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Theoretical Modelling of Rotary Valve Air Leakage for Pneumatic Conveying Systems

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ABSTRACT

Rotary valve air leakage is an important parameter for the design and operation of many dilute- and dense-phase pneumatic conveying systems. Previous methods to estimate air leakage have been largely empirical and have been found inaccurate and/or limited in their use. This paper describes a new theoretical model that has been developed to estimate rotary valve (radial clearance) air leakage. Model predictions are compared with experimental data obtained on a rotating and stationary valve subjected to a wide range of operating pressures typical of dilute- and dense-phase systems. Knowing accurately the air leakage present for any given situation, the required total supplied air mass flow rate can be determined properly for the pneumatic conveying system, so that optimal transport velocities can be established.

INTRODUCTION

The amount of rotary valve air leakage depends on many factors, such as system pressure, rotor clearances, material being handled, head of product above the valve and whether venting is used or not. Disregarding leakage at the design stage of a pneumatic conveying system can result in incorrect sizing of the prime mover (viz. fan, blower or compressor). An oversized prime mover can result in higher transport velocities, reduced solids throughput, increased abrasion and/or erosion of the plant, an increase in product degradation and/or unnecessary over-expenditure. An undersized prime mover can result in insufficient transport velocities causing increased operating pressures, system shutdown and/or pipeline blockages.[1]

Other consequences of air leakage include: fluidising the product above the rotary valve and hence, reducing its bulk density; hindering the flow of product into the valve. Either consequence will result in a reduction of the feed rate capacity of the valve. This situation is exacerbated for rotary valves that wear out over time, resulting in increased internal clearances and hence, air leakage.

For the above reasons, it is essential to know the amount of air leakage for the design and operation of pneumatic conveying systems. This paper presents results from investigations into measuring and theoretically modelling rotary valve leakage. Investigations have been limited to radial (rotor tip) clearance leakage, which is applicable to rotary valves fitted with (axial) end-plate seals, and rotary valves operating under air-only conditions. The latter situation is considered a worst-case (maximum leakage) scenario for industrial systems, where the:

- feed bin is almost empty, but product still needs to be conveyed through the pipeline;
- rotary valve operates as an air-lock (with an upstream pre-feeder or feed rate controller).

TYPES OF AIR LEAKAGE

Labyrinth leakage occurs if there is poor sealing between the rotor shaft and the side plates of the casing. This leakage can be minimised by using suitable seals.[2]

Carry-over leakage, Figure 1, occurs when compressed air is “carried” up into the feed hopper through the returning empty pockets of the rotary valve. This type of leakage can be reduced by venting air from the returning pockets before coming in contact with the feed material.[2] However, venting also will tend to increase the total amount of air leakage for a given valve and operating condition.

Clearance leakage, Figure 2, occurs when air leaks through the gaps between the moving rotor and the housing. To minimise this form of air leakage, small clearances can be used between the rotor and the valve housing. Manufacturers also have experimented with such things as spring-loaded, flexible and adjustable rotor tips[2] (i.e. to minimise rotor tip clearance).

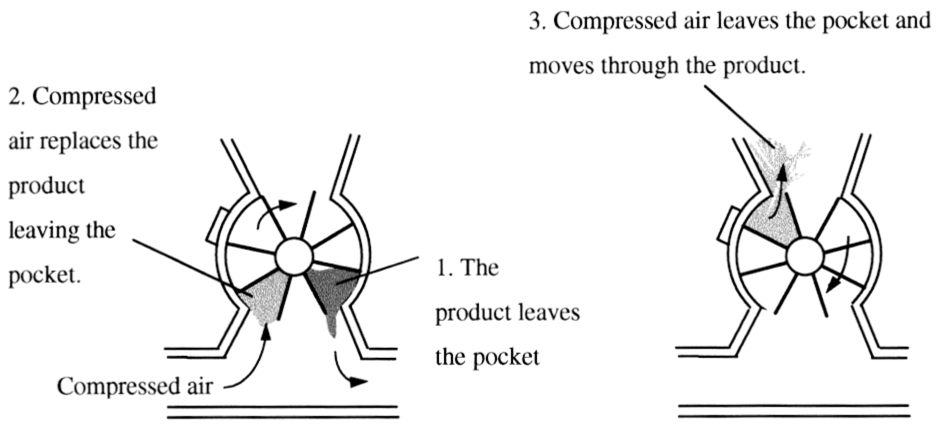

Figure 1 – Rotary valve carry-over air leakage.

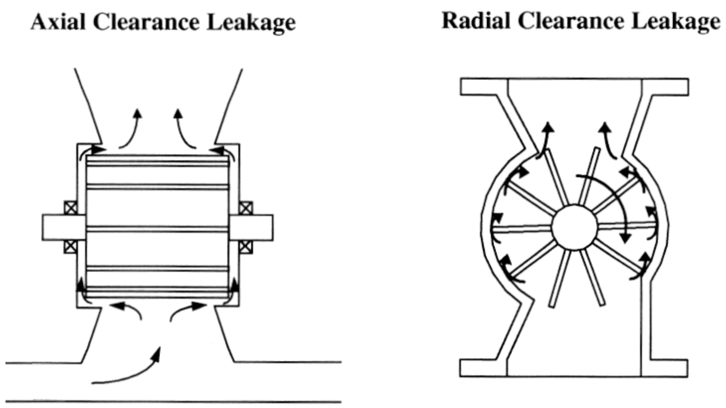

Figure 2 – Rotary valve axial (end plate) and radial (rotor tip) clearance air leakage.

