ELSEVIER

Chemical Engineering Journal 71 (1998) 153–160

Chemical Engineering Journal

Development of a pressure drop model for a variable throat venturi scrubber

Shekar Viswanathan*

Department of Chemical and Environmental Engineering, National University of Singapore, 10 Kent Ridge Crescent, Singapore 119260

Received 4 March 1998; received in revised form 12 August 1998; accepted 21 August 1998

Abstract

Experimental studies were performed using a pilot-plant scale McInnis–Bischoff variable throat venturi scrubber using operating variable ranges that are normally encountered with industrial size units. The investigation involved the examination of pressure drop as a function of various liquid-to-gas ratios, throat gas velocities and throat area. Two hundred and five separate runs were carried out by measuring pressure drop by varying the throat gas velocity, and the liquid loading. A better understanding of single-phase and two-phase flow losses in variable throat scrubber was achieved during this study. The Rothfus et al. correlation for single-phase flow in annuli has been extended to low $(L'/D)_{eq}$ values for velocities in the range of 60–120 m/s. This extended correlation was essential for the prediction of two-phase frictional losses for all the models tested using the McInnis–Bischoff variable throat venturi scrubber data. The simplified empirical correlation,

$$\Delta P_{TP} = -\Delta P_{SP} + 3.16 \times 10^{-2}\left(\frac{L}{G}\right) - 24.52$$

where the pressure drop expressed in cms of water was obtained for the prediction of two-phase pressure drop at various liquid-to-gas ratios and throat velocities in the McInnis–Bischoff scrubber. © 1998 Elsevier Science S.A. All rights reserved.

Keywords: Venturi scrubber; Variable throat; Pressure drop

1. Introduction

Venturi scrubbers, which have a successful history, are still a popular choice among gas cleaning devices since they are potentially capable of meeting future emission standards with minimum modifications necessary to improve the collection of small particles. As a result, a number of studies have been published with respect to the collection and pressure drop estimation [1–16]. Commercially, there are primarily two types of venturi scrubbers, namely, the constant throat design (usually called as Pease–Anthony type) and the variable throat (usually called McInnis–Bischoff type). Most of the published work focuses on the Pease–Anthony scrubber. Published literature concerning the performance of McInnis–Bischoff scrubber is scarce in spite of their practical importance. They are particularly useful where wide fluctuations in temperature, particulate loading and gas flow rates are experienced. Next to the particle collection efficiency, the overall pressure drop associated with the operation of the system is the most significant information associated with the successful design of a venturi scrubber. Liquid drop acceleration by the gas, irreversible drag force work, and wall losses define the magnitude of the pressure drop [14]. The objective of this work was to investigate the flow losses in the McInnis–Bischoff variable, annular, throat venturi scrubber and analyze the prediction of pressure drop over the range of venturi scrubber operations encountered in industrial applications.

2. Literature review

Calvert [5] was the first to propose a model that would predict pressure drop in a venturi scrubber. Although he derived the pressure drop in terms of force required to accelerate all of the liquid to the gas throat velocity using Newton's Law, he neglected wall friction and momentum recovery in the divergent section. Researchers have continued to seek improvements on this simplistic model which is still used in industry because accurate pressure drop predictions are limited to liquid-to-gas ratios between 0.8 to

*Corresponding author. Tel.: +65-874-4309; fax: +65-872-3154; e-mail: chesv@nus.edu.sg

1385-8947/98/$ – see front matter © 1998 Elsevier Science S.A. All rights reserved.
PII: S1385-8947(98)00123-5

1.3×10^{-3} m^3 liquid/m^3 air. Boll [4] developed a systematic approach based on 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 overpredicted, respectively. These deviations can be attributed to the neglect of multiphase losses, the existence of liquid film flowing on the walls, and nonuniform distribution of scrubbing water. Although several improvements were made on these two models, it essentially remained the same in the accurate prediction of pressure drops. Since then, a new generation of pressure drop models has evolved. Azzopardi and Govan [2] have attempted to redefine the overall momentum balance around the scrubber to include momentum losses due to accelerating droplets entrained from the film and the interfacial drag between the fast moving core and the slower moving liquid film. They made use of the entrainment correlation of Whalley et al. [17] and performed a mass balance on the liquid at incremental axial distances along the venturi scrubber. They, however, acknowledged limited success with this procedure. Viswanathan et al. [14] analysed the deviations between the experimental values and earlier models predictions and attributed them to realistic multiphase losses and developed an annular flow pressure drop model. In this model, they assumed the fraction of total injected liquid flowing as a film on the venturi walls as being constant along the entire scrubber length. They related this liquid fraction to a parameter derived from jet penetration [12,15]. In their work, Viswanathan et al. [14] assumed a constant core quality throughout the venturi, although the examination of the experimental film flow data indicated a change in the mass flowing on the wall along the scrubber length. An improved pressure drop model was proposed in [1,3] by accounting for two-phase pressure recovery in the diffuser as a key component using the growth and separation of gas boundary layer concept. Although this approach provided a better estimate for overall pressure drops, the prediction of axial variation of the same was less than optimal in many of the instances. In addition, the models still did not account for the variation in pressure recovery between gas and liquid phases. In 1996, Allen and van Santen [18] indicated the importance of dry pressure drop in venturi scrubbers and showed for the first time, agreement between the predicted and experimental pressure drop values using this concept for varying scrubber sizes. Recently, Pulley [7] found that the model of Azzopardi et al. [3] predicted pressure drop for venturi scrubbers reasonably well. For prediction to be further improved, Pulley suggests more experimental investigation such as drop size, entrainment at liquid injection and entrainment and deposition along the venturi length.

