Annular Flow Pressure Drop Model for Pease–Anthony-Type Venturi Scrubbers

An annular flow pressure drop model has been developed and compared with experimental data obtained on a pilot-plant-size Pease–Anthony-type venturi scrubber. Droplet and film accelerations as well as wall friction, based on the two-phase Lockhart-Martinelli correlation, are considered. Excellent agreement with experimental data is demonstrated for a wide range of throat gas velocities, liquid to gas ratios, and film flow rates for a single scrubber geometry.

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SCOPE

Venturi scrubbers have been recognized widely for their high particulate matter collection efficiencies. Fine-particle collection generally requires elevated pressure drops. The primary objective of this study was to develop a realistic mathematical model for prediction of pressure drop in a Pease-Anthony venturi scrubber.

Flow losses are predicted by accounting for frictional effects and acceleration of liquid drops and the liquid films flowing on the walls. Recovery in the diffuser is also considered. Model validation involved measurement of pressure gradients, film flow rates, liquid to gas ratios, and throat gas velocities for a pilot-plant-scale venturi. A comparison is made among three widely used correlations using experimental data taken from the pilot-plant-scale venturi scrubber.

CONCLUSIONS AND SIGNIFICANCE

An annular flow model is developed for accurate prediction of pressure drops in Pease-Anthony venturi scrubbers. This model—which considers the primary design parameters of liquid to gas ratio, throat gas velocity, venturi geometry, and liquid film flow rate—accurately predicted the measured pressure gradients and overall energy losses.

The Hesketh (1974) correlation underestimated pressure drops at all liquid to gas ratios and throat gas velocities tested. The Calvert modified model (Yung et al., 1977a) predicted overall pressure drops lower than those experimentally measured for liquid to gas ratios below 8.0×10^{-4} m^3 liquid/m^3 air, and greater magnitudes for liquid to gas ratios exceeding 1.3×10^{-3}. The region of good agreement was achieved in spite of the neglect of wall friction, converging section losses, and diffuser pressure recovery when these effects compensated each other.

Boll's model (1973) consistently overpredicted the experimental pressure drops. The deviation increased with increasing liquid to gas ratios. Since film flow was not considered, the droplet acceleration component was always overestimated. The frictional losses, which were predicted using a homogeneous model (Wallis, 1969), were overestimated.

INTRODUCTION

Control of particulate matter emissions in waste gas streams from industrial processes has been an important concern facing industrialized countries for several decades. Popular control equipment choices for the effective removal of particulate matter from moving gas streams include units operating on the basis of cyclonic, inertial, or electrostatic principles. One of the most widely used devices in the inertial separator category is the venturi scrubber.

The venturi scrubber is designed to atomize scrubbing liquid introduced into a throat (high-velocity) region under cross, co-current, or countercurrent flow conditions to provide a spectrum of droplets that remove particulate matter primarily through

TABLE 1. PRESSURE DROP MODELS FOR VENTURI SCRUBBERS

Investigator	Correlation	Type of Correlation
Yoshida et al. (1960,1965)	$\Delta P = \frac{\rho_G V_{G,th}^2}{2g_c}[a + b(\mathrm{L/G})]$ $a = \frac{\mathcal{J}_{dif}}{2 \tan \beta/2} + 4\mathcal{J}_{th}\frac{\rho_{th}}{D_{th}} + \nu\left(1 - 2\frac{D_{th}^2}{D_e^2}\right)$ $b = (\xi_{th} - 1)\frac{4\mathcal{J}_{th} l_{th}}{\rho_m D_{th}} + (\xi_{dif} - 1)\frac{\nu}{\rho_m}\left(1 - \frac{D_{th}^2}{D_e^2}\right)$	Theoretical
Yamauchi et al. (1964)	$\Delta P = 3.0(\Delta T)^{-0.28}\frac{\rho_G V_{G,th}^2}{2g_c}[1 + (\mathrm{L/G})]$	Experimental
Volgin et al. (1968)	$\Delta P = 6.32\, V_{G,th}^2 (\mathrm{L/G})^{0.26} (l_{th})^{0.143}$	Experimental
Calvert (1970)	$\Delta P = 1.01 \times 10^3\, V_{G,th}^2 (\mathrm{L/G})$	Theoretical
Boll (1973)	$-\frac{\Delta P}{\rho_G} = \frac{V_{G,i}^2 - V_G^2}{2g_c} + \frac{m}{g_c}\int_0^t V_G\left(\frac{dV_d}{dt}\right)dt$ $+ \frac{(m+1)}{2g_c}\int_0^t \frac{f V_G^2 V_d}{D_e}dt + \frac{1}{2g_c}\int_{Z_i}^{Z_{inf}}\frac{f V_G^2}{D_e}dZ$	Theoretical
Behie and Beeckmans (1973)	$dP = -\left(\frac{6F_d}{\pi D_d^3 \rho_d}\right)\left[\frac{1}{2}C_D \rho_G (V_G - V_d)^2 \frac{\pi D_d^2}{4}\right]dt$	Theoretical
Hesketh (1974)	$\Delta P = 0.456\, V_{G,th}^2 \rho_G A_{th}^{0.133} \times [0.56 + 935(\mathrm{L/G}) + 1.29 \times 10^{-5}(\mathrm{L/G})^2]$	Experimental
Yung et al. (1977a)	$\Delta P = -\frac{2\rho_L V_{G,th}^2}{g_c}(\mathrm{L/G})[1 - y^2 + \sqrt{(y^4 - y^2)}]$ $y = \frac{3 l_{th} C_{c,thi} \rho_G}{16 D_d \rho_L} + 1$	Theoretical

inertial impaction. Venturi scrubbers are popular because they offer significant advantages:

- Lower initial costs for comparable collection efficiencies
- Small floor area requirements
- No internal moving parts
- Capabilities for safe handling of wet, corrosive gases.

The high power requirements for fine particulate removal represent their main drawback.

Liquid drop acceleration by the gas, irreversible drag force work, and wall losses define the magnitude of the pressure drop. Experiments with transparent venturi scrubbers have demonstrated the existence of an annular, two-phase, two-component flow composed of a thin liquid layer on the walls and a high-velocity gas core carrying entrained droplets. Available pressure drop models are not applicable over a wide range of liquid and gas flow rates due to restrictive assumptions regarding the two-phase, two-component flow in typical units. The objective of this work was to develop a more realistic model for the estimation of pressure drop over the range of venturi scrubber operations encountered in industrial applications.

PREVIOUS WORK

Several experimental and theoretical models concerned with the pediction of pressure drop in venturi scrubbers are summarized in Table 1. The initial version of the Calvert model (1970) was derived from Newton's law in terms of the force required to accelerate all of the liquid to the gas throat velocity. Wall friction and momentum recovery in the divergent section were neglected. Although this simplistic model has been used widely for pressure-drop prediction, researchers have continued to seek improvements because accurate pressure-drop predictions are limited to liquid to gas ratios between 0.8 to 1.3×10^{-3} m^3 liquid/m^3 air. Hesketh's (1974) correlation was developed from a critical examination of the performance of fixed-throat industrial scrubbers with liquid injection upstream of the throat.

