± 3.5% are obtained for flow rates from 14 to 140 pounds per hour.

I. INTRODUCTION

In setting up an experimental test procedure to evaluate the effects of several variables on the flow through jets, it is desirable that some logical procedure be followed to obtain the maximum amount of information from the minimum number of experimental runs, and to organize this information in the most useful form.

A dimensional analysis of the flow problem related to small metering jets, selected as the most promising method, is developed in this report for the Carter type 120-166 main metering jet. The analysis is somewhat more general in scope than is indicated in its application to the specific type of jet used in this experimental program, since the analysis also considers variations in geometric ratios (i.e., ℓ/L and L/D) (see Fig. 1) that were necessarily constant for this jet. The results of the experimental test program are given in the form of a correlating equation which predicts flow rates to within $\pm$ 3% over the greater range of the flow rates tested (10-140 pph). Errors as high as $\pm$ 10.5% were encountered at low flow rates in the range from 10-14 pph and additional tests are being conducted to reduce this error and to gather data at flow rates below 10 pph.

II. DIMENSIONAL ANALYSIS

A dimensional analysis enables one to find dimensionless parameters which are particularly useful in experimentally evaluating the flow characteristics and in correlating the experimental data for fluid jets.

The complete procedure is carried out in the following steps:

1. The flow rate is assumed to be dependent upon certain selected physical variables.

2. The maximum number of dimensionless parameters (π-terms) formed with these variables is determined.

3. The dimensionless parameters themselves are obtained.

4. The form of the functional relationship among the several dimensionless parameters is assumed. This equation will contain several constants.

5. An examination of the dimensionless parameters is made with regard to limiting conditions of flow, with the aim of improving the parameters.

6. Experimental tests are conducted to determine the constants in the correlating flow-equation.

7. A re-evaluation of the dimensionless parameters may be necessary to improve the correlation of the experimental data.

8. The result is an equation which can be used to predict the effects of the considered physical variables on the flow rate through the jet or to determine the flow rate for a given set of variables. (See example at end of report.)

The actual mass flow rate (W) through the jet will be assumed to be dependent upon the following variables (step 1):

V = velocity in the jet as calculated by Eq. (4)

D = diameter of jet

d = diameter of rod

L = length of jet

ℓ = depth of chamfer

ϵ = displacement of rod from center of jet

ρ = mass-density of fluid

μ = viscosity of fluid.

The chamfer angle (θ) is assumed constant for this development and will not be considered in the dimensional analysis.

There exists some function (f) of the physical variables such that

$$f(W, V, D, d, L, \ell, \epsilon, \rho, \mu) = 0.$$

The dimensions of the variables in the $M\underline{L}T$ system of fundamental dimensions (M = mass, $\underline{L}$ = length, T = time) are:

$W = MT^{-1}$	$d = \underline{L}$	$\epsilon = \underline{L}$
$V = \underline{L}T^{-1}$	$L = \underline{L}$	$\rho = M\underline{L}^{-3}$
$D = \underline{L}$	$\ell = \underline{L}$	$\mu = M\underline{L}^{-1}T^{-1}$

The maximum number of independent π-terms (step 2) which can be used in correlating the experimental data is equal to the number of physical variables minus the rank of the dimensional matrix which follows:*

*Langhaar, H. L., Dimensional Analysis and Theory of Models, New York, John Wiley and Sons, Inc., 1951.

	W	V	D	d	L	ℓ	ϵ	ρ	μ
M	1							1	1
$\underline{L}$		1	1	1	1	1	1	-3	-1
T	-1	-1							-1

The numbers in the matrix represent the exponents of each fundamental dimension appearing in the dimension of each physical variable. The rank of the matrix is found to be 3, so that the maximum number of independent π-terms is

9(number of physical variables) - 3(rank of matrix) = 6.