In order to describe accurately the physical phenomena occurring in a venturi scrubber, a true representation of the liquid distribution throughout the scrubber must be available. The film on the containing walls with its wavy surface, presents a roughened interface that increases the frictional component of the pressure drop. The continuous deposition and entrainment of the liquid along the venturi scrubber length and the associated loss of momentum due to the acceleration of the newly entrained droplets, also contribute to the overall pressure drop of the system. This research work utilizes the reviewed literature in the development of an annular flow pressure drop model for a variable throat venturi scrubber.

2.1. Annular pressure drop model

Viswanathan et al. [14] developed an annular flow pressure drop model for venturi scrubbers taking into account the presence of a two-phase flow with the assumption that a homogeneous core fluid flowing cocurrently with a liquid film on the wall each with different velocities. Mathematically, one-dimensional steady-state force balance on a differential scrubber cross-section yields:

$$-\frac{dP}{\rho_G} = \left[\frac{\alpha_G \alpha_c V_G}{g_c}\right] dV_G + \left[\frac{C\eta\alpha_G\alpha_c V_G}{g_c}\right] dV_{d,x} + \left[\frac{(1-C)\eta\alpha_G\alpha_c V_G}{g_c}\right] dV_f + \left[\frac{2f_c W_c^2 \phi_c^2}{g_c D_{eq} \rho_c A_T^2 \rho_G}\right] dx \tag{1}$$

This model has been proven to be applicable for Pease–Anthony type venturi scrubbers [14]. In order to solve Eq. (1) and obtain a total pressure drop, a step-wise numerical integration needs to be performed. The steps include:

1. Determining the drop acceleraton from [14]:

$$a_x = k_D(V_G - V_{d,x}) + g_x \tag{2}$$

2. Calculating the droplet velocity from

$$V_{d,x} = V_{d,\text{initial}} + \int \left(\frac{dV_{d,x}}{dt}\right) dt \tag{3}$$

3. Calculating the void fraction, α_G, from

$$\alpha_G = \frac{Q_G}{Q_G + Q_d} \tag{4}$$

4. Determining the pressure gradients for the core and film respectively using Martinelli's equations:

$$\left[\frac{dP}{dx}\right]_c = \frac{2f_c W_c^2}{g_c D_{eq} A_T^2 \rho_c} \tag{5}$$

$$\left[\frac{dP}{dx}\right]_f = \frac{2f_f W_f^2}{g_c D_{eq} A_T^2 \rho_f} \tag{6}$$

5. Evaluating the Martinelli parameter, χ, from

$$\chi = \sqrt{\frac{(dP/dx)_f}{(dP/dx)_c}} \tag{7}$$

6. Calculating the annular void fraction, α_c, using the correlation of Hewitt and Hall–Taylor [19]:

$$\alpha_c = \left[1 + \chi^{0.8}\right]^{-0.378} \tag{8}$$

7. Calculating the two-phase flow multiplier, ϕ_c, using Lockhart and Martinelli correlations [20]:

$$\phi_c^2 = 1 + 12\chi + \chi^2 \quad \text{for } N_{\mathrm{Re,f}} < 2000 \tag{9}$$

$$\phi_c^2 = 1 + 20\chi + \chi^2 \quad \text{for } N_{\mathrm{Re,f}} < 2000 \tag{10}$$

8. Calculating the gas and film velocities using the geometrical considerations of the flow area:

$$V_G = \frac{W_G}{\rho_G \alpha_c \alpha_G A_T} \tag{11}$$

$$V_f = \frac{W_f}{\rho_f (1 - \alpha_c) A_T} \tag{12}$$

Substitution of all these values and integration of Eq. (1) will provide pressure drop as a function of length.