Conventional rotary valves used for dilute-phase conveying (where operating pressures are typically < 100 kPa g) usually have both axial and radial clearances and hence, air leakage. Special high-pressure rotary valves used for dense-phase conveying (where operating pressures are typically < 350 kPa g) usually have axial seals and hence, only radial air leakage, which is the main focus of this paper.

EXISTING AIR LEAKAGE MODELS

Only a few methods are available that can be used to estimate rotary valve air leakage. Two common but fundamentally different approaches are summarised below. Note:

- each method applies to conventional rotary valves with axial and radial clearances;
- the second method is limited to a maximum pressure ratio of $P_1/P_2 = 1.7$, where P_1 = absolute pressure or air below the rotary valve (Pa abs) and P_2 = absolute pressure of air above rotary valve, usually atmospheric pressure (Pa abs);
- the first method does not stipulate an operating pressure limit, but this is expected to be 100 kPa g;
- some additional miscellaneous methods are discussed briefly later in the paper;
- other more empirical methods are available[3], but are not considered here.

Method by Marcus[2]

In estimating total air leakage (Q), Marcus[2] considered both carry-over leakage (Q_p) and clearance leakage (Q_c). All the relevant equations are summarised below. Note: P_2 usually is atmospheric pressure; Q_c is based on an orifice analogy; Q_p ignores the volume of the rotor. A possible improvement to the latter is to consider the actual swept volume of the rotary valve: $Q_p = \psi N$, where ψ = swept volume ($m^3\ rev^{-1}$) and N = rotor speed (rpm); this modification is considered later in the paper.

$$Q = \left(\frac{P_1}{P_2}\right)\left(Q_c + Q_p\right) \quad \text{“free” } m^3\ min^{-1} \text{ at pressure } P_2 \tag{1}$$

$$Q_c = 36\, F\, A_c \sqrt{\frac{2\,\Delta p_v}{\rho_1}} \quad \text{“actual” } m^3\ min^{-1} \text{ at pressure } P_1 \tag{2}$$

$$F = \frac{1}{3.5}\left(4.35 - \frac{P_1}{P_2}\right) - 0.2 \tag{3}$$

$$Q_p = \frac{\pi}{4} D^2\, L\, N \quad \text{“actual” } m^3\ min^{-1} \text{ at pressure } P_1 \tag{4}$$

where:

A_c = total clearance area (m^2); F = correction factor to allow for gas expansion; Δp_v = rotary valve pressure difference (Pa); D = rotor diameter (m); L = rotor length (m); ρ_1 = density of air below valve ($kg\ m^{-3}$).

Method of Reed et al.[4]

Reed et al.[4] developed a semi-empirical method to estimate rotary valve air leakage. During their tests, they found that carry-over leakage was negligible. Total air leakage was represented by the following equation.

$$Q = 0.001\ b\ v_l\ L\ c \quad \text{“free” } m^3\ min^{-1} \text{ (at 101 kPa and 15 ºC)} \tag{5}$$

where:
c = rotor clearance (mm); L = rotor length (m); b = blockage factor, which is taken from either Figure 3 (based on static conditions) or Figure 4 (based on dynamic or actual conditions); v_l = notional leakage velocity (m min^{-1}), which is taken from Figure 5. Note to apply the method of Reed[2] to higher operating pressures (i.e. $P_1/P_2 > 1.7$), extrapolation of the appropriate v_l curve would be required.

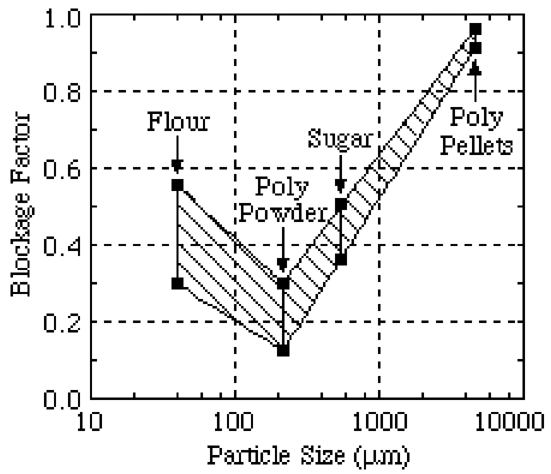

Figure 3 – Static blockage factor[4]