We now solve for the three fundamental dimensions (M, $\underline{L}$, T) in terms of three variables containing all the fundamental dimensions. Choosing the set (ρ, V, D), we get

$$\left.\begin{aligned} \rho &= M\underline{L}^{-3} \\ D &= \underline{L} \\ V &= \underline{L}T^{-1} \end{aligned}\right\} \quad \text{so that} \quad \left\{\begin{aligned} M &= \rho\underline{L}^3 = \rho D^3 \\ \underline{L} &= D \\ T &= \underline{L}V^{-1} = DV^{-1} \end{aligned}\right.$$

Solving for the remaining six physical variables in terms of the selected set of three (ρ, V, D), we obtain the six π-terms

$$\begin{aligned}
&(1) \quad W = MT^{-1} = (\rho D^3)(D^{-1}V) = \rho D^2 V && \pi_1' = (W/\rho D^2 V) \\
&(2) \quad d = \underline{L} = D && \pi_2 = (d/D) \\
&(3) \quad L = \underline{L} = D && \pi_3 = (L/D) \\
&(4) \quad \ell = \underline{L} = D && \pi_4 = (\ell/D) \\
&(5) \quad \epsilon = \underline{L} = D && \pi_5 = (\epsilon/D) \\
&(6) \quad \mu = M\underline{L}^{-1}T^{-1} = (\rho D^3)(D^{-1})(D^{-1}V) = DV\rho && \pi_6 = (DV\rho/\mu)
\end{aligned}$$

π_1' is changed to the simpler form

$$\pi_1 = \pi_1' \cdot \pi_6 = (W/\rho D^2 V)\cdot(DV\rho/\mu) = (W/D\mu).$$

The solution can then be written as

$$f'\left[(W/D\mu),\ (DV\rho/\mu),\ (d/D),\ (L/D),\ (\ell/D),\ (\epsilon/D)\right] = 0,$$

where f' is used to indicate some functional relationship, as yet unknown, among the six π-terms. We can assume that the functional relationship can be expressed in the following manner (step 4):

$$(W/D\mu) = C(d/D)^a(L/D)^b(\ell/D)^c(\epsilon/D)^e(DV\rho/\mu)^f \qquad (1)$$

where C, a, b, c, e and f are constants which are to be evaluated from experimental data. This assumed form of the relationship will have to be checked as to its adequacy for correlating the experimental results. If the correlation is poor, revisions of the π-terms will be necessary (step 7) to improve the correlation.

According to step 5, each π-term is examined with regard to its "best" form for describing flow conditions.

$\pi_6 = (DV\rho/\mu)$; for noncircular cross-sectional areas, the Reynolds number $(DV\rho/\mu)$ is better expressed in terms of the "hydraulic diameter," or equivalent diameter, defined by

$$D_e \equiv 4 \times \frac{\text{flow cross-sectional area}}{\text{wetted perimeter}} .$$

For an annular area,

$$D_e = 4\,\frac{(\pi/4)(D^2-d^2)}{\pi(D+d)} = \frac{(D+d)(D-d)}{(D+d)}$$

$$D_e = (D-d). \qquad (2)$$

We therefore change π_6 to the new form

$$\boxed{\pi_6 = (D_e V\rho/\mu)} .$$

$\pi_1 = (W/D\mu)$; we shall assume that the hydraulic diameter is more significant than the jet diameter, so that

$$\boxed{\pi_1 = (W/D_e\mu)} .$$

$\pi_2 = (d/D)$; when d = 0, the π-term should reduce to 1. When d = D, the π-term should reduce to zero. Therefore choose

$$\boxed{\pi_2 = \left(1 - \frac{d}{D}\right) = \frac{D-d}{D} = (D_e/D)} .$$

$\pi_3 = (L/D)$; when L = 0, the π-term should reduce to 1. When $L = \infty$, the π-term should reduce to zero. Therefore choose

$$\boxed{\pi_3 = (e^{-L/D})} .$$

$\pi_4 = (\ell/D)$; it appears that a better relationship would be

$$\frac{\pi_4}{\pi_3} = \frac{\ell/D}{L/D} = (\ell/L).$$

When $\ell = 0$, the π-term should reduce to 1. When $\ell = L$, the π-term should reduce to some constant. Therefore choose

$$\boxed{\pi_4 = (e^{\ell/L})} ,$$

where e = 2.718 . . .