3. Experimental details

3.1. Venturi scrubber

A schematic representation of the McInnis–Bischoff variable throat venturi scrubber is shown in Fig. 1[21]. It utilises a movable plug displacer to create a variable annular opening over the length of the entrance section. This approach has several advantages, particularly where wide fluctuations in temperature, particulate loading and gas volume are associated with cyclic or batch type processes, such as cupola operations. The simplicity and practicability of the variable throat often make possible the use of lower horsepower electric motors and lead to more efficient use of static pressure for achieving optimum gas cleaning efficiencies [22]. The McInnis–Bischoff pilot plant was used to study the axial pressure gradients and irreversible losses as functions of

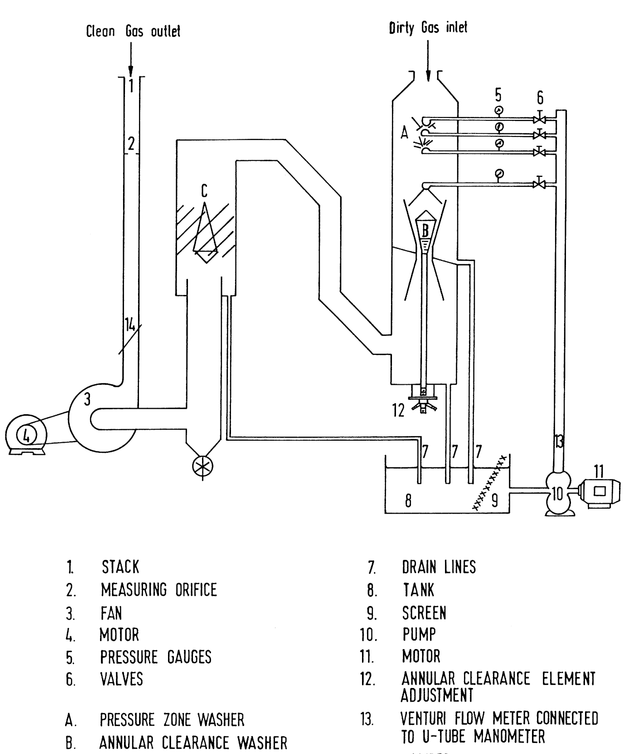


Fig. 1. Schematics of McInnis–Bischoff pilot plant unit.

1. liquid-to-gas ratio;
2. plug position;
3. the number and location of liquid injecting nozzles.

This system forces the gas stream to flow down the vertical shell enclosing the venturi. Water is sprayed from axial nozzles at different positions upstream of the venturi throat. This inlet section serves to condition the gas stream and to remove large particulate matter. Only a fraction of the injected water enters the converging section of the venturi where finer particles are scrubbed.

3.1.1. Principle of operation

The dirty gas stream enters the pressure zone (A) portion of the washer, which is designed to cool and saturate the gas as well as remove the larger particles by direct impingement on the water stream. Four water nozzles, each with a large single orifice, distribute the water from central locations at different levels. The second stage consists of the annular clearance washer (B) as shown in Fig. 1. Here the fine dust is removed. The venturi tube and a tapered movable plug in the converging section characterize this section. The plug position is adjustable along its vertical axis. Its displacement reduces or increases the annular clearance between it and the tube. As the gas flows through the annulus, which narrows in the downstream direction, its velocity increases and entrains the water injected into the gas flow. The annular clearance washer provides control over the fan parameters such as gas flow rate, static pressure drop and motor horsepower [23]. In practice, the annular clearance plug can be used to create a higher overall static pressure drop to increase the collection efficiency as a result of the increased gas velocity and scrubbing action of the water droplets in the throat section. In the diffuser section, the velocities of the gas and liquid decrease to provide a kinetic energy recovery. However, the water droplet deceleration is normally less than that of the gas because of greater inertia and minimal gas–liquid interaction.

3.2. Experimental procedure

The movable plug (displacer) was first set at the desired position by matching the location of a pointer attached to the plug adjustment element to a linear scale mounted on the structural frame as shown in Fig. 2. For test purposes, the

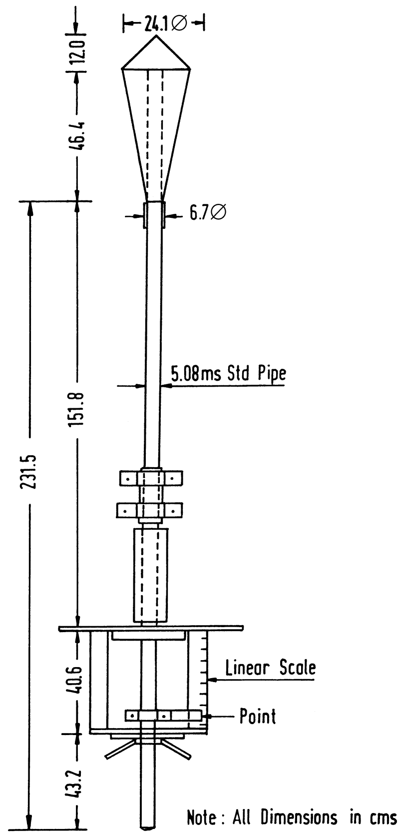


Fig. 2. Details of movable plug.