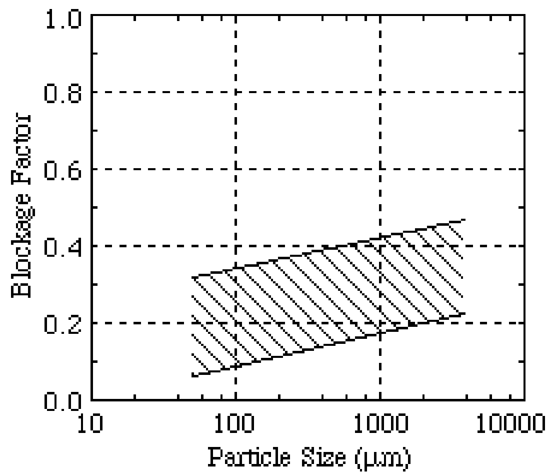

Figure 4 – Dynamic blockage factor[5]

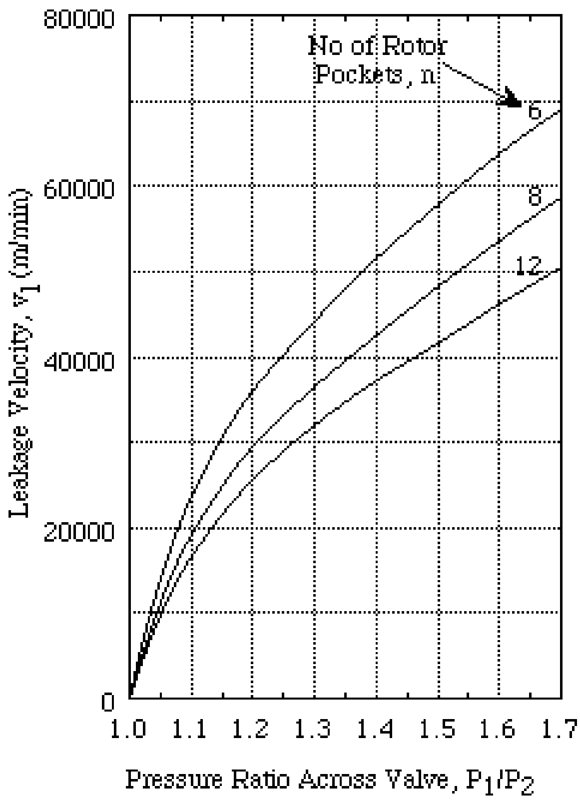

Figure 5 – Notional leakage velocity[2]

Miscellaneous Methods

In 1997, Rose[6] conducted a series of experiments to determine the loss of air through a single slit (simulating rotor tip clearance), where the test chamber had the same volume as one rotor pocket of a Waeschle ZGR-250 high-pressure rotary valve (with axial seals) that was available in the Bulk Solids Handling Laboratory at the University of Wollongong. The single-slit test rig developed and used by Rose is shown in Figure 6. Test chamber pressures were varied from 5 to 200 kPa g and the resulting air leakage was measured for various clearances ranging from c = 0.05 to 0.25 mm.

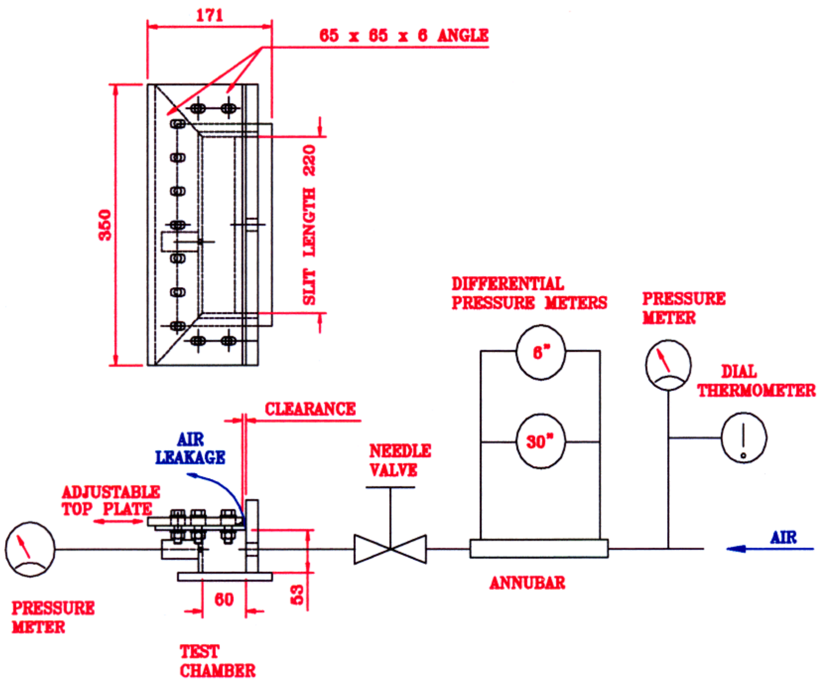

Figure 6 – Single-slit test rig to simulate rotor tip clearance[6]

Selecting the experimentally derived air leakage curve for a single slit of c = 0.189 mm (which apparently represented the clearance of the ZGR-250 valve at the time) and assuming three rotor blades in close proximity to the housing at any one time as well as a linear variation in pressure across the valve (i.e. from pocket to pocket), Rose[6] estimated the total leakage through the ZGR-250 valve.