Boll (1973) developed a mathematical model based on the simultaneous solution of the equations of drop motion and momentum exchange for ducts of variable cross section. His model provided better agreement with experimental data except for low ($<6.7 \times 10^{-4}$) and high ($>1.2 \times 10^{-3}$) L/G ratios where the measured pressure drops were underpredicted and overestimated, respectively. These deviations in pressure-drop prediction

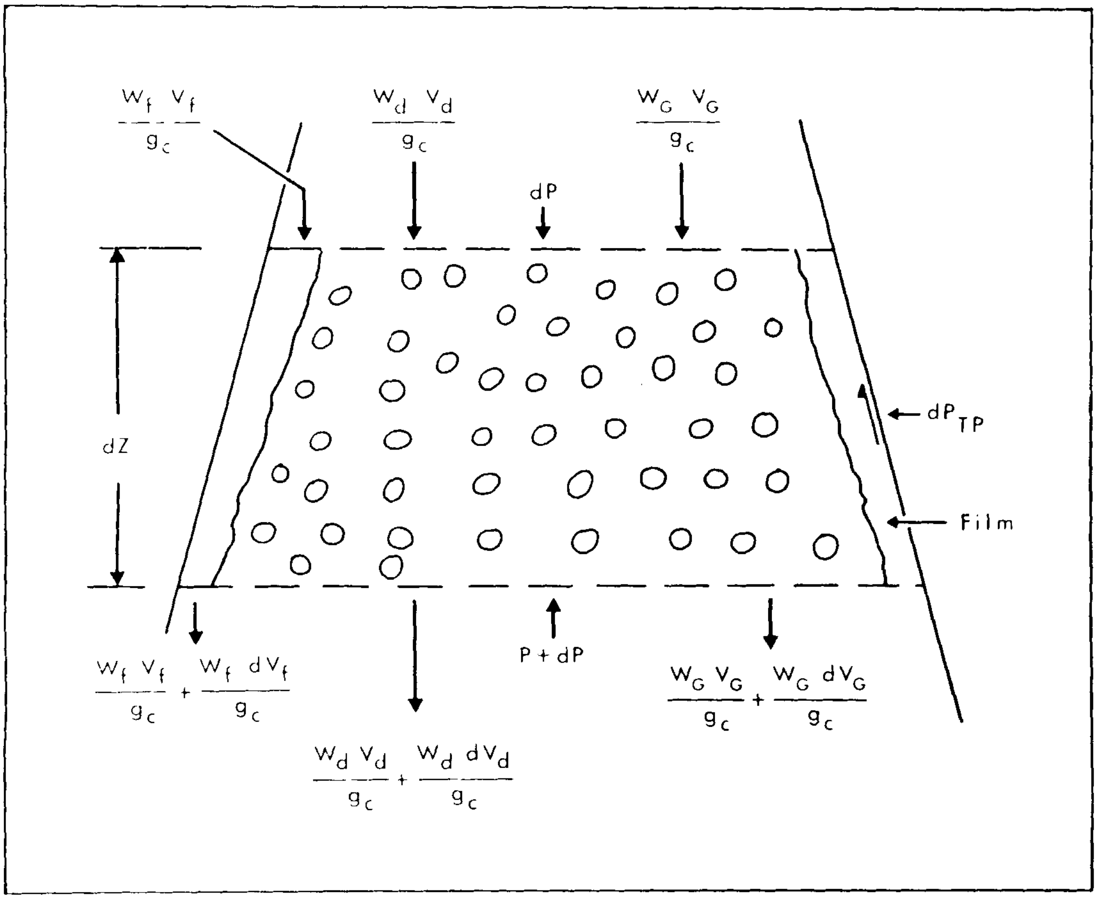

Figure 1. Force balance on a differential element.

can be attributed to the neglect of realistic multiphase losses, the existence of a liquid film flowing on the walls, and nonuniform distribution of scrubbing water. The model of Yung et al. (1977a) modified Calvert's equation by accounting for the fact that liquid drops are not accelerated to gas throat velocities. Wall friction and pressure recovery in the diffuser, however, remained neglected. Since wall frictional losses are more significant than pressure recovery in the diffuser at low liquid to gas ratios, this model was found to underestimate the experimental pressure drop. The reverse was found to occur at high liquid to gas ratios.

THEORETICAL DEVELOPMENT

The separated flow model incorporates the following assumptions:

- The existence of a homogeneous core fluid flowing cocurrently with a liquid film on the venturi walls.
- Constant, but not necessarily equal, velocities for the homogeneous core and liquid film across the cross-sectional flow area.
- Constant physical properties over the differential control volume.
- Attainment of thermodynamic equilibrium between phases.
- Drop formation in the core producing one uniform size.
- Equal static pressures for both the core and liquid film.
- Unchanged flow patterns along the entire scrubber length.

Figure 1 defines the control volume and related variables for a differential length of the venturi. The core quality X, defined in terms of mass flow rates, is:

$$X = \frac{W_G}{W_C} = \frac{W_G}{W_d + W_G} \tag{1}$$

The film flow rate and entrainment can be related to a core entrainment factor, C, and total liquid flow rate as:

$$W_f = (1 - C)W_L \tag{2}$$

$$W_d = C\,W_L \tag{3}$$

$$W_L = m\,W_G = W_d + W_f \tag{4}$$

If A_T represents the cross-sectional flow area of the venturi and A_c represents the core cross-sectional area, then the fractional area occupied by the core, α_c, is given by:

$$\alpha_c = \frac{A_c}{A_T} \tag{5}$$

The volume fraction of the core occupied by the gas, α_G, equals the ratio of the volumetric gas flow rate to the total volumetric flow rate in the core assuming homogeneous flow:

$$\alpha_G = \frac{Q_G}{Q_G + Q_d} \tag{6}$$

Neglecting gravitational forces, the one-dimensional steady-state macroscopic balance for a differential length, dZ, of the scrubber cross section shown in Figure 1 yields:

$$A_T dP + \frac{W_G\,dV_G}{g_c} + \frac{W_d\,dV_d}{g_c} + \frac{W_f\,dV_f}{g_c} + A_T\,dP_{TP} = 0 \tag{7}$$

