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Prediction of non-Newtonian head losses through diaphragm valves at different opening positions

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ABSTRACT

Recent work on fully opened rubber-lined diaphragm valves showed that due to the lack of geometric similarity, dynamic similarity could not be established. The laminar flow loss coefficient constant therefore becomes diameter dependent as is the case of turbulent flow loss coefficients. The purpose of this work was to establish if this is the case for all types of diaphragm valves, by testing diaphragm valves from a different manufacturer. Accurate loss coefficient data is critical for energy efficient hydraulic design. Saunders type straight-through diaphragm valves ranging from 40 mm to 100 mm were tested in the fully open, 75%, 50% and 25% open positions, using a range of Newtonian and non-Newtonian fluids. It was found that the laminar flow loss coefficient constant suggested by Hooper (1981) is sufficient for all valve diameters at Reynolds numbers below 10. However, for transitional and turbulent flow the same loss coefficients cannot be applied for more accurate designs for diaphragm valves from different manufacturers.

A new correlation has therefore been developed to predict the loss coefficients for straight-through Saunders diaphragm valves at various openings from laminar to turbulent flow regimes.

Keywords: Laminar flow; Diaphragm valves; Head loss; Pressure; Non-Newtonian; Efficient design

1. Introduction

Spellman and Drinan (2001) defined a valve as any device by which the flow may be started, stopped, or regulated by a movable part that opens or obstructs passage. Valves are therefore an important part of any pipeline system. Diaphragm valves offer distinct advantages in applications where absolute sealing is required, and where the line fluid cannot be contaminated by the ingress of atmosphere. Even when slurries are being handled, or solids are present in the liquids, leak-tightness is assured; due to the ability of the diaphragm to engulf particles on closure, and release them downstream when the valve is again opened (Myles K and Associates cc, 2000). There is no need for any gland-packing devices for the stem, as the diaphragm provides total sealing between the medium and atmosphere.

Valve losses may be neglected without serious uncertainty in long pipelines, but in shorter pipelines an accurate knowledge of their effects must be known for correct engineering calculations (Streeter and Wylie, 1985). Miller (1990) classifies loss coefficients for diaphragm valves as class 3, which means that they have not been verified by independent studies. Miller (1990) and Perry (1997) published some loss coefficient data in the turbulent regime at various valve opening positions (for fully, 75%, 50%, and 25%) that is defined as the mass or volumetric flow delivery percentage function of the travel of the hand-wheel of the valve (Hutchison, 1976). Unfortunately, the valve size has not been mentioned and this raised questions regarding its applicability for accurate design purposes. In 2004, ESDU provided a correlation supported by some graphs to compute the loss coefficient at various opening positions for different valve sizes in both laminar and turbulent flow, but correction factors were only available for two types of diaphragm valves. These coefficients were all determined for Newtonian fluids. Pienaar et al. (2004) tested non-Newtonian kaolin slurry through a 40 mm Natco diaphragm valve. They

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Nomenclature

Symbols

α_1	correction factor for partial opening of valve
α_2	correction factor for low Reynolds number
λ_Ω	loss coefficient at fully open position
θ	opening position
$\dot{\gamma}$	shear rate (s^{-1})
%	percentage
ρ	density of the fluid (kg/m^3)
τ	shear stress (Pa)
τ_0	shear stress at the wall (Pa)
τ_y	yield stress (Pa)
Δp_v	total pressure loss in the valve (Pa)
C_v	laminar flow valve loss coefficient constant
D	diameter (m)
DS	downstream
f	fanning friction factor
g	gravitational acceleration (m/s^2)
H_f	friction loss (m)
H_v	valve loss (m)
ID	internal diameter
k_v	loss coefficient of the valve
k'	uncorrected pressure loss coefficient
K_1	k_v for the fitting at Re = 1
K_∞	k_v for a large fitting at Re = ∞
K'	apparent fluid consistency index (Pa s^n)
K_d	constant for loss coefficient in the 3-K method
K_i	constant for loss coefficient in the 3-K method
K_m	constant for loss coefficient in the 3-K method
L	the length of the pipe (m)
m	mass (kg)
n	flow behaviour index
n'	apparent flow behaviour index
p	point pressure (static) (Pa)
PD	positive displacement
Q	volumetric flow rate
Re	Reynolds number for Newtonian fluids
Re_3	modified Reynolds number for yield pseudo-plastic and Bingham plastic fluids
US	upstream
V	mean velocity (m/s)
V_{ann}	average velocity in sheared annulus where shearing of a yield stress fluid takes place in a pipe (m/s)

Subscripts

3	slatter
Ann	annulus
exp	experimental
f	friction
O	pipe wall
v	valve
Ω	fully open position

found that the preliminary results compared well with Hooper (1981) in laminar flow, and the loss coefficient given by Perry (1997) in turbulent flow. In order to move loss coefficients for diaphragm valves to class 2 or 1 (Miller, 1990) from class 3, Fester et al. (2007) have tested a set of Natco diaphragm valves using non-Newtonian fluids. They provided loss coefficient data for 5 different sized valves in both laminar and turbulent flow. However, valves were only tested in the fully open position.

The objective of this study was firstly to experimentally determine the loss coefficients for Saunders diaphragm valves ranging from 40 mm to 100 mm at different opening positions, for a range of Newtonian and non-Newtonian materials, and compare it to that of Natco valves. The second objective was to extend Hooper's correlation for the determination of loss coefficients to account for the valve opening.

1.1. *Definition and determination of the loss coefficient*

The loss coefficient is defined as the non-dimensionalised difference in the overall pressure between the ends of two long straight pipes when there is a valve installed, and when there is no valve (Miller, 1990).

The estimation of the head losses in a pipeline system requires knowledge of the frictional losses in the straight pipes as well as the losses encountered in different fittings such as straight-through diaphragm valves. The head losses in straight pipes can be determined by Eq. (1) (Massey, 1970):

$$H_f = \frac{4fLV^2}{D2g} \tag{1}$$

where H_f is the head loss, f is the fanning friction factor, V is the average velocity, D is the internal pipe diameter and g is the gravitational acceleration.

The head loss in a valve is expressed in terms of the velocity energy head from the energy equation

$$H_v = k_v \frac{V^2}{2g} \tag{2}$$

where H_v is the valve loss and k_v the loss coefficient of the valve.

The loss coefficient of the valve is given by

$$k_v = \frac{\Delta p_v}{1/2\rho V^2} \tag{3}$$

where Δp_v is the pressure loss in the valve.

In turbulent flow the loss coefficient is independent of the Reynolds number, but in laminar flow a hyperbolic relationship exists between the loss coefficient and the Reynolds number (Edwards et al., 1985):

$$C_v = k_v Re \tag{4}$$

where C_v is a characteristic of a specific valve including its dimensions (Edwards et al., 1985).