In 1988, Reed et al.[7] suggested that the air leakage through a high-pressure rotary valve with axial seals (Buhler-Miag, Model OKFH-HP, Size 22/22) subjected to operating pressures up to 300 kPa g was approximately half that of the leakage from a conventional valve having identical dimensions.

COMPARISON OF METHODS

To compare all the previous methods, each one is used to estimate the total air leakage from a Waeschle ZGR-250 high-pressure valve (with axial seals) with the following characteristics and ambient conditions:

- Rotor diameter, D = 250 mm and Rotor length, L = 242 mm.
- Number of pockets, n = 10 and Radial clearance, c = 0.189 mm.
- Swept volume, ψ = 8 litres per rev and Rotor speed, N = 20 rpm.
- Air temperature, T = 293 K and Atmospheric air pressure, P_{atm} = 101.3 kPa abs.

Note the procedure used to apply the method of Reed et al.[7] (for high-pressure rotary valves) to the above valve was to: estimate total air leakage for a conventional rotary valve having identical dimensions (i.e. using the method of Reed et al.[4]); for a given operating pressure reduce this leakage by 50%. Note this method is limited to the maximum operating pressure of ≈70 kPa g (i.e. $P_1 \approx 171.3$ kPa abs or $P_1/P_2 \approx 1.7$, see Figure 5).

A comparison of all the previous methods trying to predict the same total air leakage for a given rotary valve is presented in Figure 7, which includes two additional curves: air leakage curve from the supplier of the valve; modified Marcus method incorporating rotor swept volume into the prediction of Q_p (as discussed previously). From Figure 7, it can seen that:

(a) for Δp_v < 100 kPa, all methods predict air leakage within an approximate envelope ±0.25 $m^3\ min^{-1}$;
(b) the uncertainty of air leakage increases for lower values of Δp_v (e.g. at Δp_v = 25 kPa, possible Q = 0.4 to 0.8 $m^3\ min^{-1}$; such a variation would be significant for a relatively small-diameter pipeline conveying system, say 50 mm NB);
(c) the differences between the models increase for Δp_v > 100 kPa;
(d) the method of Marcus[2] cannot be used for Δp_v > 125 kPa (probably due to exceeding the limits of the empirical equation for F);
(e) a new model obviously is needed to estimate the air leakage through high-pressure rotary valves.

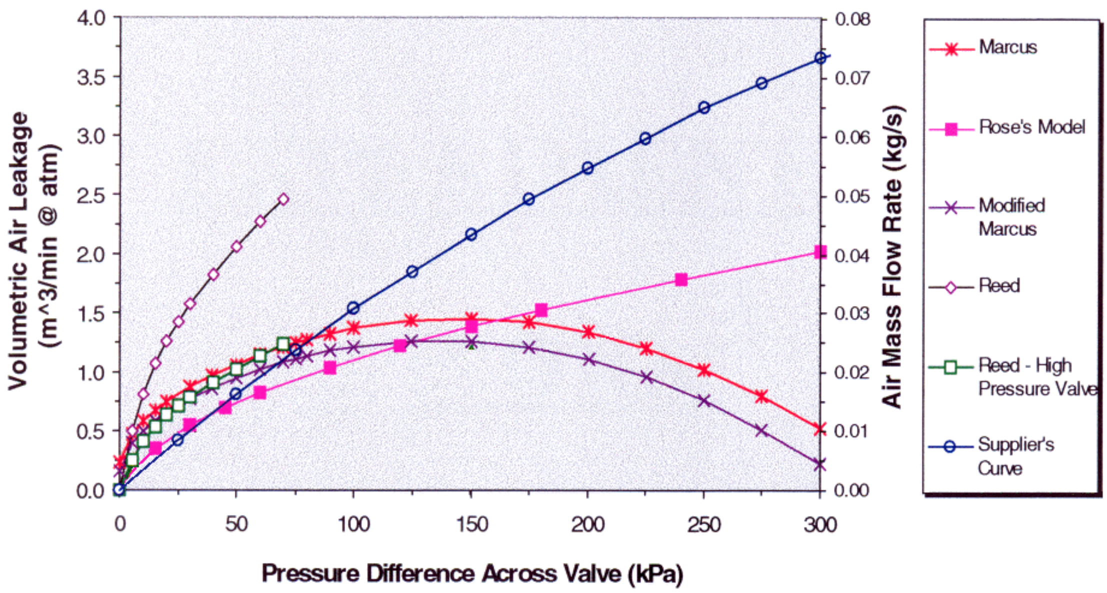

Figure 7 – Comparison of air leakage methods for a Waeschle ZGR-250 valve

MEASUREMENT OF AIR LEAKAGE

To compare/validate the existing methods and also assist in developing a new theoretical model, air leakage experiments were performed on a specially designed and built test rig.

Test Facility

Air leakage tests were performed on a Waeschle Model ZGR-250 high-pressure rotary valve, Figure 8. The test rig developed for this purpose is shown schematically in Figure 9.