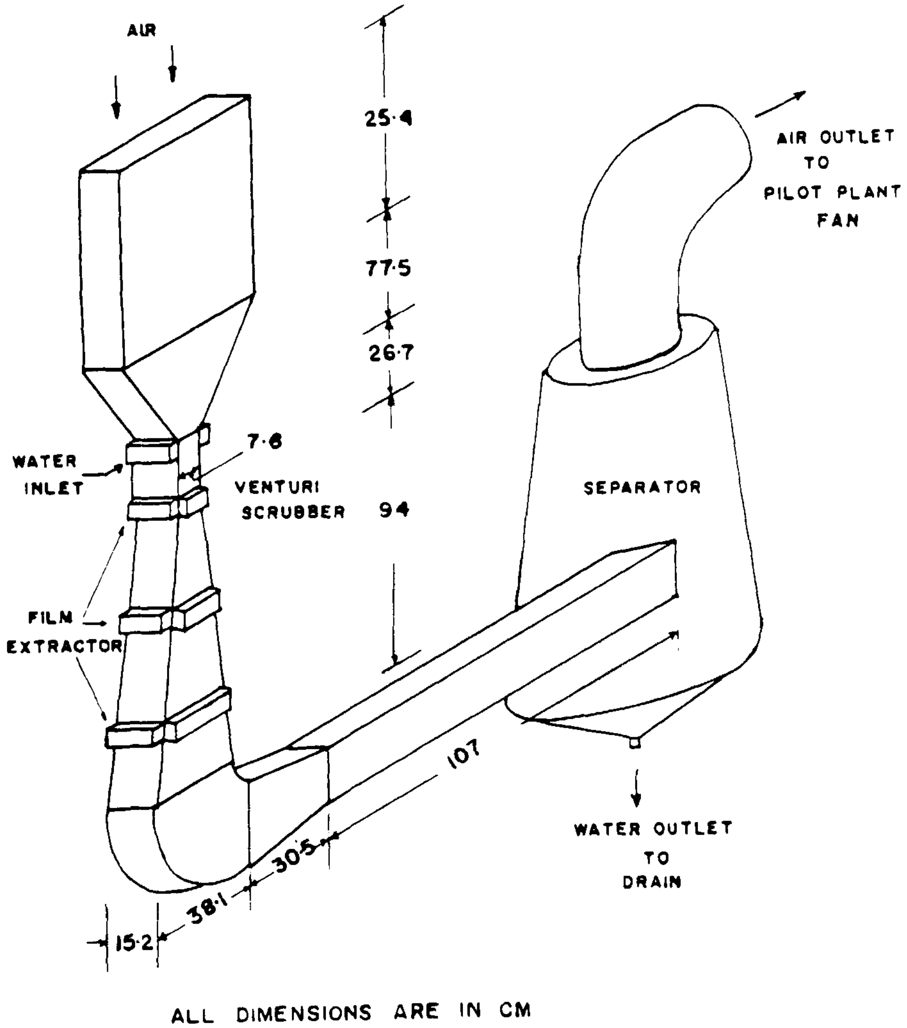

Figure 2. Isometric view of experimental venturi scrubber-separator system.

The two-phase frictional pressure drop, dP_{TP}, can be expressed in terms of the homogeneous pressure gradient as:

$$dP_{TP} = (dP/dZ)_c \, \phi_c^2 \, dZ \tag{8}$$

The core frictional pressure gradient, $(dP/dZ)_c$ is that for a fictitious single-phase fluid of density ρ_c, given by:

$$\frac{1}{\rho_c} = \frac{X}{\rho_G} + \frac{(1-X)}{\rho_d} \tag{9}$$

and viscosity μ_c, given by:

$$\frac{1}{\mu_c} = \frac{X}{\mu_G} + \frac{(1-X)}{\mu_d} \tag{10}$$

On this basis,

$$\left(\frac{dP}{dZ}\right)_c = \frac{2}{g_c}\frac{\rho_c}{D_e}\left(\frac{W_c}{A_T\,\rho_c}\right)^2 f_c \tag{11}$$

The drop acceleration, obtained from a force balance (Boll, 1973) is:

$$\frac{d\,V_d}{dt} = \frac{3}{4}\frac{\mu_G}{\rho_d}\frac{(V_G - V_d)}{D_d^2}C_{DN} \tag{12}$$

Droplet diameter is estimated from the correlation of Nukiyama and Tanasawa (1938).

The mass flow rates, W_G, W_d, and W_f are expressed in terms of the core area A_c and core void fraction α_G according to:

$$W_G = \alpha_G\,A_c\,V_G\,\rho_G \tag{13}$$

$$W_d = C\,m\,\alpha_G\,A_c\,V_G\,\rho_G \tag{14}$$

$$W_f = (1-C)m\,\alpha_G\,A_c\,V_G\,\rho_G \tag{15}$$

Substitution of Eqs. 11, 13, 14, and 15 into Eq. 7 yields, after simplification:

$$-\frac{dP}{\rho_G} = \left[\frac{\alpha_G\,\alpha_c\,V_G\,d\,V_G}{g_c}\right] + \left[\frac{C\,m\,\alpha_G\,\alpha_c\,V_G\,d\,V_G}{g_c}\right] + \left[\frac{(1-C)m\,\alpha_c\,\alpha_G\,V_G\,d\,V_f}{g_c}\right] + \left[\frac{2\,f_C\,W_c^2\,\phi_c^2\,dZ}{g_c\,D_e\,\rho_c\,A_t^2\,\rho_G}\right] \tag{16}$$

In order to obtain the total pressure drop, model Eqs. 12 and 16 are integrated numerically. The annular void fraction; α_c, is evaluated from the correlation (Hewitt and Taylor, 1970):