In many cases it was shown that a Reynolds number should be used that accounts for the viscous characteristic of the fluids (Edwards et al., 1985; Polizelli et al., 2003; Fester et al., 2007). A Reynolds number (Re_{mod}) that can be used for Newtonian fluids, power-law and Herschel–Bulkley fluids is given by (Chhabra and Richardson, 2008),

$$Re_{mod} = \frac{8\rho V_{ann}^2}{\tau_y + K(8V_{ann}/D_{shear})^n} \tag{5}$$

where Re_{mod} is the Reynolds number modified by Slatter (1996), ρ is the density of the fluid, V_{ann} is the average velocity in annulus, τ_y is the yield stress, K is the fluid consistency index and n is the flow behaviour index.

The rheological behaviour of these viscous fluids is often described by the Herschel–Bulkley equation as given in Eq. (6) (Chhabra and Richardson, 1999):

$$\tau_0 = \tau_y + K(\dot{\gamma})^n \tag{6}$$

where τ_0 is the shear stress at wall and $\dot{\gamma}$ is the shear rate.

2. Correlations for predicting loss coefficient

Hooper (1981) derived the two-K correlation and defined a dimensionless factor k_v as the excess head loss in a pipe fitting, expressed in velocity heads. k_v does not depend on the roughness of the fitting (or attached pipe) or the size of the system, but is a function of the Reynolds number and the exact geometry of the fitting and is given by:

$$k_v = \frac{K_1}{Re} + K_\infty\left(1 + \frac{1}{\mathrm{ID}}\right) \tag{7}$$

where K_1 is k_v for the fitting at Re = 1, K_∞ is k_v for a large fitting at Re = ∞ and ID the pipe internal diameter in inches. This correlation is not only useful for diaphragm valves but for different kind of valves and Polizelli et al. (2003) used this model to predict the friction losses through butterfly valves (fully, 10°, 20°, 40° and 60° open), plug valves (fully and half open), bends (45°, 90° and 180°), and a union, they found a good agreement between experimental and predicted results for power-law fluids.

Darby (1999) improved this correlation to account for the non-linear nature of scale-up of pipe fittings. The 3-K model is given in Eq. (8) where K_i is a fitting parameter determined experimentally, but is not provided for diaphragm valves:

$$k_v = \frac{K_m}{Re} + K_i\left(1 + \frac{K_d}{\mathrm{ID}^{0.3}}\right) \tag{8}$$

where K_m, K_i and K_d are constants for loss coefficient in the 3-K model.

ESDU (2004) used correlations available in the literature as well as information from valve manufacturers and suggests the following correlation for the calculation of the loss coefficient:

$$k_v = k' \times \alpha_1 \times \alpha_2$$

where k' is the turbulent flow loss coefficient (determined at Re > 10^4), α_1 is the correction factor for partial opening of valve and α_2 is the correction factor at low Reynolds number. These values are presented in the ESDU Item 69022 in graphs and software formats, which are available exclusively to subscribers to their software.

Fester et al. (2007) derived some equations that could predict the loss coefficient for both laminar and turbulent flow for the fully open position for Natco straight-through diaphragm valves ranging from 40 mm to 100 mm bore diameter and found different laminar loss coefficient for accurate design purposes for Reynolds between 10 and 1000 using Eq. (9):

$$k_v = \frac{C_v}{Re} + \lambda_\Omega \tag{9}$$

where k_v is the loss coefficient of the valve, C_v is the laminar flow valve loss coefficient constant and λ_Ω is the turbulent loss coefficient at fully open position.

Table 1 – Nominal turbulent flow loss coefficient (λ_Ω) for fully open position for various pipe sizes for Natco valves.

Bore valve diameter (mm)	Loss coefficient (λ_Ω) Natco
40	7.85
50	2.53
65	1.35
80	2.66
100	1.33

Finally in 2008, Mbiya derived the two-constant model (Eq. (10)) for prediction of head losses through diaphragm valves for different pipe sizes at various opening positions under different flow conditions. The experiments were conducted using Natco valves. It has not been established if the loss coefficients provided can also be used to predict losses in diaphragm valves produced by a different manufacturer. This will be done in this work. Table 1 gives the values of λ_Ω that should be used in Eq. (10) for the Natco valves

$$k_v = \frac{38.6}{D^{1.24}\sqrt{Re}\theta^2} + \frac{\lambda_\Omega}{\theta^2} \tag{10}$$

where D is the internal diameter of the pipe and θ is the opening position.

The objective of this work was therefore to evaluate the existing correlations and compare these with the experimental data obtained in Saunders valves and to ascertain if the same loss coefficient for different manufacturers for straight-through Saunders diaphragm valves can be used.

3. Methodology

A brief description of the instrumentation and the experimental procedure for measuring the pressure losses and the calculation of the loss coefficient is given in this section.

3.1. *Experimental apparatus*

The experimental rig consists of six lines of PVC pipes with diameters ranging from 50 mm to 110 mm. Each line is 25 m long and contains a test diaphragm valve. This length was chosen to allow a fully developed flow before and after each test valve. A schematic diagram of the valve test rig is shown in Fig. 1.

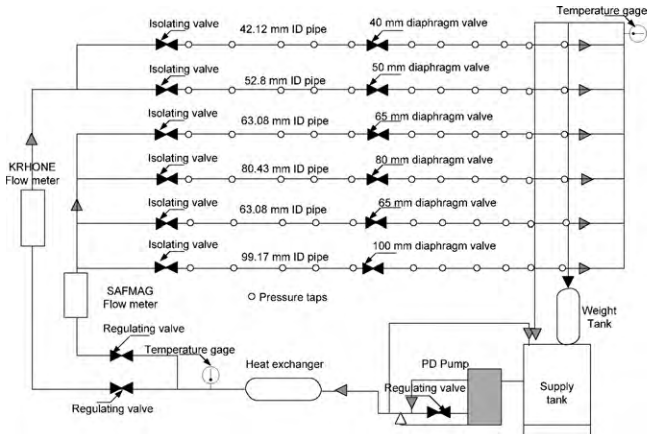

Fig. 1 – Schematic diagram of the valve test rig.