pointer was set at the 2, 4, 6, 8, and 10 levels on the scale to provide different throat sectional areas in the annular clearance washer. Air was drawn directly from the atmosphere through 30.5 cm inlet. Adjusting the dampers located on the discharge side of the blower varied air volumetric flow rates. Water was introduced through a manifold into pressure nozzles. The design of the nozzle system was such that gate valves located in the take-off from the main line could adjust the water flow rates through each nozzle. Calibrated pressure gauges were provided in each injection nozzle line to allow correlation of the venturi pressure drop to different experimental conditions. Nozzles were 12.7 mm in diameter. By design the first and fourth nozzle face downwards in the direction of gas flow while the second and third point upwards. It was important to determine how much of the atomized liquid actually passed through the venturi throat. Since all of the liquid injected into the gas stream do not reach the throat, the investigation was designed to provide

1. the relationship between the actual liquid-to-gas ratio (*L*/*G*) existing in the throat and the total pressure drop across the scrubber
2. the ratio of the total mass flow rate introduced through the nozzles to the liquid flow rate passing through the venturi throat in order that the nozzle number and design can be optimized to minimize pumping costs
3. a comparison of previous results with the present study

In order to evaluate the effects of the four nozzles placed at different positions above the plug in the pressure zone section, the overall axial venturi pressure drop was studied for all possible combinations of the water injection nozzles. Experiments were carried out for five different plug positions, with liquid-to-gas ratios varying from 1.6×10^{-4} to 2.6×10^{-3} m^3 liquid/m^3 air, gas throat velocities between 60 and 120 m/s. A total of 205 runs were made for single-phase air and air–water flows.

4. Results and discussion

The results of this investigation is discussed in terms of liquid film mass flow rate, fraction of the total injected liquid flowing on the wall as a film, liquid film thickness, liquid film velocity, measured pressure drop data, and comparisons to proposed pressure drop model.

4.1. Single-phase air

Table 1 represents average values of six independent data sets corresponding to each plug setting. Single-phase determinations were made before and after completion of each liquid injection nozzle combination to check the reproducibility of the gas flow rates during the test period.

These data cover all five-plug positions, which gave throat velocities ranging from 60 to 120 m/s. Because of the corroded and scaled surface of the venturi, friction factors were expected to be higher than for new commercial steel surfaces. According to Simpson [22], the friction factors should be about 20–30% higher but could increase by 100% if the condition of the surface has been severely affected by corrosion, erosion or scaling. For modelling

Table 1
Single phase air pressure drop data

Run No.	Plug position	Area of throat (m^2)	Area of throat (m^2)	Overall axial pressure drop (Pa)
1	10	0.0155	60.4 ± 1.5	1422 ± 49
42	8	0.0134	67.7 ± 1.8	1422 ± 9.8
83	6	0.0109	77.4 ± 2.4	1716 ± 98
124	4	0.0082	91.4 ± 3.0	2863 ± 147
165	2	0.0042	120.0 ± 5.2	6178 ± 226

Table 2
Relationship of entrance section friction factors to the fully developed flow friction factors, f/f_∞

Plug setting	Average Reynolds number in throat	Throat length (mininum throat diameters)	Ratio of friction factor for underdeveloped flow to the fully developed, f/f_∞
3	1.36×10^5	15.53	2.31
4	1.8×10^5	9.10	3.31
6	2.2×10^5	6.90	3.92
8	2.7×10^5	5.90	4.80
10	2.7×10^5	5.16	5.0

purposes, it was assumed that the friction factors would be 30% higher than those corresponding to new commercial steel surfaces with an absolute roughness value of 0.0046 cm [24]. Rothfus and co-workers [25] have concluded that the friction factors based on hydraulic radius concept are in fair agreement with experimental results for flow in annuli. They indicated that better agreement with the experimental values could be obtained for fluid flowing in an annulus by calculating friction factors based on a radius corresponding to the maximum velocity. In addition, friction factors for entrance regions in the annuli were found to be appreciably higher than for fully developed turbulent flow. On this basis, single-phase air pressure drops were compared by

- setting the absolute roughness of the venturi surface at 0.0046 cms to correspond to new commercial venturi;
- considering the friction factors to be 30% higher than values expected for new commercial steel. This is equivalent to increasing the absolute roughness by a factor of approximately 10;
- computing friction factors from the Colebrook equation [24] and multiplying by the appropriate correction factor for fluid flow in an annulus;
- correcting the friction factor values in terms of multipliers accounting for the undeveloped turbulent flow or entrance effects for each plug setting.

The values of the multipliers listed in Table 2 are those obtained by forcing agreement between experimental results and theoretical predictions. The resulting extension of the Rothfus and co-workers [25] correlation to low $(L'/D)_{eq}$ values is shown in Fig. 3. Values of f/f_∞ derived from the single-phase measurements were used in the prediction of two-phase pressure drops for each plug setting.