$$\alpha_c = [1 + \chi^{0.8}]^{-0.378} \tag{17}$$

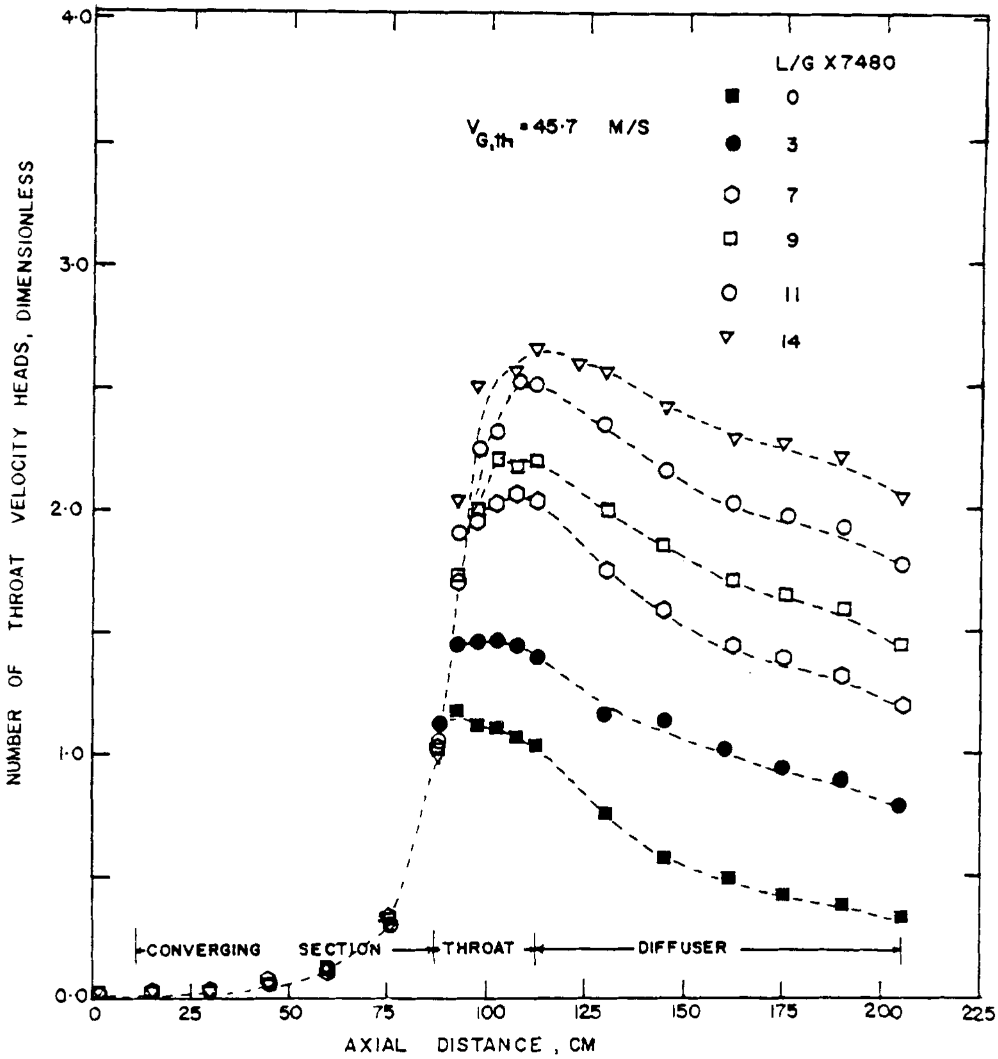

Figure 3. Axial pressure drop profiles for $V_{G,th}$ = 45.7 m/s.

where

$$\chi = \text{Martinelli parameter} = \sqrt{\frac{\left(\frac{dP}{dZ}\right)_f}{\left(\frac{dP}{dZ}\right)_c}}$$

The pressure gradients are obtained by assuming that the core fluid and liquid film are flowing independently in the scrubber. The two-phase friction multiplier, ϕ_c^2, can be calculated using any one of several available correlations.

Wallis (1969) proposed the relationship:

$$\phi_c^2 = [1 + 75\,(1 - \alpha_c)]/\alpha_c^{2.5} \tag{18}$$

assuming no entrainment and ignoring interfacial velocity.

According to Lockhart and Martinelli (1949):

$$\begin{aligned} \phi_c^2 &= 1 + 12\,\chi + \chi^2 \text{ for } N_{Re,f} < 2{,}000 \\ \phi_c^2 &= 1 + 20\,\chi + \chi^2 \text{ for } N_{Re,f} > 2{,}000 \end{aligned} \tag{19}$$

Collier (1970) incorporated entrainment and interfacial velocity to obtain

$$\phi_c^2 = \left[\frac{1 + 75(1 - \alpha_c)}{\alpha_c^{2.5}}\right]\left[\frac{W_G + W_d}{W_G}\right] \times \left[1 - 2\left(\frac{\alpha_c}{1 - \alpha_c}\right)\left(\frac{W_f}{W_G}\right)\left(\frac{\rho_G}{\rho_d}\right)\right]^2 \tag{20}$$

The gas and film velocities can be computed from:

$$V_G = \frac{W_G}{\rho_G\, \alpha_c\, \alpha_G\, A_T} \tag{21}$$

and

$$V_f = \frac{W_f}{\rho_f(1 - \alpha_c)A_T} \tag{22}$$

EXPERIMENTAL WORK

Experiments were conducted with a pilot-plant-size Pease–Anthony-type venturi scrubber, fabricated from 9.5 mm Plexiglas sheets, providing a cross section with the dimensions shown in Figure 2. The water flow

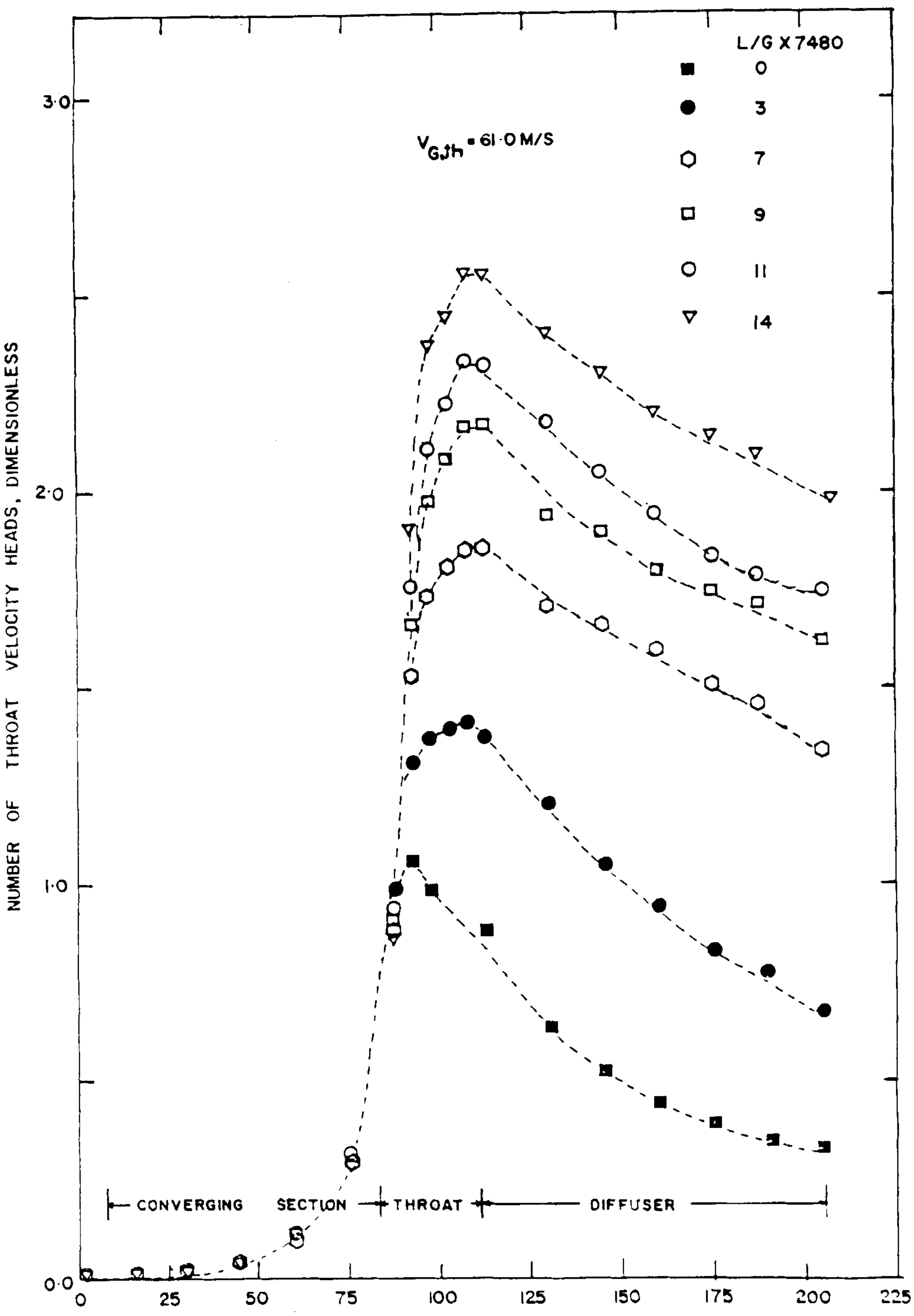

Figure 4. Axial pressure drop profiles for $V_{G,th}$ = 61.0 m/s.

rates to the manifolds on each side of the throat section were metered by a matched pair of rotameters. Scrubbing liquid was distributed through seventeen 2.1 mm dia. orifices located in two staggered rows on each side of the throat. Gas flow rates were varied by means of a damper located on the discharge side of the blower. Film flow rates on the constant and variable dimension sides were determined at three axial locations. The liquid film was extracted through stainless steel sintered plates using a vacuum sampling system. In addition, the static pressure was measured at 18 different locations along the venturi axis. Data were collected for liquid to gas ratios varying between 4.0×10^{-4} and 1.9×10^{-3} m^3 liquid/m^3 air and throat gas velocities ranging between 45.7 and 76.2 m/s.