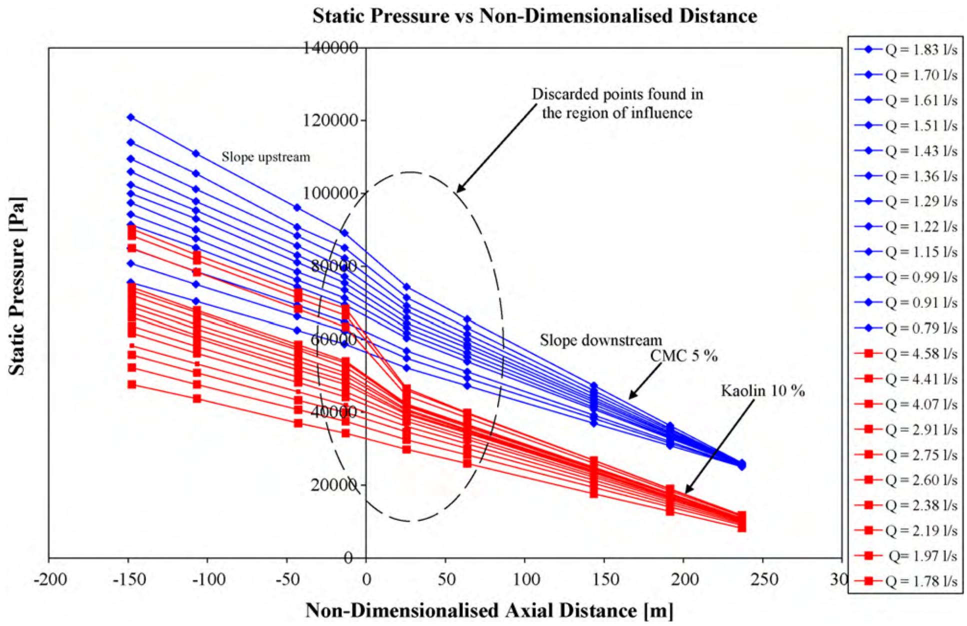

Fig. 2 – Hydraulic gradient line for a 42 mm ID pipe.

Test fluids were mixed in a 1.7 m^3 supply tank. The tank was rubber-lined to avoid chemical reactions of fluid with metal. The fluids were circulated in a continuous loop, as follows: from the supply tank, fluids were pumped out with a positive displacement pump before passing through a heat exchanger. The heat exchanger was followed by two valves coupled in parallel that directed the flow either to the upper part of the rig (which contained the smaller pipes of 42 mm and 50 mm ID) or the lower part (which contained the bigger pipes, 2×63 mm, 80 mm and 100 mm ID). Each of the two routes was fitted with a flow meter (A KHRONE flow meter of 50 mm ID and a SAFMAG flow meter of 100 mm ID). After the flow meters the fluids could enter any of the 6 test sections. An on/off valve was situated at the beginning of each line for isolation, so that only one line was tested at a time. After a fluid had passed a test section it was collected via a common pipe and directed to the supply tank. At the outlet it was possible to send the fluid through a weigh tank used for calibration purposes. Different pipe diameters ranging from 40 mm to 100 mm bore diameter of Saunders diaphragm valves were used in this experimental investigation. All the valves were positioned horizontally. Under steady state conditions the liquid flow rates were recorded at various flow rates. The percentage openings of the valves used in the experiment were full, 75%, 50% and 25% open.

Each pipe was fitted with four pressure transducers before the test valve and five after the test valve to measure the static pressure at different points along the test line.

A Fuji Electric differential pressure transducer (DP cell) with a maximum range of 130 kPa with an accuracy of 0.25% and a Fuji Electric point pressure transducer (PPT) of the same range were used to measure the pressure drop. The latter was used to measure the point pressure less than 130 kPa on the line while the differential pressure transducer was utilised when the point pressure exceeded 130 kPa. Fig. 2 presents the hydraulic gradients for kaolin 10% and CMC 5% in the 42 mm nominal diameter at various flow rates.

For each pipe the non-dimensionalised axial distances and the distances from the first point were recorded as illustrated in Table 2. The negative points for the non-dimensionalised axial distance refer to the points before the test valve.

3.2. Test diaphragm valves

In order to achieve our objectives, straight-through Saunders diaphragm valves were used for this investigation. A picture of

Table 2 – Pressure taps locations.

ID pipe (mm)	Pod 1	Pod 2	Pod 3	Pod 4	Pod 5	Pod 6	Pod 7	Pod 8	Pod 9
42									
Axial distances (m)	−6.25	−4.52	−1.83	−0.57	1.08	2.68	6.05	8.07	9.97
Distances [m]	0.00	1.73	4.42	5.68	7.32	8.93	12.29	14.32	16.22
50									
Axial distances (m)	−6.57	−3.53	−2.28	−0.78	1.26	3.01	6.57	8.57	10.57
Distances [m]	0.00	3.05	4.29	5.79	7.83	9.59	13.14	15.14	17.14
63									
Axial distances (m)	−6.97	−4.89	−2.89	−0.94	0.99	1.97	2.94	3.92	4.86
Distances [m]	0.00	2.09	4.09	6.04	7.96	8.94	9.91	10.89	11.83
80									
Axial distances (m)	−6.42	−4.01	−2.41	−1.21	1.91	3.91	5.98	8.46	9.96
Distances (m)	0.00	2.41	4.01	5.21	8.32	10.32	12.39	14.88	16.37
100									
Axial distances (m)	−5.55	−4.04	−2.54	−0.70	0.70	1.50	2.50	3.50	4.50
Distances (m)	0.00	1.51	3.01	4.85	6.25	7.05	8.05	9.05	10.05

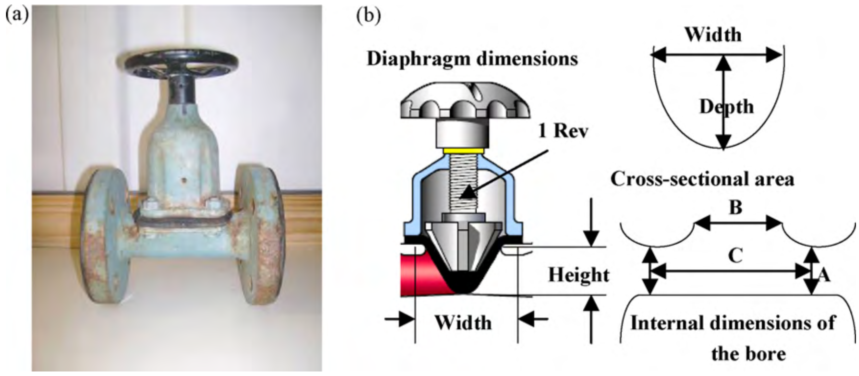

Fig. 3 – 50 mm bore diameter diaphragm valve (a) and internal dimension of the Saunders diaphragm valve (b) (Saunders valve, 2006).

Table 3 – Internal valve dimensions for various bore sizes.