A bank of six adjustable sonic nozzles was used to supply and control the air to the rig over a wide range of conditions. An inverted bin (chamber) was positioned above the rotary valve and used as an air reservoir to dampen the effects of the pulsating air travelling through the valve as it was rotating. An annubar was attached to the top of the bin and used to measure the flow rate of air travelling through the rotary valve.

Pressure meters were used to measure the static pressure above and below the valve and one also used to measure the static pressure at the annubar. Differential pressure meters were attached to the annubar to record the differential pressures caused by the various air flows.

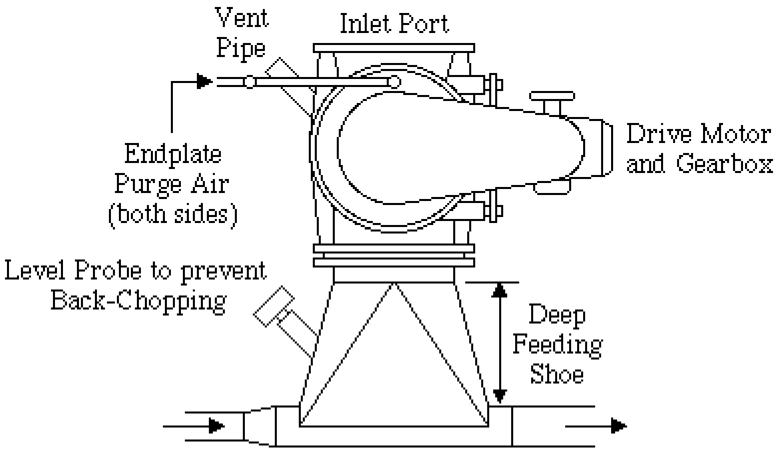

Figure 8 – High-pressure rotary valve used for air leakage experiments

Experimental Procedure and Results

Five sets of air-only tests were performed on the rotary valve, N = 0, 5, 15 and 30 rpm. For N = 0 rpm, the air leakage was found to vary between a minimum and maximum value, depending on the position of the rotor. By slowly rotating the rotor, it was found the number of rotor blades “in close contact” with the housing at any one time varied from 2 to 4 (i.e. 2 or 3 blades on the feed side of the valve; 3 or 4 blades on the return side of the valve).

The testing procedure involved creating a constant back pressure under the valve, then recording the pressures above and below the rotary valve and also the readings from the annubar. Using the relevant annubar equation, the air mass flow rate was calculated. This procedure was repeated for the necessary range of back pressures and each rotary valve speed.