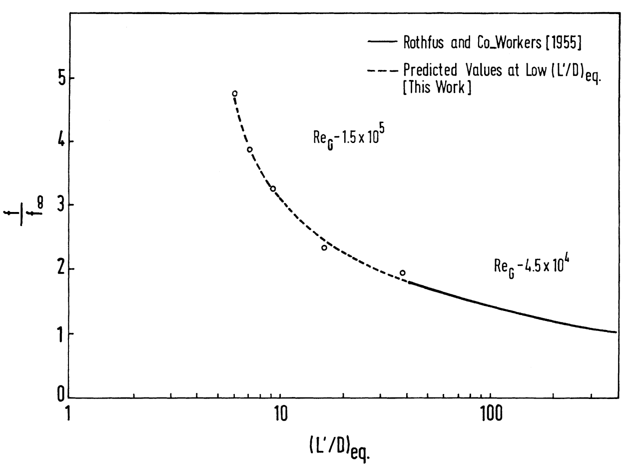


Fig. 3. Single-phase extension of Rothfus and coworkers [24] plot for entrance effects.

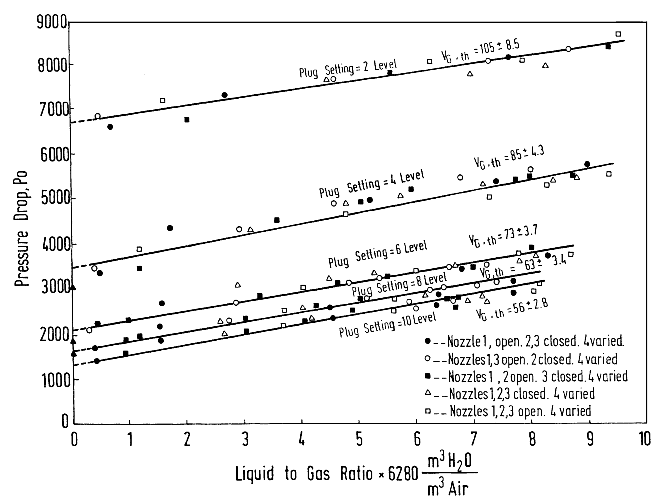


Fig. 4. Validation of overall pressure drop with liquid to gas ratio.

4.2. Air water flows

Fig. 4 illustrates the dependence of pressure drop on the liquid-to-gas ratio, L/G, through the throat, at the five different plug settings. The effect of water addition, for various liquid-to-gas ratios through the throat, was isolated by subtracting the extrapolated single-phase air losses in Fig. 4 from the two-phase pressure drop. The results are illustrated in Fig. 5. A linear regression analysis of the data provided in Fig. 5 yielded the equation

$$\Delta P_{TP} = -\Delta P_{SP} + 3.16 \times 10^{-2}\left(\frac{L}{G}\right) - 24.52$$

for the two-phase flow pressure drop in the McInnis–Bishoff system. This relationship is similar to the one proposed by Allen and van Santen [18]. As observed by these researchers, for short-throated venturis such as the one presented in this work, the acceleration loss of gas phase may dominate

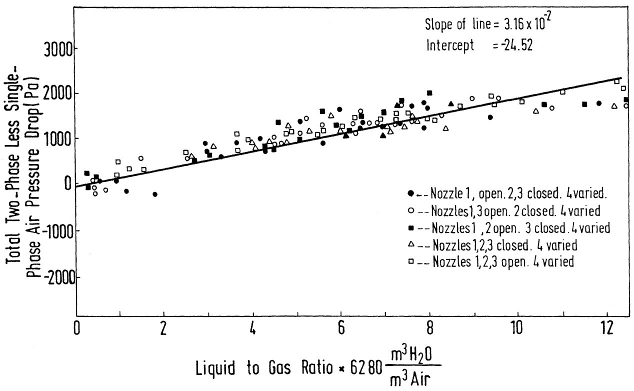


Fig. 5. Effect of water addition on overall pressure drop.

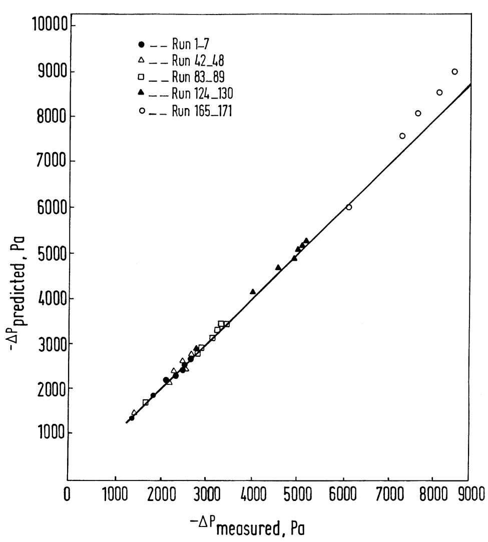


Fig. 6. Comparison of experimental pressure drop with annular flow model.

the total pressure drops. Hence the accurate estimation of dry pressure drop is very critical for the prediction of overall pressure drop in variable size venturi units.