Valve size (mm)	Cross-section area (mm)		Diaphragm dimensions (mm)			Bore dimensions (mm)		
	Depth	Width	Height	Width	Per Rev	A	B	C
40	35.26	42.78	36.00	47.38	3.44	28.20	54.06	90.04
50	46.65	64.26	47.00	66.34	3.88	35.26	67.15	133.4
65	62.42	90.82	63.00	92.14	3.64	51.98	82.36	152.5
80	68.92	112.00	69.00	114.20	2.98	58.64	118.5	171.4
100	74.72	124.46	75.00	129.92	2.78	59.56	126.7	262.6

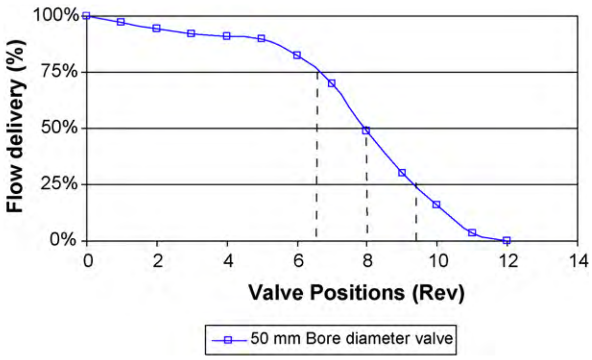

Fig. 4 – Determination for the opening position of the 50 mm bore diameter valve.

the 50 mm bore diameter diaphragm valve is given in Fig. 3a. The internal sizes and specifications of the test diaphragm valves are directly related to the flow path and are given in Table 3 and shown in Fig. 3b.

3.3. *Determination of opening position*

The valve positions were determined for the 25%, 50%, 75% and fully open positions using a gravity flow system at constant head where the flow delivery of water was measured at one revolution travel intervals of the hand-wheel of the valve. The 50% open position, for instance, was determined by counting the number of revolutions of the hand-wheel at which the water flow rate was 50% of the flow rate when the valve is fully open. This can be seen in Fig. 4. Fig. 5 shows the actual available flow area for the 50 mm bore diameter valve at these opening positions.

3.4. *Test fluids*

In order to conduct this investigation, a Newtonian fluid (water) and non-Newtonian fluids (kaolin slurry mixed in volumetric concentrations of 6%, 10% and 13%, and the concentration of CMC solution was 5% by mass) were used. The rheological properties were determined using in-line tube viscometry. Verification of the rheological properties was done daily to account for the changes of viscous properties.

The Herschel–Bulkley model was used to characterise the fluids. Typical rheological characteristics of the fluids used are given in Table 4.

3.5. *Experimental procedure*

The fluids were maintained at a temperature between 25 °C and 30 °C while using the positive displacement pump. The measurement of static pressure at different points upstream (US) and downstream (DS) of the test valve was taken. In total nine points were used, four upstream and five downstream of the test valve. The three points close to the test valve, one point upstream and two points downstream, were discarded because they are in the region of influence of the valve as presented in Fig. 2.

The slope and intercept were determined upstream and downstream using the selected data points in the fully developed region. Table 5 shows that the difference in the slope

Table 4 – Rheological characteristics of test fluids.

Fluids	Concentration (%)	Density (kg/m^3)	τ_y (Pa)	K' (Pa s^n)	n'
Kaolin	6	1103.6	3.071	2.038	0.264
Kaolin	10	1169.4	8.965	7.098	0.175
Kaolin	13	1215.5	18.97	16.14	0.242
CMC	5	1026.8	0	1.542	0.645

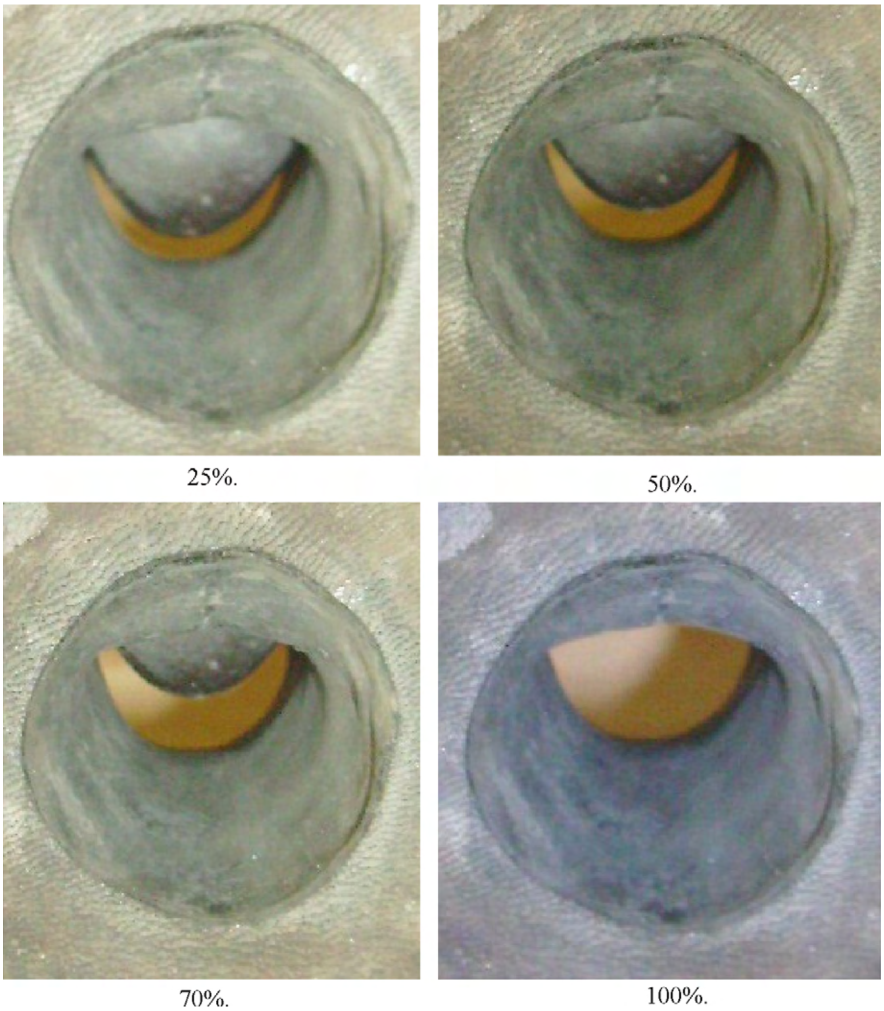

Fig. 5 – Opening position for 50 mm bore diameter for Saunders valve.

did not exceed 4% for the range of materials and flow rates between the hydraulic grade line before and after the test valve as tested.

The pressure was calculated at the centre-line of the valve, both upstream and downstream. The pressure drop across the valve (Δp_v) was determined by subtracting the downstream pressure from the upstream pressure at the centre-line of the valve as illustrated in Fig. 6. The loss coefficient, k_v, was then calculated using Eq. (3). The loss coefficient, C_v, is defined as the hyperbolic constant as given in Eq. (4) and was evaluated using the logarithmic least square uncertainty.

Table 5 – Percentage uncertainty between upstream and downstream slopes.