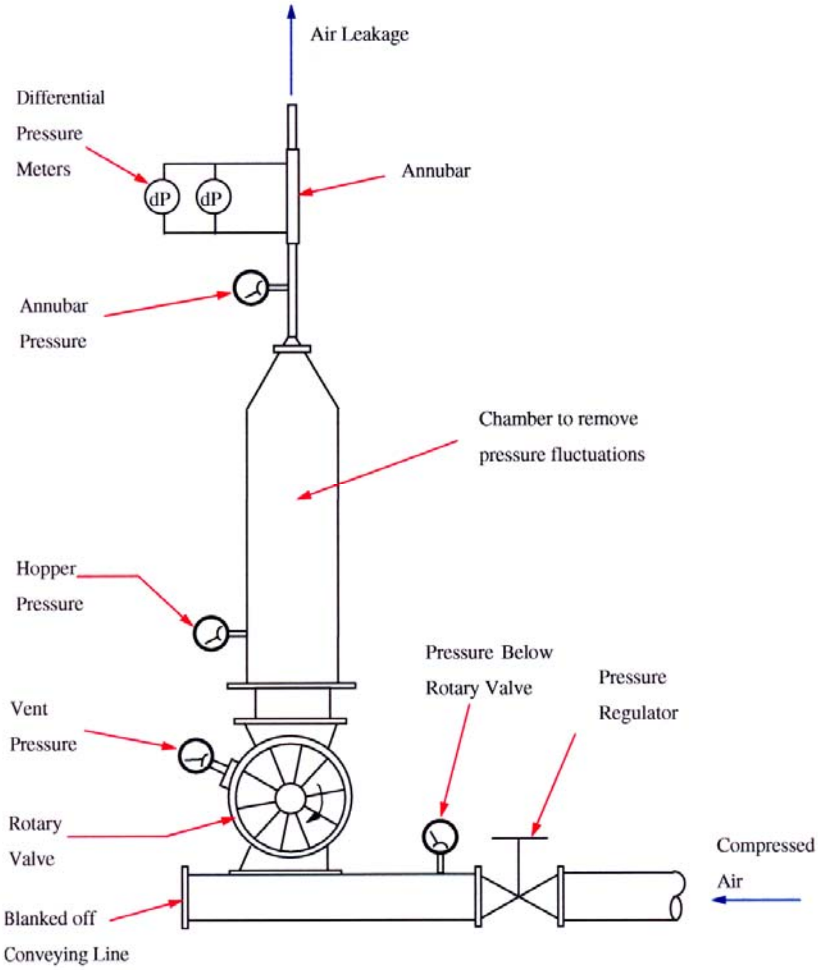

Figure 9 – Air leakage test facility

Once all testing was completed, the air mass flow rate of the air leakage was converted to volumetric flow rate at ambient conditions and plotted against rotary valve pressure difference. The resulting graph is shown in Figure 10. Note due to the rotor position effect described previously, the air leakage was found to fluctuate also for each rotor speed (especially for N = 5 rpm). The "Maximum Leakage" and "Minimum Leakage" curves shown in Figure 10 refer to the stationary rotor tests (i.e. N = 0 rpm), where the rotor position was adjusted by hand to achieve these two air flow conditions. An average air leakage was determined for each rotor speed experiment (viz. N = 0, 5, 15 and 30 rpm). Figure 10 shows clearly that the average air leakage does not change significantly over the range N = 0 to 30 rpm. This shows that carry-over leakage is insignificant for the ZGR-250 valve, and his result agrees with other research findings obtained on a conventional rotary valve[4], as well as a high-pressure rotary valve.[7] However, it should be stated that carry-over leakage is expected to be more significant with rotary valves having larger diameters and swept volumes.

To measure the pressure of air contained in a rotor pocket, the vent port of the valve was blanked off and a pressure meter connected to the blank flange – see Figure 9. The purpose here was to explore how the air pressure varies around the rotor and also the validity of the "linear" assumption made by Rose.[6] The vent pressure was recorded for N = 0, 5, 15 and 30 rpm. Note for N = 0 rpm, the rotor was positioned again to achieve maximum and minimum vent port (rotor pocket) pressure. The results are shown in Figure 11.

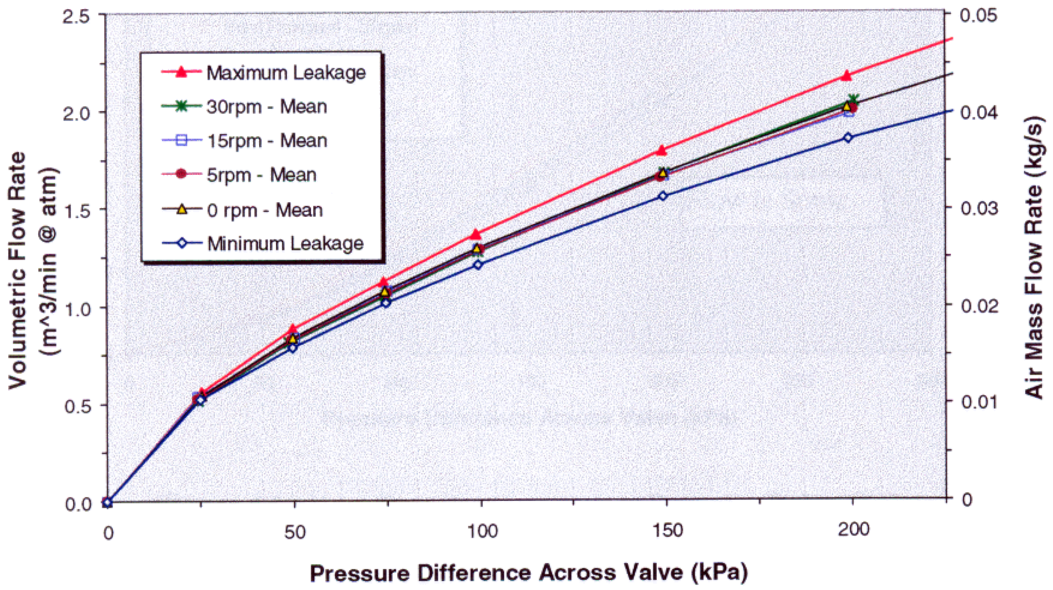

Figure 10 – Air leakage results for stationary and rotating rotary valve

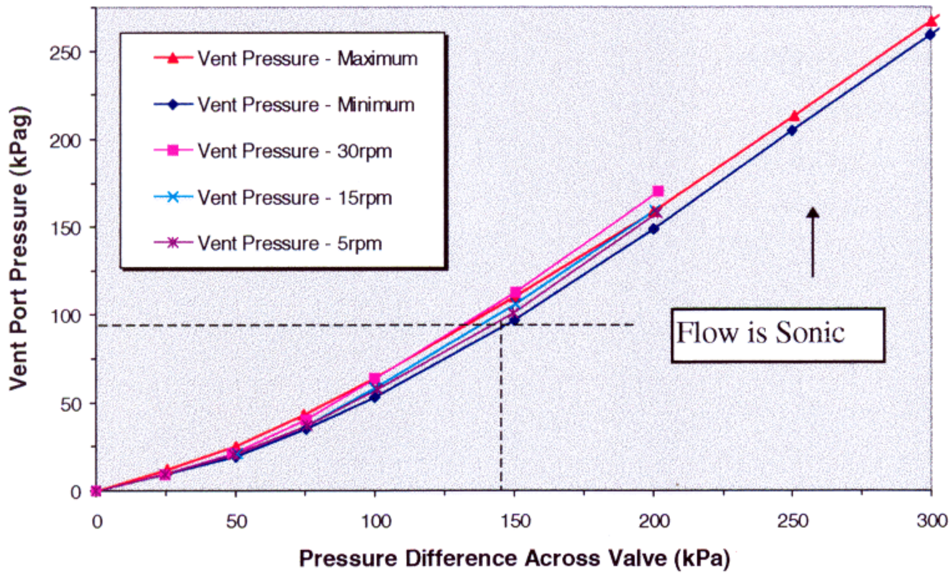

Figure 11 – Rotary valve vent port pressure for stationary and rotating rotary valve