RESULTS AND DISCUSSION

Figures 3, 4 and 5 illustrate the variation in axial pressure drop for five liquid to gas ratios and three throat gas velocities. At each fixed-throat gas velocity, the pressure drop rises exponentially up

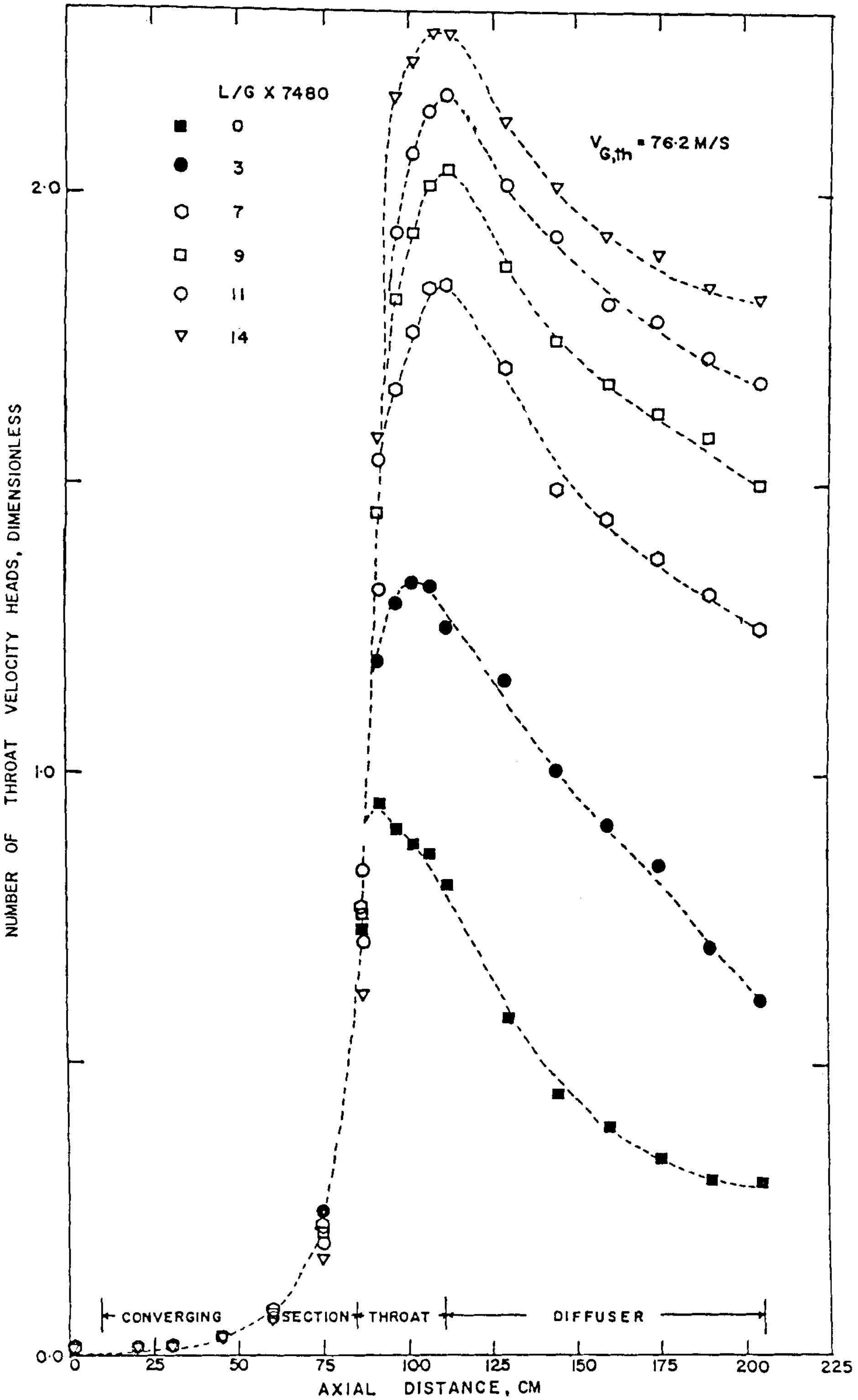

Figure 5. Axial pressure drop profiles for $V_{G,th}$ = 76.2 m/s.

to the throat where the gradient becomes essentially linear. Significant pressure recovery results in the diffuser. The single-phase profiles showed that pressure recovery began in the throat. This was due to the possible formation of a "vena contracta" in the throat region which was eliminated by liquid addition.

Comparison of Pressure Drop Models

The three more widely-used models (Hesketh, 1974; Boll, 1973; Yung et al., 1977a) are compared with the proposed method. The Milne fourth-order predictor-Hamming corrector nu-