CMC 5%		Kaolin 10%	
Flow rate (l/s)	Uncertainty (%)	Flow rate (l/s)	Uncertainty (%)
1.83	4	4.58	3
1.7	4	4.41	1
1.61	2	4.07	1
1.51	2	2.91	1
1.43	3	2.75	1
1.36	2	2.60	2
1.29	1	2.38	2
1.22	1	2.19	0
1.15	2	1.97	0
0.99	1	1.78	0
0.91	2	1.47	0
0.79	3	1.24	1
		0.90	4
		0.63	0
		0.36	1
		0.18	0

C_v is solved by an iterative process in minimising the LSE for all different valve sizes

$$\mathrm{LSE} = \mathrm{Min} \sum \left(\ln \frac{C_v}{\mathrm{Re}} - \ln\, k_{v\,\mathrm{exp}} \right)^2 \tag{11}$$

where $k_{v\mathrm{exp}}$ is the experimental loss coefficient of the valve.

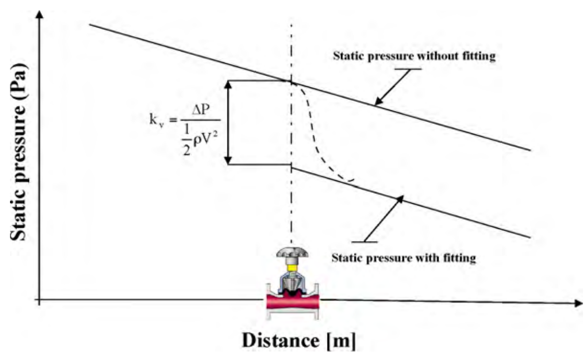

Fig. 6 – Definition of loss coefficient.

Table 6 – Experimental uncertainty for 50 mm bore diameter (expressed as %).

Opening position	Fluid	$\Delta Q/Q$	$\Delta\rho/\rho$	$\Delta(\Delta Pv)/\Delta Pv$	$\Delta D/D$	$(\Delta kv/kv)$Exp	$\Delta Re/Re$	$(\Delta kv/kv)$Calc
25% Open	Kaolin 10%	0.674	0.003	5.937	0.32	6.291	2.984	6.221
50% Open	Kaolin 10%	1.761	0.003	9.701	0.32	6.983	4.236	10.40
75% Open	Kaolin 10%	1.074	0.003	22.22	0.32	23.37	2.901	22.36
100% Open	Kaolin 10%	0.387	0.003	48.48	0.32	48.05	1.942	48.51

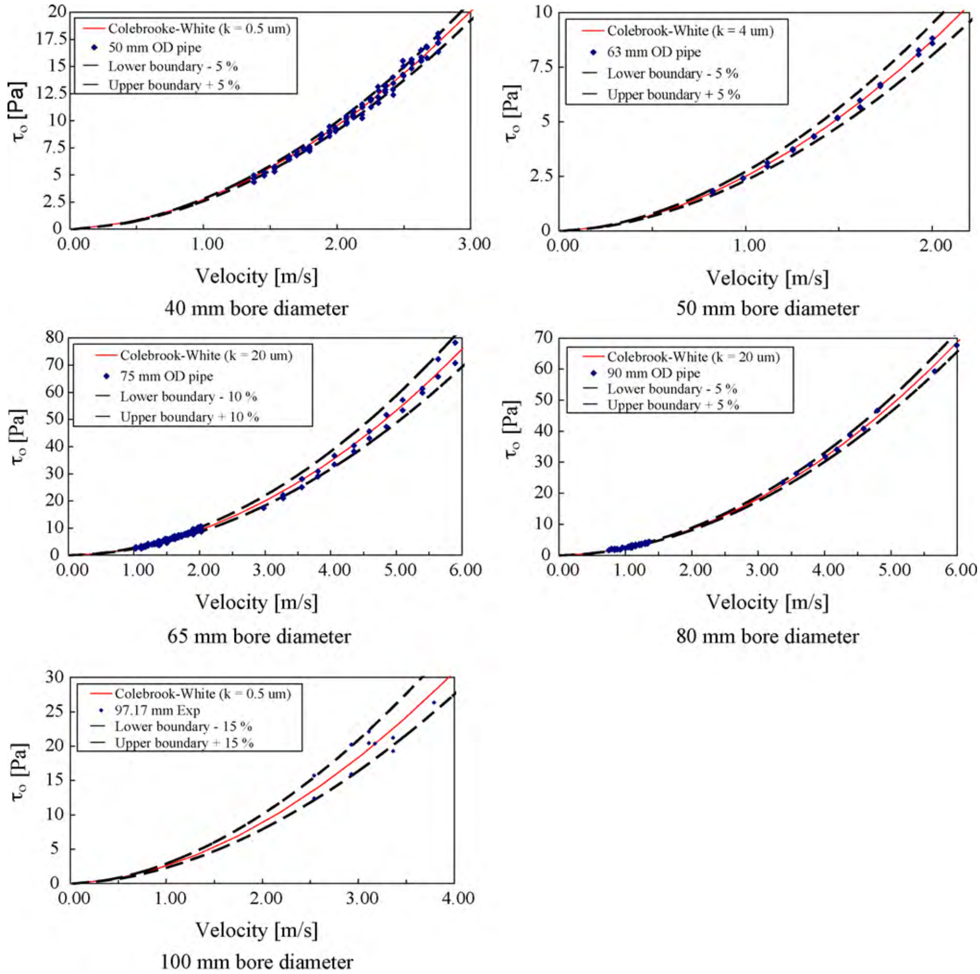

Fig. 7 – Comparison of water tests with Colebrook and White for all pipe sizes.

3.6. *Experimental uncertainty*

Mbiya (2003) derived an equation for estimating the experimental uncertainty for the loss coefficient as follows:

$$\left(\frac{\Delta k_v}{k_v}\right)^2 = \left(\frac{\Delta(\Delta P_v)}{\Delta P_v}\right)^2 + \left(\frac{\Delta\rho}{\rho}\right)^2 + 4\left(\frac{\Delta Q}{Q}\right)^2 + 16\left(\frac{\Delta D}{D}\right)^2 \quad (12)$$

where ΔP is the total pressure loss in the valve and Q is the volumetric flow rate.

The relative uncertainty obtained in this experimental investigation falls within the predictable uncertainty ranges as presented in Table 6.

4. Results

This section provides the experimental results found in this work and then compares them to those available in literature.

4.1. *Comparison of straight pipe data with theory in laminar and turbulent flow*

Fig. 7 shows a comparison of experimental results with the Colebrook & White equation for all the pipes tested. The pipe roughness was determined by measuring the pressure drop across a known length of pipe and by comparing it with the Colebrook and White equation (King, 2002). The surface rough-

Table 7 – Surface roughness for various pipe sizes.