After careful inspection of the rotary valve internal configuration and dimensions, it was found that only one rotor blade generally was in close contact with the housing between the vent port and valve inlet. Hence, the vent port pressure shown in Figure 11 essentially represents the pressure in the "last" rotor pocket (or the pressure drop across the "last" rotor blade). The surprising result here then is that the majority of Δp_v occurs across the last rotor blade. Hence, the pressure variation through the valve is not linear as was assumed by Rose[7].

Furthermore, the air passing through the last slit reaches sonic conditions when the pressure ratio across the last blade is equal to the critical value of P_{cr} = 0.5283. This occurs when the pressure at the vent port exceeds 101.3/0.5283 = 191.7 kPa abs = 90.4 kPa g. From Figure 11, it can be seen that the corresponding $\Delta p_v \approx 140$ kPa. Also, under sonic flow conditions, the vent port pressure is seen to vary linearly with the pressure below the rotary valve. Hence, the air leakage curves shown in Figure 10 actually should be a linear function for $\Delta p_v > 140$ kPa.

MODEL DEVELOPMENT

A long thin slit is basically an orifice and air leakage through a rotary valve can be represented by a series of slits or orifices (i.e. on the return and feed sides of the valve), as shown in Figure 12. The mass flow rate of air through a single orifice can be expressed[8] as:

$$m = \frac{k\,A_t}{\sqrt{1-(A_t/A_o)^2}}\sqrt{2\,\overline{\rho}\,(P_i - P_e)} \tag{6}$$

where:
m = mass flow rate of air (kg s^{-1}); k = orifice discharge coefficient; A_t = orifice throat area (m^2); A_o = area before or upstream of orifice (m^2); $\overline{\rho}$ = average density of air in slit (kg m^{-3}); P_i = absolute pressure before orifice (Pa abs); P_e = absolute pressure after orifice (Pa abs); T = average air temperature through orifice (K). Note Equation (6) can be applied to sonic flow conditions by introducing the critical pressure ratio, P_{cr}.

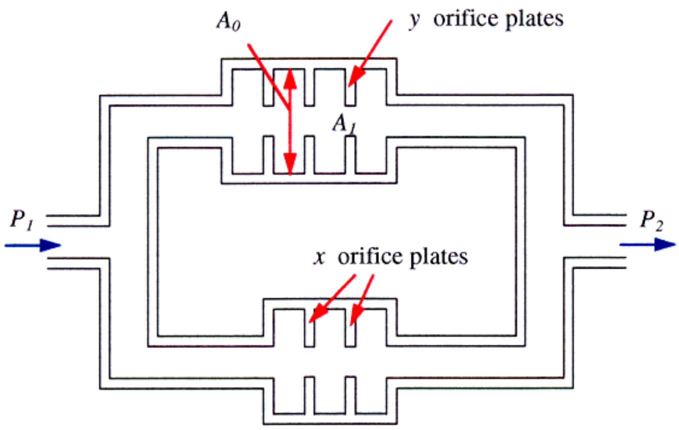

Figure 12 – Orifices in series and parallel representing rotary valve clearance

Applying Equation (6) to x orifices on the return side of the rotary valve and y orifices on the feed side of the valve and assuming k, A_t and T are constant, results in:

$$m = \frac{k\,A_t}{\sqrt{1-(A_t/A_o)^2}}\sqrt{\frac{\left(P_1^2 - P_2^2\right)}{R\,T}\left[\frac{x+y+2\sqrt{x\,y}}{x\,y}\right]} \tag{7}$$

where:
k = 0.86 for a thick square-edged orifice plate, as reported by Daugherty.[9] Validation of the new model for rotary valve air leakage was pursued by estimating and comparing theoretical air leakage with:

(a) the single-slit experimental data of Rose[6], by selecting x = 1 and y = 0 in Equation (7);
(b) the rotary valve experimental data presented previously in Figure 10;
(c) the high-pressure rotary valve leakage data presented by Reed et al.[7].

This effort resulted in Figures 13 and 14. Note for (b) above, it was decided to dismantle the rotary valve and measure the average radial clearance more accurately (i.e. by mounting the rotor on a lathe; using micrometers to determine the average rotor diameter and housing internal diameter). The average clearance was found to be c = 0.165 mm (and not c = 0.189 mm, as reported previously[6]).