4.3. Model validation

Fig. 6 provide the agreement between the experimentally measured pressure drops to the values predicted by the annular flow model. These data correspond to water injection from nozzle 4 in a downward direction cocurrent to the airflow. The design of the McInnis–Bishoff venturi system did not permit the measurement of liquid flows in the core or on the walls. Consequently, to fit the model, optimum values of entrainment coefficient C were derived by a trial and error procedure, which forced agreement between experimental pressure drops, and values predicted by the annular flow model employing the Lockhart–Martinelli approach. Figs. 7 and 8 graphically compare the pressure drop predictions as a function of L/G ratios of Calvert, Boll, and annular flow models with the experimental data. It is clear from these data that there are variations between the pre-

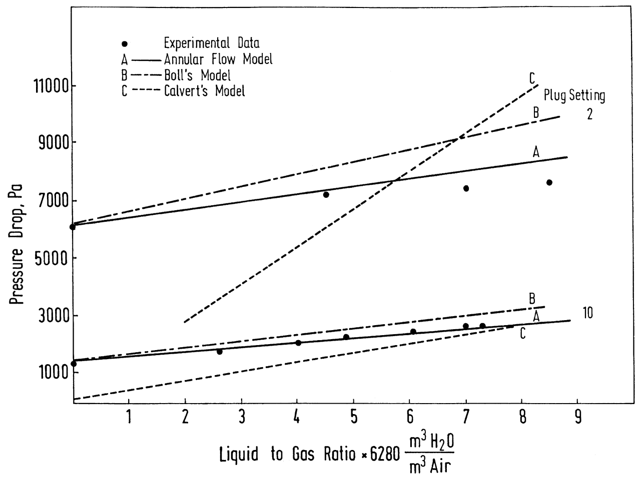


Fig. 7. Comparison of various model predictions with experimental pressure drop.

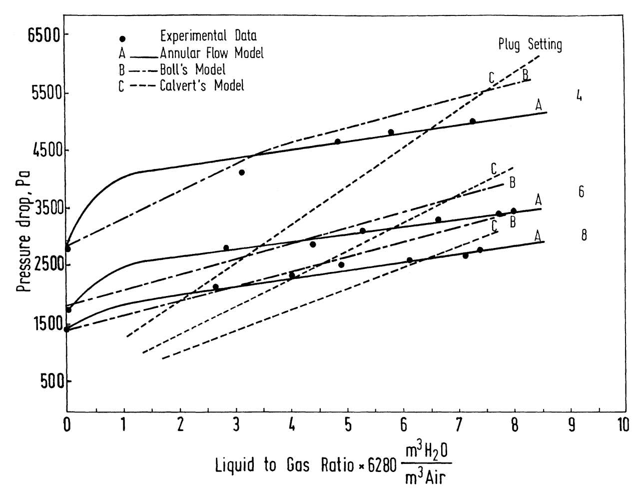
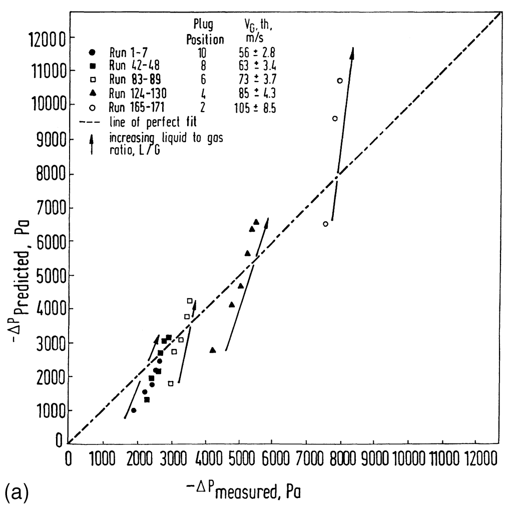


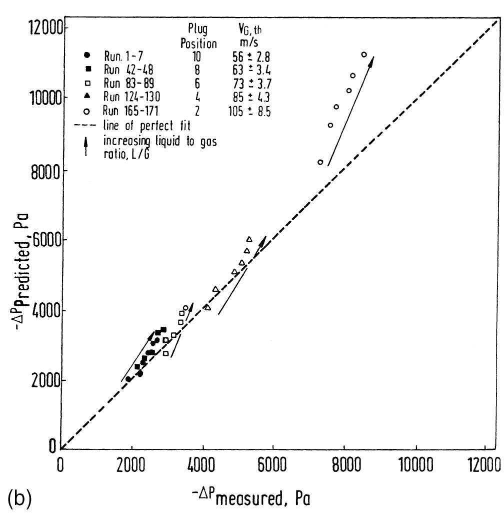
Fig. 8. Comparison of various model predictions with experimental pressure drop.

dictions of these models. To verify the applicability of these models that are commonly used in the industry, a comparison of predictions are made as shown in Fig. 9(a)–(c). The 95% confidence intervals based on absolute mean difference was found to be 1.4, 1.0, and 0.4 for the Calvert, Boll, and annular flow models, respectively. These values clearly indicate that the annular flow model provides the best agreement with the experimental data for the entrainment coefficient ranges indicated in Fig. 9(c).