Nominal diameter (mm)	Internal diameter (mm)	Surface roughness (μm)	Uncertainty (%)
40	42.12	0.5	4
50	50.80	4	1
65	63.08	20	4
80	80	20	4
100	99.17	0.5	9

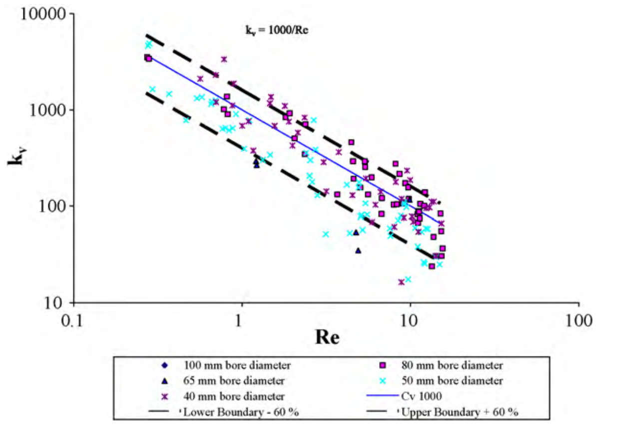

Fig. 8 – Loss coefficient versus Reynolds number less than 10.

Table 8 – Uncertainty for laminar loss coefficient C_v at various opening position for different pipe sizes.				
Bore valve diameter (mm)	Opening position (%)			
	25	50	75	100
40	31	37	0	23
50	5	11	44	30
65	170	40	52	55
80	6	23	1	5
100	17	29	92	66

ness (k) of all the pipes is given in Table 7 and was less or equal to 20 μm, as specified for smooth pipes. The results obtained from different pipes revealed an experimental uncertainty of 5% in the 42 mm, 50 mm and 63 mm nominal diameter; 10% in the 80 mm and 15% in the 100 mm nominal diameter pipes.

A summary of the results are given in Table 7.

4.2. *Loss coefficient data obtained for valves tested in this experimental investigation*

The loss coefficient k_v has been determined using Eq. (3). The accuracy of the slope upstream and downstream are critical in the determination of this parameter and was illustrated in Table 5 that the difference in the slope did not exceed 4% over a range of flow rates tested.

4.3. *Laminar flow*

For Re < 10, the loss coefficient k_v is a function of Re only and can be predicted using $C_v = 1000$ as given by Hooper (1981). It can be seen in Fig. 8 that the laminar loss coefficient C_v does not depend on the valve size or opening positions. Table 8 shows the relative uncertainty found in this work for different valves at various opening position are less than 60% as predicted by Fester et al. (2007), except for the 65 mm ID for the 25% open position and in the 100 mm ID for the 75% and fully open position where it was higher.

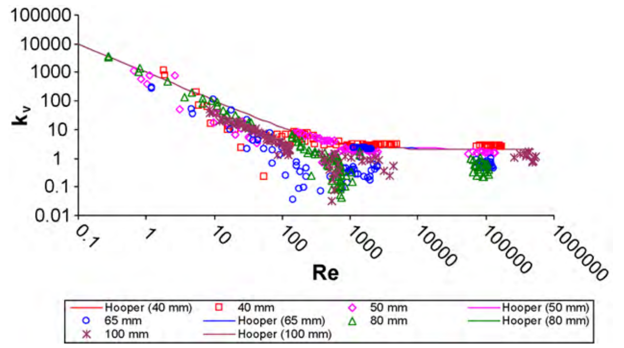

Fig. 9 – Comparison between Saunders data and the two-K method (Hooper, 1981) for various valve diameters at fully open position.

4.4. *Turbulent flow*

This paper provides the Saunders diaphragm valve loss coefficient as presented in Table 9. The loss coefficient increases with decreasing of the valve opening position. Except for the fully open position where the minimum is reached in the 80 mm pipe diameter, a qualitative trend is perceived for the other opening positions where the minimum is attained in the 65 mm pipe diameter and rises again.

5. Comparison between new data and literature

5.1. *Comparison between Saunders data and the two-K method derived by Hooper (1981)*

The two-K method agrees well for the small pipe diameter data, but deteriorates with increasing valve diameter as shown in Fig. 9.

5.2. *Comparison between Saunders data and the ESDU (2004)*

Fig. 10 shows that the ESDU correlation for the fully open position over predicts k_v over the range of valve diameters tested. ESDU defines the opening position as the number of handwheel turns (regardless of the percentage flow compared to fully open) and are therefore not used to predict losses at other opening positions.

Table 9 – Loss coefficient for Saunders diaphragm valves.								
Valve position (%)	25		50		75		100	
Bore diameter (mm)	k_v	Uncertainty (%)	k_v	Uncertainty (%)	k_v	Uncertainty (%)	λ_Ω Saunders	Uncertainty (%)
40	68.79	16	32.82	24	8.15	12	2.68	9
50	28.46	31	10.25	24	3.88	6	1.60	6
65	22.43	8	3.63	8	1.77	10	0.57	64
80	88.79	6	18.86	19	4.27	5	0.46	56
100	72.26	12	17.84	16	4.75	7	1.04	38

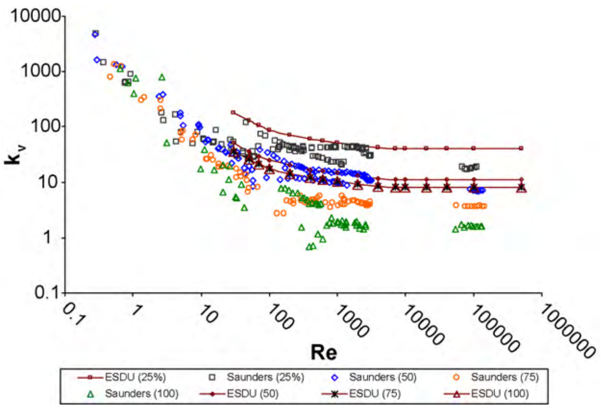

Fig. 10 – Comparison between Saunders data and ESDU (2004) for 50 mm bore diameter valve.

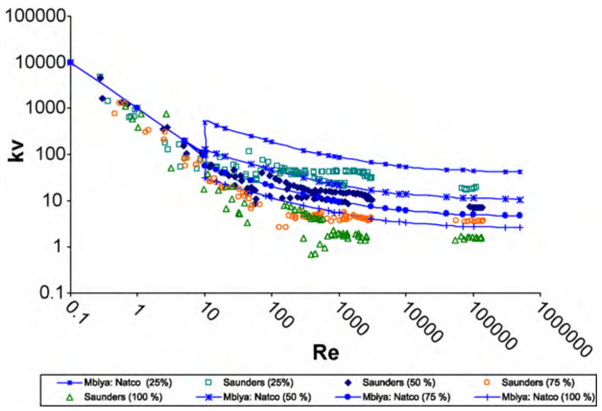

Fig. 11 – Comparison between Saunders data and Natco for 50 mm bore diameter valve.