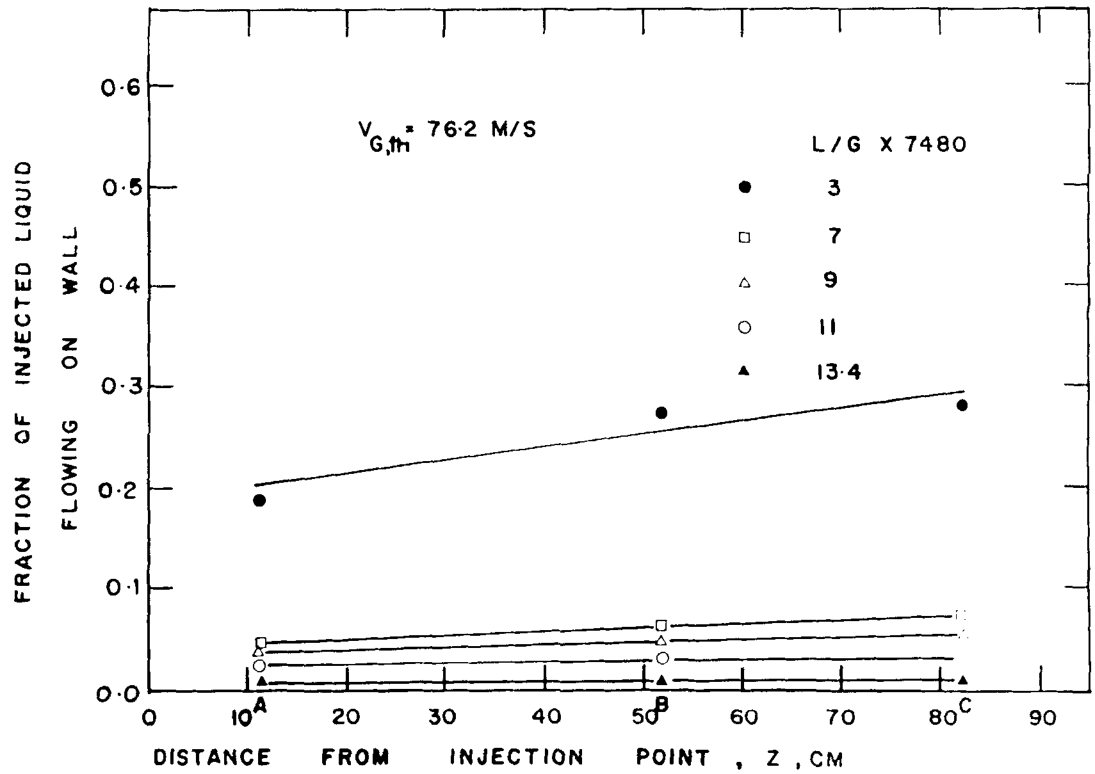

Figure 6. Variation of film flow over scrubber length.

merical procedure (James et al., 1977) was used to integrate the differential equations. Values of C_{DN} were obtained from a series of linear approximations that reproduced the standard drag curve within 1% over the range of $0 \leq N_{Re} \leq 1{,}000$. The predicted pressure drops were expressed in terms of throat velocity heads, N_{vh}, for comparison with experimental correlations of data.

In the diffuser, the fluid velocities are reduced (and the absolute pressure increases) under conditions that provide minimal interaction of the gas and entrained liquid. For this reason, the pressure recovery performance of a two-phase flow diffuser is expected to be less than that obtained for single-phase flows. The diffuser effectiveness for single-phase air flows was evaluated by a trial-and-error procedure that showed an energy recovery of 80% and provided excellent agreement with experimental pressure gradients. Pressure recovery for gas flows under two-phase flow conditions was assumed to be the same as that determined from single-phase pressure drop measurements. The liquid pressure recovery was arbitrarily assumed to be 100% efficient for the annular flow model used here.

Annular Flow Model

Wall film flow rate measurements that were obtained simultaneously with pressure drop data are shown in Figure 6. Overall pressure drops predicted using three different two-phase flow multipliers were compared with experimental results. The Lockhart-Martinelli correlation provided the best agreement with experimental values at all liquid to gas ratios and throat gas velocities. Figure 7 demonstrates the excellent agreement between measured pressure losses and values predicted with the annular flow model.

Figures 8 to 10 compare the overall pressure drops predicted by Hesketh's (1974) experimental correlation, the modified Calvert (Yung et al., 1977a), Boll (1973), and annular flow models with experimental values obtained during this study. Although there are other publications (Yoshida et al., 1960; Yamauchi et al., 1964; Volgin et al., 1968) reporting experimental pressure drop data for venturi scrubbers, most are of limited value for model validation because significant parameters such as pressure gradients and scrubber dimensions are not provided.

Hesketh's Correlation

Figures 8 to 10 show that the Hesketh correlation consistently underestimates the experimental data at all throat gas velocities and liquid to gas ratios. These observations are consistent with earlier evaluations of this model by Yung et al. (1977b). Underprediction by Hesketh's correlation is not surprising since it was based on examination of many fixed-throat industrial scrubbers with liquid injection upstream of the throat. When the scrubbing liquid is injected at the throat, Hesketh expected his model to predict pressure drops about 10% lower than the actual values.

Modified Calvert Model

At all throat gas velocities this model underestimated the data for liquid to gas ratios below 8.0×10^{-4} m^3 liquid/m^3 air, but the reverse was true for liquid to gas ratios exceeding 1.3×10^{-3}. Discounting this trend, the relatively good agreement is surprising since wall friction losses, converging section accleration losses, diffuser section recoveries, and wall film flows are all neglected. Only the drop acceleration losses in the throat were considered in order to simplify the analysis. The authors reasoned that wall friction is compensated by pressure recovery.

An evaluation of the individual pressure loss and recovery components showed that at an L/G ratio of 4.0×10^{-4} m^3 liquid/m^3 air and a throat gas velocity of 76.2 m/s the neglect of wall friction and converging section losses was compensated by