$\pi_5 = (\epsilon/D)$; when $\epsilon = 0$, the π-term should reduce to 1. When $\epsilon = \frac{1}{2}D_e$, the π-term should reduce to some constant. Therefore choose

$$\boxed{\pi_5 = \left(e^{\epsilon/D_e}\right)} .$$

The new form of the assumed correlating flow equation becomes

$$\boxed{(W/D_e\mu) = C(D_e/D)^a(e^{-L/D})^b(e^{\ell/L})^c(e^{\epsilon/D_e})^e(D_eV\rho/\mu)^f} \qquad (3)$$

where the velocity is to be computed from the following equation:

$$\boxed{V = \sqrt{2g_c\left(\frac{P_1-P_2}{\rho}\right)} \quad \sqrt{\frac{1}{1-(A_2/A_1)^2}}} \qquad (4)$$

P = pressure

ρ = mass density

g_c = dimensional constant

$= 32.174 \frac{\text{lbm}}{\text{lbf}} \frac{\text{ft}}{\text{sec}^2}$ or equivalent.

III. EXPERIMENTAL TEST PROGRAM

The evaluation of the constants in Eq. (3) constitutes step 6. If all π-terms are held constant except the flow parameter $Y_1 \equiv (W/D_e\mu)$ and Reynolds number, and the results are plotted as log Y_1 vs. log Re, the slope of the curve will be equal to f, the exponent of Reynolds number in Eq. (3). Similar plots on

log-log coordinates must be made for Y_1 vs. π_2, Y_1 vs. π_3, Y_1 vs. π_4, and Y_1 vs. π_5 to evaluate exponents a, b, c, and e. The value of the coefficient C is the slope of the curve showing Y_1 vs. $\pi_2^a \pi_3^b \pi_4^c \pi_5^e \pi_6^f$ on rectangular coordinates. The resulting plots must form straight lines in each case; otherwise the values for the exponents and coefficient are not constant as assumed. A revision in the π-term, as it appears in Eq. (3), may then be necessary.

The resulting correlating equation obtained from this test program will permit the prediction of actual flow rates when all physical factors are specified. A comparison of actual to theoretical flow rate will give the discharge coefficient for the jet.

The tests described in this report were carried out with the equipment illustrated in Figs. 2, 3, 4, and 5. The plastic test section shown in Fig. 4 was used to determine the effect of rod eccentricity on flow rates. The brass test section, shown in Fig. 5, with various-diameter downstream-tube sections was used for all other tests.

IV. EXPERIMENTAL RESULTS

A. EFFECT OF ROD DIAMETER AND ECCENTRICITY

All results given in this report are for the Carter type 120-166 main metering jet. Therefore, all terms in Eq. (3) involving changes in jet geometry were constant for these tests and could be lumped with the coefficient C. The equation reduces to

$$(W/D_e\mu) = C(D_e/D)^a(e^{\epsilon/D_e})^e(D_eV\rho/\mu)^f. \qquad (5)$$

Figure 6 shows the results of the first series of tests, taken with the plastic test section shown in Fig. 4, with rods of different size and different eccentricities. It can be seen that the effect of eccentricity is very slight. When the eccentricity is small, there is practically no change in the pressure drop across the jet for a constant flow rate. However, as the rod nears the wall of the jet, there is a sudden, although small, increase in the pressure drop required to maintain constant flow. The greatest pressure differential for constant flow rate occurs with the rod resting against the jet wall.

The resulting data did not exhibit a smooth trend in flow rate as the eccentricity was varied, and the maximum effect on flow was small. Because of this and the fact that the jet normally operates with the rod held against its side, the effect of eccentricity was neglected in all further work described in this report. However, with refinements in the apparatus, this effect may again be investigated.

All additional tests were conducted with the brass test section shown in Fig. 5 to overcome the difficulties with leaks experienced with the plastic setup.