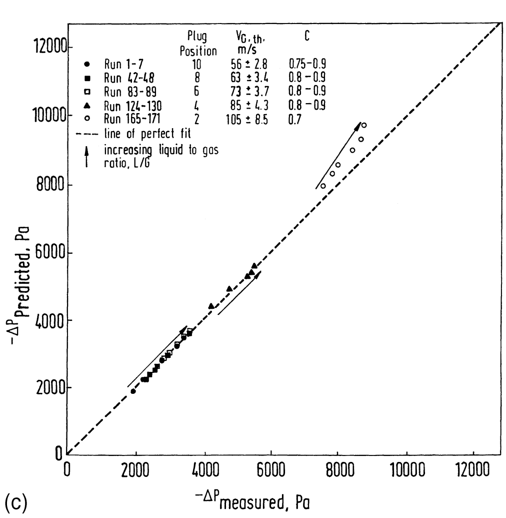
According to Fig. 9(a), Calvert's model underestimates the pressure drop at low L/G ratios for all throat velocities examined. This behaviour is consistent with Calvert's assumption of negligible frictional losses. With increasing liquid injection rates at essentially fixed throat velocities better agreement is attained since the neglected frictional pressure drop compensates for the overestimated acceleration losses, which predominate. At liquid-to-gas ratios



a. Calvert Model



b. Boll Model



c. Annular Flow Model

Fig. 9. Pressure drop model validation for McInnis–Bischoff Venturi scrubber.

exceeding 7, the frictional losses become negligible so the model overestimates the total pressure drop because

- a portion of the injected liquid flows along the walls at relatively low velocity;
- the atomized liquid is not accelerated up to the gas throat velocity because of the short throat section in the variable throat venturi system.

Fig. 9(b) shows that the Boll approach, using single-phase friction factors obtained for corroded surfaces from Fig. 6 rather than a constant value of 0.00675, is a considerable improvement over the simplistic Calvert model. At low *L/G* ratios for throat gas velocities below 91 m/s the close agreement could be fortuitous. Since Boll assumed that all of the liquid is atomized, there should always be overprediction in terms of the accelerational component. Good agreement is possible if the homogeneous flow model underestimates the frictional losses. At the highest throat velocities tested, the Boll model consistently overpredicts the overall pressure drop because the accelerational component accounts for more than 70% of the total pressure drop. Assumption of complete atomization of all injected liquid increases the magnitude of the overprediction with increasing *L/G* ratios and throat gas velocities.

5. Conclusions

A better understanding of single-phase and two-phase flow losses in variable annular throat scrubber was achieved during this study. The Rothfus et al. correlation for single-phase flow in annuli has been extended to low $(L'/D)_{eq}$ values for velocities in the range of 60–20 m/s. This extended correlation was essential for the prediction of two-phase frictional losses for all the models tested using the McInnis–Bischoff variable throat venturi scrubber data. The simplified empirical correlation,

$$\Delta P_{TP} = -\Delta P_{SP} + 3.16 \times 10^{-2}\left(\frac{L}{G}\right) - 24.52$$

where the pressure drop expressed in cms of water was obtained for prediction of two-phase pressure drop at various liquid-to-gas ratios and throat velocities in the McInnis–Bischoff scrubber.

6. Nomenclature

a	drop acceleration (m/s^2)
A	cross-sectional area of the scrubber (m^2)
C	core entrainment factor (dimensionless)
C_D	standard drag coefficient (dimensionless)
C_{DN}	modified drag coefficient, $C_{DN} = C_D \times N_{Re}$ (dimensionless)
dP	total static pressure drop (N/m^2)

dP_{TP}	two-phase frictional pressure drop (N/m^2)
D	diameter (m)
f	single phase fanning friction factor (dimensionless)
f/f_∞	ratio of friction factor for undeveloped flow to fully developed flow (dimensionless)
g	acceleration due to gravity (m/s^2)
G	Gas flow rate (m^3/m)
g_c	gratvational conversion constant (kg m N s^2)
k_D	constant equal to $0.75(\mu_g/\rho_g D_d)C_{DN}$ and defined by Eq. (2) (1/s)
L	Liquid flow rate (m^3/m)
$(L'/D)_{eq}$	ratio of throat length to minimum throat diameter (dimensionless)
x	scrubber length (m)
V	velocity (m/s)
W	mass flow rate (kg/s)