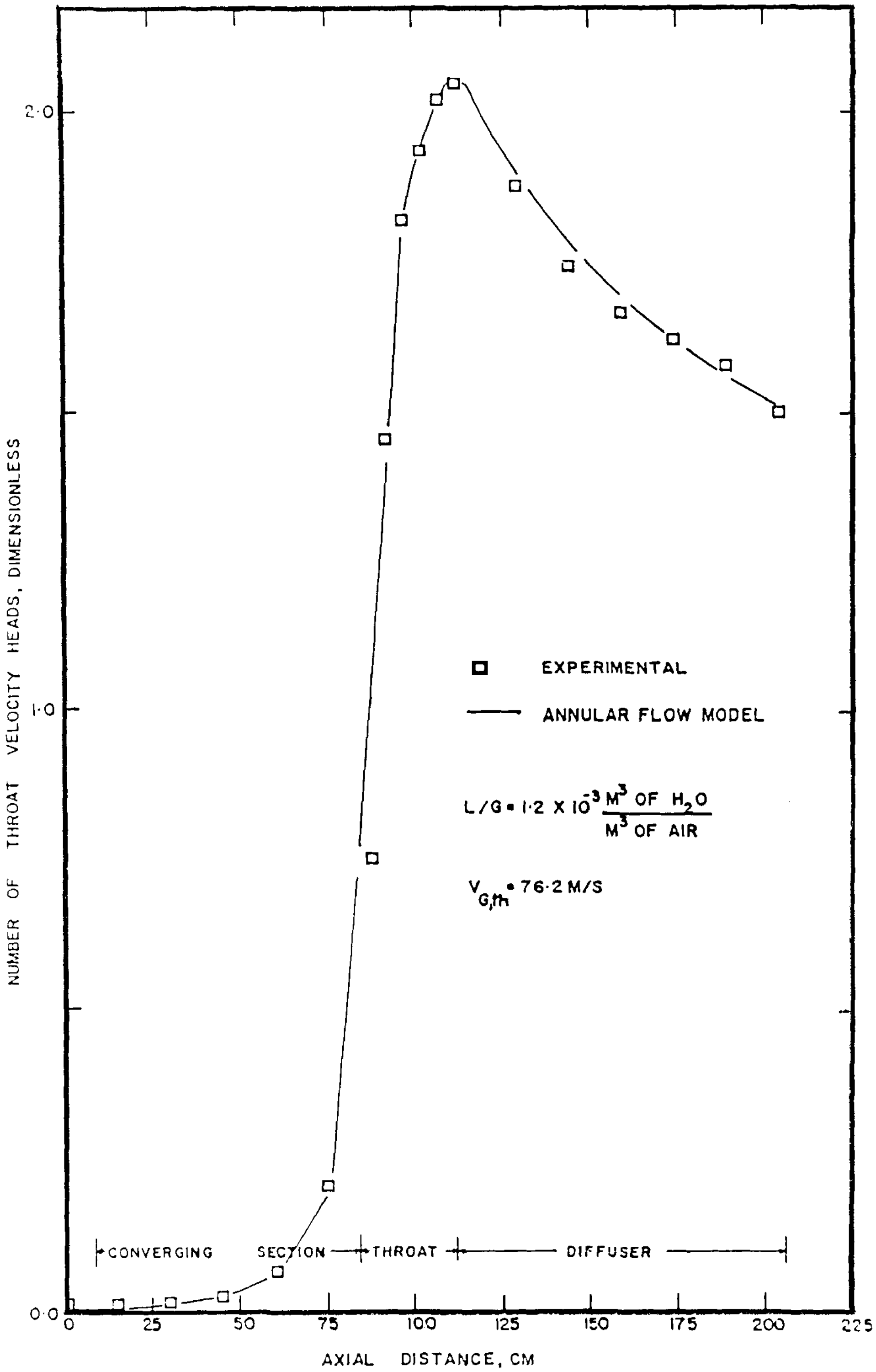

Figure 7. Comparison of experimental pressure drop data with annular flow model predictions.

pressure recovery in the diffuser. Consequently, the model must be underestimating the droplet acceleration losses by approximately 15%, which is the deviation between predicted and measured pressure drops. At the same throat gas velocity and an L/G ratio of 1.9×10^{-3} m^3 liquid/m^3 air, the wall friction and gas acceleration losses exceed the pressure recovery so that the model should underestimate the total losses. However, the predicted values exceed the measured ones by approximately 10%. This means that the droplet acceleration losses are in error by more than this amount. This deviation could be attributed to the neglect of film flow on the wall that would not be accelerated to the droplet core velocity.

The modified Calvert model depends only on the venturi throat geometry. The configurations of the converging and di-

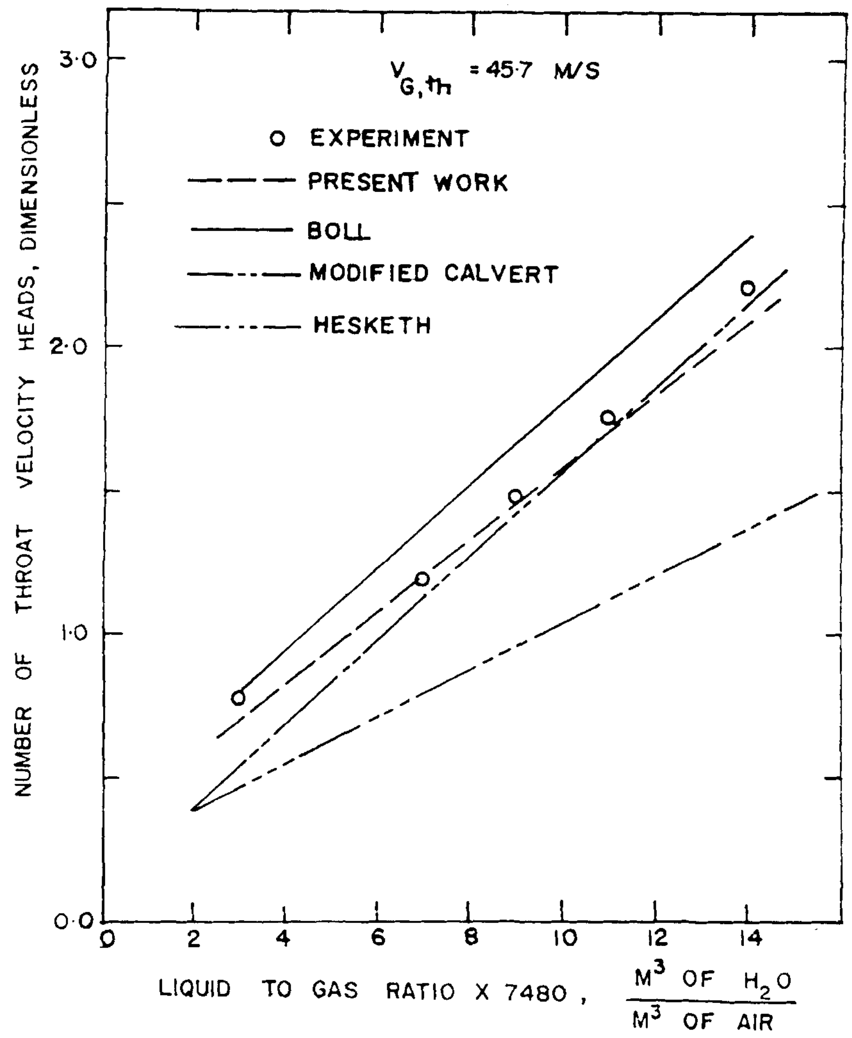

Figure 8. Comparison of experimental and predicted overall pressure drop data for $V_{G,th}$ = 45.7 m/s.

verging sections are neglected. In actual design practice, these have been shown to be primary design variables that can influence the scrubber collection efficiency and overall pressure drop. For these reasons, this model should be used for quick and approximate estimates of the overall pressure drop.

Boll's Model

Figures 8 to 10 show that the Boll model consistently overpredicted the measured pressure drop. The deviation increased with increasing L/G ratios. Since Boll assumed that all of the liquid is entrained (no film flow), the droplet acceleration component would always be overestimated. In addition, the estimate of wall friction in terms of a homogeneous flow model with a fluid density equal to $(m + 1)\rho_G$ should increase the overestimation of the actual frictional component with respect to the homogeneous flow model (Boll, 1973; Wallis, 1969). This overestimation will be more pronounced as the liquid to gas ratio is increased.

The newly developed annular flow pressure drop model provides better agreement with experimental data than any available correlations for the prediction of pressure drop in Pease–Anthony-type venturi scrubbers. The success is due to a more accurate estimation of multiphase flow losses that are evaluated in terms of experimentally measured film flow rates and two-phase frictional multipliers.

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NOTATION

A = cross-sectional flow area of the scrubber, m^2
C = core entrainment factor, dimensionless
C_D = standard drag coefficient, dimensionless
C_{DN} = modified drag coefficient, $C_{DN} = C_D{}^*N_{Re}$, dimensionless
dP = total static pressure drop, N/m^2
dP_{TP} = two-phase frictional pressure drop, N/m^2
D = diameter, m
F = flux, $g/cm^2 \cdot s$
f = friction factor, dimensionless
$\bar{f}$ = average friction factor, dimensionless
G = gas volumetric flow rate, m^3/m
g_c = gravitational conversion constant, $kg.m/N.s^2$
L = liquid volumetric flow rate, m^3/m