Figure 7 shows the effects of both changes in Reynolds number and rod diameter on the flow rate. Reynolds numbers were varied from about 500 to 15,000 (the highest values do not appear on the figure) and rod diameters of 0.031, 0.058, 0.065, and 0.072 in. were used.

The slopes of the lines in Fig. 7 vary from 1.026 to 1.078. Since no smooth trend between slope and rod diameter is indicated, an average value of 1.05 was selected as the value for the exponent f on the Reynolds number.

At this point, neglecting the effect of eccentricity, the correlating equation is given by

$$(W/D_e\mu) = C(D_e/D)^a(D_eV\rho/\mu)^{1.05}.$$

The fact that the lines in Fig. 7 are nearly parallel indicates the values of Y_1 for one of the upper lines are greater than the values for the lowest line by a constant multiplier. The vertical distance between the lines is proportional to the logarithm of the multiplier. Starting with the lowest line, measurements from Fig. 7 give values of 1, 1.93, 3.95, 5.22, and 7.17 for the multipliers. When these values are plotted against (D/D_e), the resulting curve is a straight line which intersects the x-axis at $(D/D_e) = 0.46$ and which has a slope of 1.847. These results indicate a correlating equation of the form

$$(W/D_e\mu) = C\left[1.847\ (D/D_e - 0.46)\right](D_eV\rho/\mu)^{1.05}.$$

The constant 1.847 can be included in the coefficient C.

If the procedure originally outlined for finding the exponent a is followed, and $\log\left[Y_1/(D_eV\rho/\mu)^{1.05}\right]$ is plotted vs. $\log(D_e/D)$, the resulting diagram shows a great deal of scatter, indicating poor correlation of the experimental results by the $(D_e/D)^a$ term. Therefore the term $(D/D_e - 0.46)$, appearing in the above equation, is selected for use in the correlating equation.

The value of the constant (1.847 C) appearing in this equation is determined from Fig. 8 where Y_1 is plotted vs. $(D/D_e - 0.46)Re^{1.05}$. The constant is equal to the slope of the curve.

The resulting correlating equation is given by

$$\boxed{(W/D_e\mu) = 0.8\underline{6}\left[(D/D_e) - 0.46\right](D_eV\rho/\mu)^{1.05}.} \qquad (6)$$

Calculated flow rates obtained from this equation are in error by only ± 3% for better than 95% of the experimental runs between 14 and 140 pph. The other 5% of the runs in this flow range show errors of less than 3.5%. The errors in the range from 10-14 pph were considerably higher, a maximum of 10.5% being recorded. It is felt that the poor results in the low flow ranges were due to errors in reading pressures with ordinary manometers. These have been replaced with inclined manometers, shown in Fig. 2, and additional data for low flow rates are being gathered.

B. EFFECT OF DOWNSTREAM-TUBE SIZE

To clarify the significance of the measured downstream pressure, tests were conducted with downstream sections, shown in Fig. 5, having diameters of 3/16, 3/4, and 1-5/8 in. Several pressure taps were located at various distances downstream from the jet, starting with a position at the surface of the plate in which the jet is mounted. All downstream pressures were indicated simultaneously on the multiple manometer board shown in Fig. 9.

Downstream pressure measurements made with the 1-5/8 in.-diameter tube showed very little pressure variation among the various pressure taps. Measurements with the 3/4-in. tube showed some variation, while a very pronounced change in pressure, especially between the first flow-pressure taps, occurred with the 3/8-in. tube. The latter case is shwon in Fig. 9. In all cases, for a given flow rate, the downstream pressures measured at the first tap were approximately the same. Therefore, this pressure tap was used for pressure measurements in all flow tests.

The rapid increase in downstream pressure with the smallest tube is due to the large shearing stresses imposed on the jet stream by the neighboring tube wall. This shearing action causes a rapid conversion of velocity head into pressure head. With the large tube, a large eddy exists between the issuing jet and the tube wall so that the shearing stresses at the jet surface is greatly reduced. Therefore the rate of conversion from velocity head to pressure head is reduced considerably. At the first pressure tap, the viscous action has had little chance to come into play, so that the pressure measurements at this point made with the various tube sections are about equal.