Greek symbols

α_c	fractional area occupied by core (dimensionless)
α_G	volume fraction of core occupied by gas (dimensionless)
ΔP	pressure drop (cm of H_2O)
ρ	density (kg/m^3)
μ	viscosity (kg/m s)
η	ratio of mass flow rate of liquid-to-gas (dimensionless)
ϕ^2	two-phase frictional multiplier (dimensionless)
χ	Martinelli parameter (dimensionless)

Subscripts

x	direction
c	core
d	drop
eq	equivalent
f	film
G	gas
L	liquid
SP	single-phase
T	total
TP	two-phase
th	throat

Acknowledgements

The author would like to thank Mr. Y.C. Jain's assistance in the collection of experimental data and Professors Alex Gnyp and Carl St.Pierre's assistance in the review of this paper.

References

[1] B.J. Azzopardi, Gas–liquid flows in cylindrical venturi scrubbers: boundary layer separation in diffuser section, Chem. Eng. J. 49 (1992) 55.
[2] B.J. Azzopardi, Govan, A.H., The modelling of venturi Scrubbers, Filtration and Separation, May/June (1984) 196.
[3] B.J. Azzopardi, S.F.C.F. Teixeira, A.H. Govan, T.R. Bott, An improved model for pressure drop in venturi scrubbers, Trans. Inst. Chem. Engrs B 69 (1991) 237.
[4] R.H. Boll, Particle collection and pressure drop in venturi scrubbers, Ind. Eng. Chem. Fund. 12 (1973) 40.
[5] S. Calvert, venturi and other atomizing scrubbers, AIChE J. 16 (1970) 392.
[6] H.E. Hesketh, Fine particle collection efficiency related to pressure drop, scrubbant and particle properties, and contact mechanisms, J. Air. Poll. Control Assoc. 24 (1974) 939.
[7] R.A. Pulley, Modelling the performance of venturi scrubbers, Chem. Eng. J. 67 (1997) 9.
[8] J. Yamauchi, T. Wada, H. Kamei, Pressure drop across the venturi scrubbers, Kagaku Kogaku 27 (1964) 974.
[9] T. Yoshida, N. Morishima, M. Hayashi, Pressure loss in flow through venturi tubes, Kagaku Kogaku 24 (1960) 20.
[10] T. Yoshida, N. Morishima, M. Hayashi, Pressure loss for the acceleration of atomized droplets, Kagaku Kogaku 9 (1965) 308.
[11] S.C. Yung, H.F. Barbarika, S. Calvert, Pressure loss in venturi scrubbers, J. Air. Poll. Control Assoc. 27 (1977) 348.
[12] S. Viswanathan, A.W. Gnyp, C.C. St Pierre, Jet penetration measurements in a venturi scrubber, Can. J. Chem. Eng. 61 (1983) 504.
[13] S. Viswanathan, A.W. Gnyp, C.C. St. Pierre, Examination of gas–liquid flow in a venturi scrubber, Ind. Eng. Chem. Fund. 23 (1984) 303.
[14] S. Viswanathan, A.W. Gnyp, C.C. St. Pierre, Annular flow pressure drop model for Pease–Anthony type venturi scrubbers, AIChE J. 31 (1985) 1947.
[15] S. Viswanathan, A.W. Gnyp, C.C. St, Pierre, Estimating film flow in a venturi scrubber, Particulate Sci. Technol. 15 (1997) 65.
[16] S. Viswanathan, Modelling of venturi scrubber performance, Ind. Eng. Chem. Res. 36(10) (1997) 4308.
[17] P.B. Whalley, P. Hutchinson, G.F. Hewitt, Heat transfer, Scripta Book Co. 4 (1974) 290.
[18] R.W.K. Allen, A. van Santen, Designing for pressure drop in venturi scrubbers: the importance of dry pressure drop, Chem. Eng. J. 61 (1996) 203.
[19] Hewitt, G.F., Hall, N.S., Taylor, Annular Two-Phase Flow, Pergamon Press, Oxford, 1970, p. 31.
[20] R.W. Lockhart, R.C. Martinelli, Proposed correlation of data for isothermal two-phase, two-component flow in pipes, Chem. Eng. Prog. 49 (1949) 39.
[21] McInnis Equipment Limited, Windsor, Ontario, Canada, Report No. 33-01-05, March 1972.
[22] R.G. Montrose, Variable throat venturi scrubber, Eng. Digest 43 (1973) 41.
[23] L. Simpson, Piping sizing for process plant, Chem. Eng. 32 (1968) 197.
[24] P.M. Gerhart, R.J. Gross, J.I. Hochstein, Fundamental of Fluid Mechanics, Addison-Wesley Publishing Company, New York, 1992, pp. 481-488.
[25] R.R. Rothfus, C.C. Monrad, K.G. Sickhchi, W. Heideger, Isothermal skin friction in flow through annular sections, Ind. Eng. Chem. 47 (1955) 5.