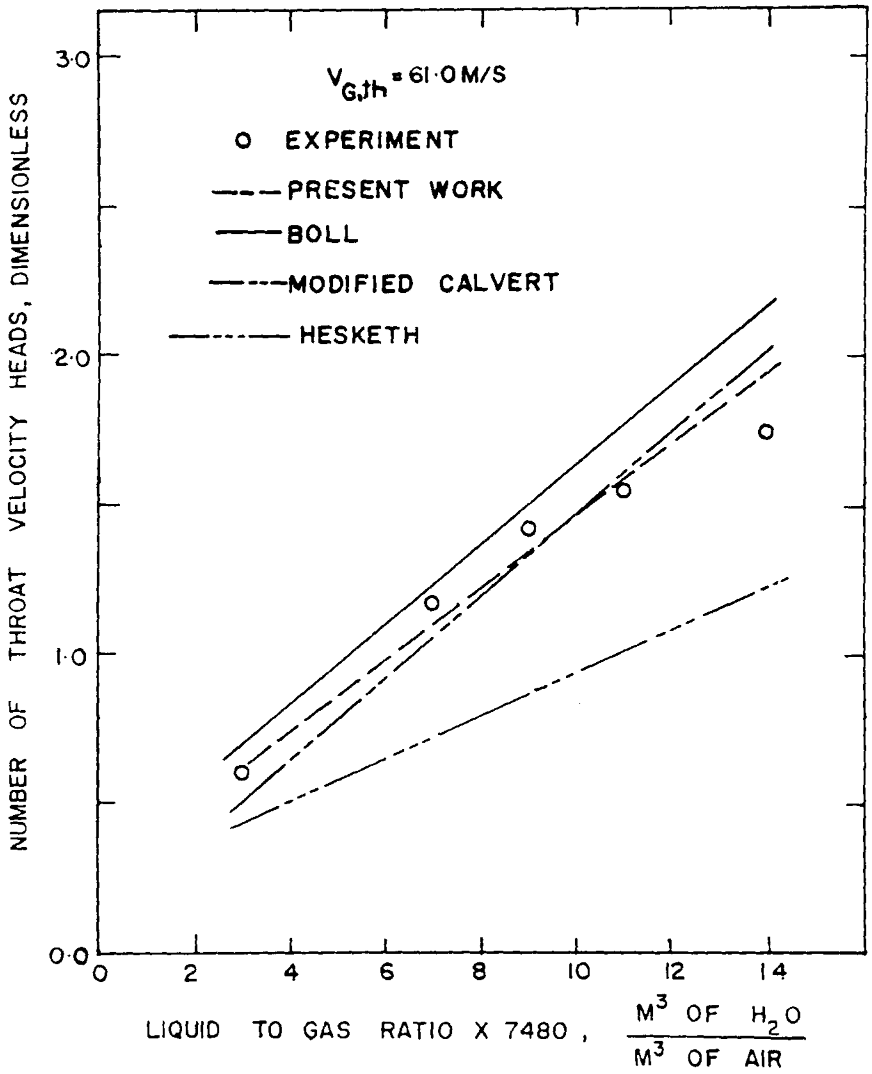

Figure 9. Comparison of experimental and predicted overall pressure drop data for $V_{G,th}$ = 61.0 m/s.

l = length, cm
m = ratio of mass flow rate of liquid to gas, dimensionless
N_{vh} = number of throat velocity heads, N_{vh} = static pressure drop/$(\rho_G V_{G,th}^2/2g_c)$, dimensionless
N_{Re} = Reynolds number, $D_d V_d \rho_G/\mu_G$, dimensionless
Q = volumetric flow rate, m^3/s
t = time, s
V = velocity, m/s
W = mass flow rate, kg/s
X = core quality, dimensionless
Z = scrubber length, m

Greek Letters

α_c = fractional area occupied by homogeneous core, dimensionless
α_G = volume fraction of core occupied by gas, dimensionless
β = convergent angle, degrees
ν = coefficient on divergent loss of gas flow, dimensionless
ΔP = pressure drop, N/m^2
ΔT = temperature change, K
ρ = density, kg/m^3
μ = viscosity, kg/m.s
ξ = head loss ratio, dimensionless
ϕ^2 = two-phase friction multiplier, dimensionless
χ = Martinelli parameter, dimensionless

Subscripts

c = core
d = drop
dif = diffuser
e = equivalent
f = film
G = gas
L = liquid
m = manometer liquid
T = total
th = throat
thi = throat inlet

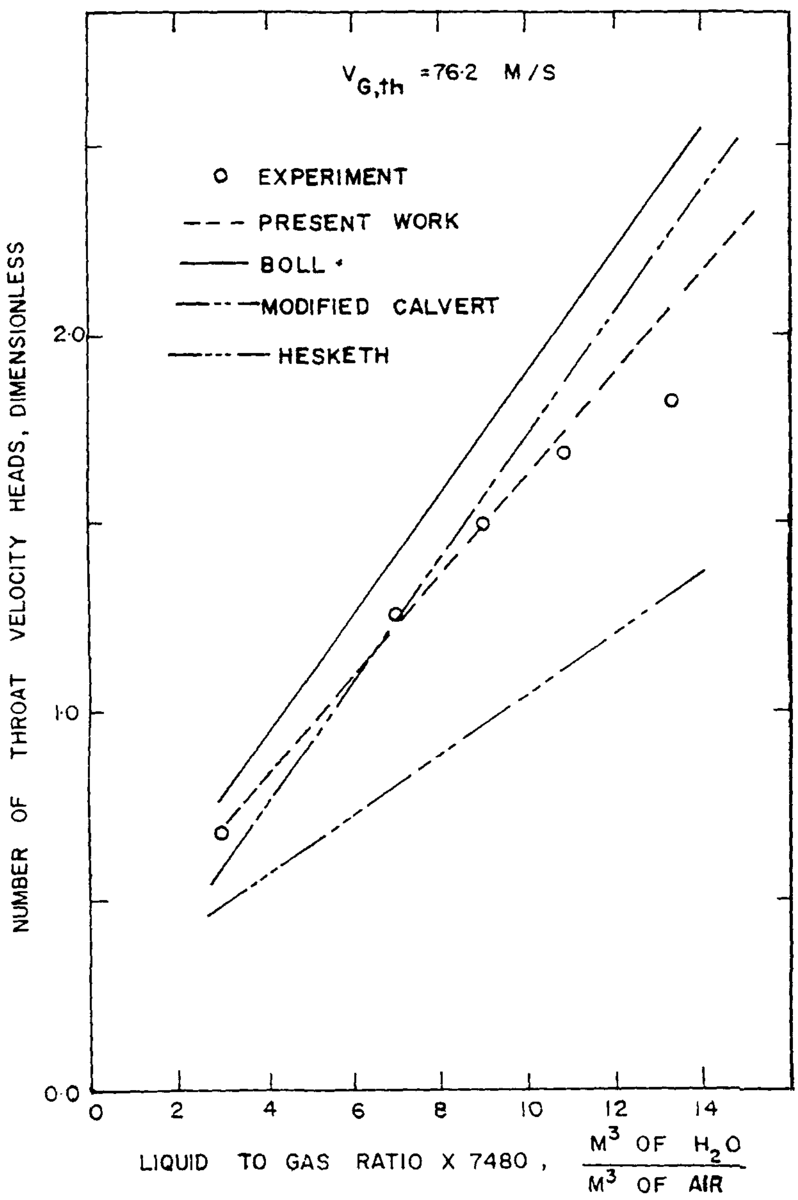

Figure 10. Comparison of experimental and predicted overall pressure drop data for $V_{G,th}$ = 76.2 m/s.

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