C. JET IN CARBURETOR CASTING

Another series of tests was performed to obtain information on the added pressure drop in the fuel-flow channel. The measured pressure drop occurred across the main jet, without rod, and that portion of the channel in the main casting not including the venturi cluster. The setup is shown in Fig. 5.

The results are plotted in Fig. 10 as discharge coefficient vs. pressure drop, where the discharge coefficient is defined as

$$C_D = \frac{\text{actual flow rate}}{\text{calculated flow rate}} = \frac{W}{\rho A V},$$

and A is the cross-sectional area of flow. The discharge

coefficients for jets without rods, obtained from tests with all three downstream sections, are plotted for comparison. The results indicate the large reduction in flow, for a given pressure drop, caused by the flow channel in the main body casting.

D. EFFECT OF SCREWDRIVER SLOT

To determine the effect of the screwdriver slot on jet characteristics, the slot in one of the jets was filled with a hard material and then scraped to fit the jet contour. Results from this test indicated an appreciable drop in pressure (approximately 7%) required for a given flow rate. However, the results are not conclusive, since the surface of the jet was scratched when removing the filler material. This test will be repeated soon to check the earlier results, and these results will be submitted at a later date. The orientation of this slot in the carburetor casting may also affect the flow rate and will be investigated.

V. APPLICATION OF CORRELATING EQUATION

Equation (6) can be reduced to a more usable form by expressing velocity (V) in terms of pressure drop across the jet. The velocity is obtained from Eq. (4), where the area term under the second radical can usually be neglected. Equation (6) then becomes

$$\boxed{\left(\frac{W}{D_e\mu}\right) = 1.238\left[\left(\frac{D}{D_e}\right) - 0.46\right]\left[\frac{D_e}{\mu}\sqrt{g_c\rho(\Delta P)}\right]^{1.05}} \qquad (7)$$

<u>Example</u>:

Given: Jet, type 120-166 — $D = 0.0935$ in.
Rod — $d = 0.065$ in.
Fluid (Stanisol) — gravity, $^{o}API = 48.9$; viscosity ($80^{o}F$) $\nu = 1.1$ centistokes
Pressure drop — $\Delta h = 24.64$ in. fluid

Find: The predicted flow rate (W) in pounds per hour (pph) and compare with the measured value.

(a) $D_e = (D-d) = (0.0935 - 0.065)/12 = 2.375 \times 10^{-3}$ ft

(b) $D/D_e = 0.0935/0.0285 = 3.28$

(c) $\rho = S.G. \times 62.4 = \left[141.5/(131.5 + 48.9)\right] 62.4 = 48.9$ lbm/ft^3

(d) $\mu = \rho\nu = (48.9)(1.1 \times 10^{-2})(.001078) = 58 \times 10^{-5}$ lbm/ft sec

(e) $\Delta P = (g/g_c)\rho(\Delta h) = (48.9/12)\Delta h = 40.7(\Delta h)$ lbf/ft^2

(f) $g_c = 32.174 \frac{\text{lbm}}{\text{lbf}} \frac{\text{ft}}{\text{sec}^2}$.

Substituting in the above equation, we obtain

$$W = 21.02\ (\Delta h)^{.525} \times 10^{-4} \qquad \text{(lbm/sec)}$$

or $$W = 7.57\ (\Delta h)^{.525} \qquad \text{(lbm/hr)}.$$

Selecting a pressure drop of 24.64 in., which was obtained for one of the experimental test runs, we compute the flow rate to be

$$W = 40.75\ \text{(lbm/hr)}.$$

The actual flow rate for these conditions was measured and found to be 40.8 lbm/hr. The percent error in this case is

$$\text{Error} = \frac{40.8 - 40.75}{40.8} \times 100 \approx 0.1\%.$$