5.3. *Comparison between Saunders data and two-constant model (Mbiya, 2008) using λ_{Ω} Natco*

Fig. 11 shows a comparison of loss coefficient data obtained in Saunders valves and Eq. (10). The data for λ_{Ω} are those obtained by Mbiya (2008) and is given in Table 1. The loss coefficient data obtained in the Natco diaphragm valves are higher than those found in this work for all different opening positions in the turbulent regime. In laminar flow, the prediction is almost similar and falls within the uncertainty margin of ±60%.

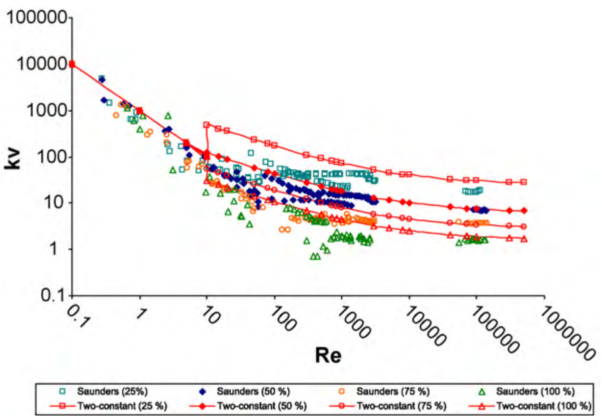

Fig. 12 – Comparison between Saunders data and the two-constant model for 50 mm bore diameter valve.

Fig. 12 shows a good agreement between the data obtained in this work and the two-constant model. However, this model performance deteriorates with increasing valve diameter and valve opening position.

Table 10 gives a summary of percentage uncertainty found between the data obtained from this investigation compared to the correlation found in the open literature. The deviations are very significant and this will in a poor prediction of pressure losses on a pipeline system.

Hooper (1981) derived a two-K method Eq. (7) to predict the loss coefficient through the dam diaphragm valve from laminar to turbulent for only the fully open position. The first term from Eq. (13) represents the viscous contribution to the loss as it can be seen in Fig. 6, and the second term is the contribution from inertia forces. A minimum value of 2 was found for the turbulent loss coefficient and 1000 for the laminar loss coefficient at Reynolds number equals to 1.

In 2007, Fester et al. have derived some equations that can predict the loss coefficient for both laminar and turbulent flow only for the fully opening position for Natco straight-through diaphragm valves ranging from 40 mm to 100 mm bore diameter and found different laminar loss coefficient for accurate design purposes for Reynolds between 10 and 1000 using Eq. (9).

6. New correlation

The new correlation builds and extends the two-K method derived by Hooper (1981) to include different opening positions. A relationship between the turbulent loss coefficient λ_{Ω} and the valve opening θ was derived. Fig. 13 shows that the plot of the opening position against the loss coefficient follows a power law trend for different bore diameter size. Due to the fact that the power law coefficient is more or less equal to the loss coefficient for the fully open position as it can be seen in Fig. 13, an average power constant of 2.5 has been calculated and taken into consideration from different bore diameter sizes to predict the turbulent loss coefficient for different bore diameter sizes and the coefficient is similar to the loss coefficient at the fully open position as seen in Table 11.

Benziger and Aksay (1999) stated that R^2 values above 0.90 represent good fits of experimental data. Table 12 gives us the power law trend for the prediction of the loss coefficient for different opening positions as well as the power law constant used in this work.

It can be seen in Fig. 13 that the coefficient is similar in value to the experimental loss coefficient at the 100% open position, and the latter is a function of the opening position

$$k_v = \frac{1000}{Re_3} + \frac{\lambda_{\Omega}}{\theta^{2.5}} \qquad (13)$$

Finally Eq. (13) represents the relationship to predict the loss coefficients at different opening positions for laminar and turbulent flow.

The new correlation Eq. (13) will be applied for Saunders straight-through diaphragm valves ranging from 40 mm to 100 mm in the fully, 75%, 50% and 25% open position.

Table 13 shows the loss coefficient (λ_{Ω}) that will be used in Eq. (13) in order to predict the head losses through the Saunders diaphragm valves and they are different to those found in the Natco diaphragm valves presented in Table 1.

Table 10 – Percentage error between Saunders data and literature.

Bore diameter (mm)	Opening position (%)	Miller	Hooper	Perry and Chilton	ESDU	Natco (Mbiya)
40	25	−94%	−97%	−69%		207%
	50	−96%	−94%	−87%		7%
	75	−90%	−75%	−68%		121%
	100	−70%	−25%	−14%	199%	202%
50	25	−86%	−93%	−26%		199%
	50	−88%	−80%	−58%		144%
	75	−79%	−48%	−33%		109%
	100	−50%	25%	44%	400%	56%
65	25	−82%	−91%	−6%		181%
	50	−67%	−45%	18%		341%
	75	−55%	13%	47%		58%
	100	40%	251%	304%	1286%	111%
80	25	−95%	−98%	−76%		−25%
	50	−94%	−89%	−77%		−5%
	75	−81%	−53%	−39%		59%
	100	74%	335%	400%	1422%	443%
100	25	−94%	−97%	−71%		38%
	50	−93%	−89%	−76%		63%
	75	−83%	−58%	−45%		111%
	100	−23%	92%	121%	592%	35%

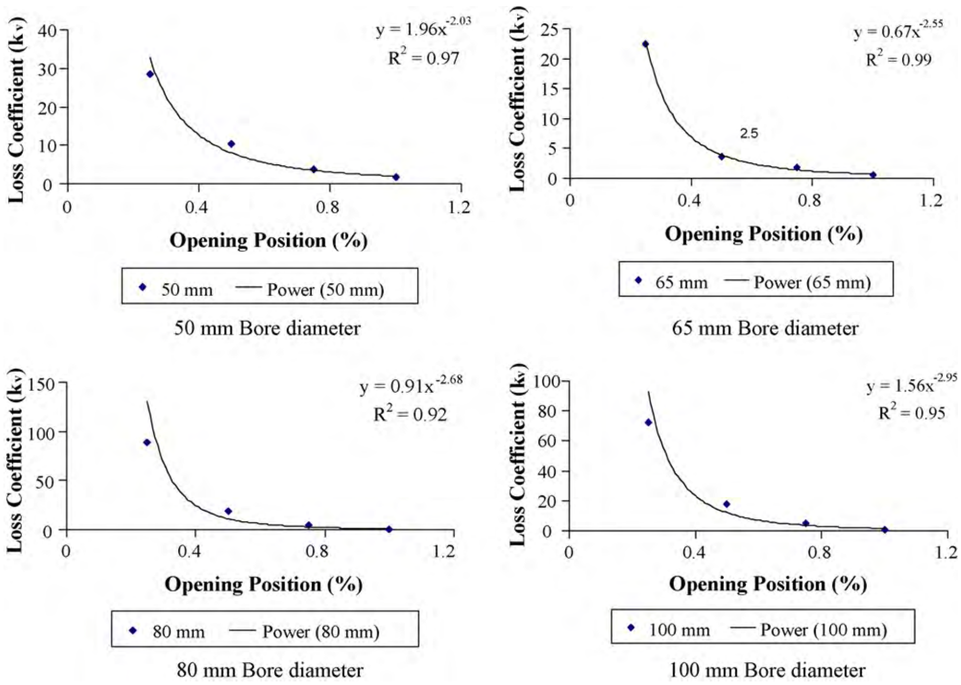

Fig. 13 – Opening position against the turbulent loss coefficient for various pipe sizes.

6.1. *Comparison between Saunders data and the new correlation*

The new correlation shows a good agreement of loss coefficient compared to the data obtained in the experimental investigation and deteriorates with decreasing opening position less than 50% as it can be seen in Fig. 14.

The relative uncertainty obtained in this work to evaluate the accuracy of the new correlation shows that the prediction uncertainty for the fully opening position was less than 2%

Table 11 – Power law fit for turbulent loss coefficient for various pipe sizes.

Nominal bore diameter	Power law trend	Power law constant	R^2
40	$k_v = 3.82/\theta^{2.30}$	2.30	0.91
50	$k_v = 1.96/\theta^{2.03}$	2.03	0.97
65	$k_v = 0.67/\theta^{2.55}$	2.55	0.99
80	$k_v = 0.91/\theta^{3.58}$	3.58	0.92
100	$k_v = 1.56/\theta^{2.95}$	2.95	0.95
Mean average		2.50	
Standard deviation		0.35	

Table 12 – General power law fit for turbulent loss coefficient.

Power law	Average	Standard deviation
$k_v = \lambda_\Omega/\theta^{2.5}$	2.5	0.35

Table 13 – Nominal turbulent coefficient (λ_Ω) for fully open position for various pipe sizes for Saunders.

Bore diameter (mm)	Loss coefficient (λ_Ω) Saunders
40	2.68
50	1.60
65	0.57
80	0.46
100	1.04

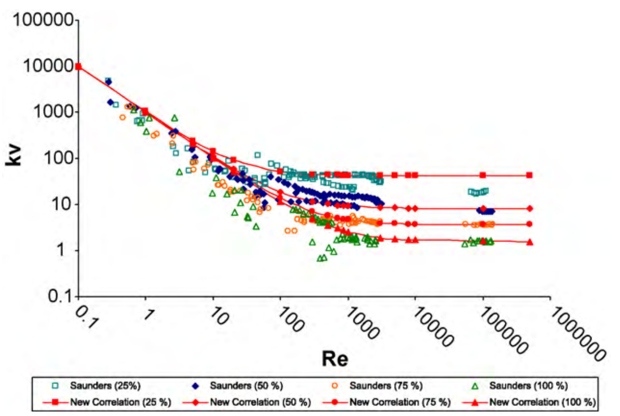

Fig. 14 – Comparison between Saunders data and the new correlation for a 50 mm bore diameter valve.

except for 50 mm internal diameter while for the 75% and the 50% opening position were within 6% and 75% depending on the pipe sizes and 40% for the 25% opening position and a poor prediction for the 100 mm internal diameter pipe (Table 14).

Table 14 – Percentage error between Saunders data and new correlation.

Bore diameter (mm)	Opening position (%)	New correlation
40	25	−39%
	50	−68%
	75	−42%
	100	0%
50	25	46%
	50	−20%
	75	−5%
	100	6%
65	25	40%
	50	17%
	75	−25%
	100	2%
80	25	−40%
	50	−74%
	75	−71%
	100	2%
100	25	354%
	50	4%
	75	−28%
	100	1%

The percentage uncertainty for predicting the loss coefficient for straight-through Saunders diaphragm valves using the new correlation is much better compared to those found in the literature.

7. Discussion

The objective of this study was firstly to experimentally determine the loss coefficients for Saunders diaphragm valves ranging from 40 mm to 100 mm at different opening positions, for a range of Newtonian and non-Newtonian materials. The second objective was to extend Hooper's correlation for the determination of loss coefficients to account for the valve opening. The third objective was to compare the results of Natco and Saunders diaphragm valves.

In laminar flow the loss coefficient constant was determined for all the valve sizes at Reynolds number less than 10, and increased with decreasing Reynolds number. This approach has been adopted due to the fact that Pienaar and Slatter (2004) showed that the Reynolds number range is significant when determining the laminar flow loss coefficient constant, C_v. Only data at Reynolds number less than 10 should be used to calculate C_v and will give the best fit at slope −1. Deviation of the loss coefficient results from this slope may start as early as Reynolds number 10 in some cases. Loss coefficient data that do not include Reynolds numbers less than 10 may result in a different slope (Banerjee et al., 1994). The statistical analysis of the data was done over the range of $0 < Re < 10$, and a good agreement has been found among different valve sizes as seen in Fig. 6. The laminar loss coefficient obtained in this work is more or less the same to that published by Hooper (1981) within 60% uncertainty.

In turbulent flow none of the data found in the open literature or the commercially available model from ESDU (2004) performed well over the wide range of conditions tested in this work, i.e. Reynolds number, valve opening position and valve size as can be seen in Figs. 7–10. The loss coefficients found in the Natco valves were higher than those found in the Saunders valves. This may be due to the more tortuous flow path of the Natco valve. A new correlation has been developed for prediction of the head losses through the Saunders diaphragm valves and performs well. In a nutshell, all the methods found in the open literature over-predict at high flow for various pipe sizes at different opening positions. The application of the two-constant model using the loss coefficient at fully open position predicts quite reasonably.

In conclusion, the same loss coefficient in turbulent flow for diaphragm valves from different manufacturers cannot be used for accurate design purposes and the new correlation should be used to predict the loss coefficient through diaphragm valves.

8. Conclusion

While predicting energy loss in straight pipe is a standard design task, most piping systems have valves and fittings that considerably complicate this task, particularly for laminar flows and non-Newtonian materials. The energy loss, as the sum of the losses of pipes and fittings, has to be calculated for each fitting, using a loss coefficient when applying the energy equation. This study has provided the loss coefficient of Saunders diaphragm valves at different opening positions, for a range of Newtonian and non-Newtonian materials that have been compared to the loss coefficient data for fully, 75%, 50%, and 25% open valve position for Natco straight-through diaphragm valves.

It was found that the loss coefficient suggested by Hooper (1981) is sufficient for low Reynolds numbers (Re < 10) for various opening positions and different valve diameters, and over predicted the turbulent loss coefficient. The same loss coefficients in the turbulent flow cannot be applied for more accurate designs for diaphragm valves from different manufacturers. A new correlation has been established to account for the prediction of head losses through diaphragm valves at various opening positions that is useful for design purposes.

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