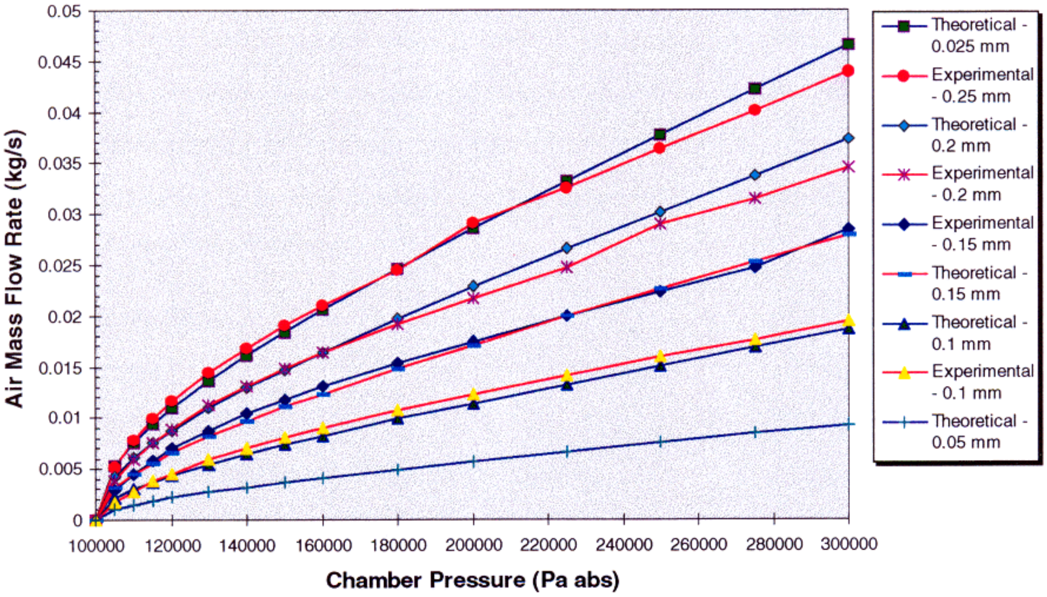

Figure 13 – Theoretical leakage compared with data from Rose.[6]

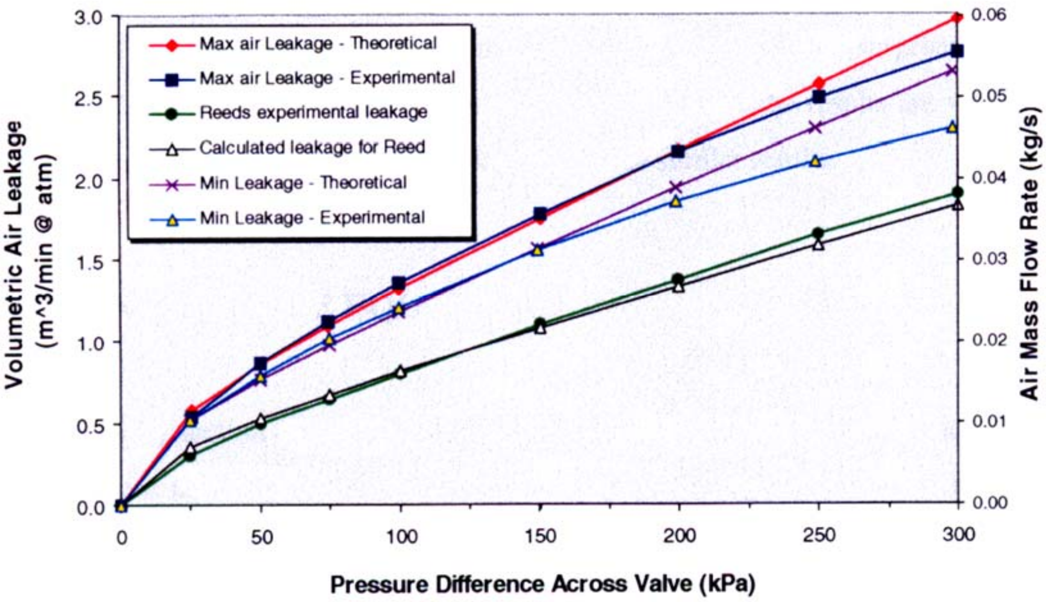

Figure 14 – Theoretical leakage compared with data from author and Reed et al.[6]
(x = 3 and y = 3 (min leakage); x = 2 and y = 3 (max leakage); $A_t = 39.9542\ mm^2$)

Some minor differences can be seen between the model predictions and the experimental results (e.g. minimum air leakage above Δp_v = 200 kPa g, Figure 14). However, for most cases, the agreement is quite close. Considering that the predictions shown in Figures 13 and 14 are based on a purely theoretical model (applied to single slits and actual rotary valves subjected to a wide range of pressures), the agreement with the experimental results is considered to be quite good – and certainly a step-change in improvement compared with previous methods.

CONCLUSIONS

Based on the results and findings presented in this paper, the following conclusions can be made.

(1) Rotary valve air leakage is an important issue that needs to be addressed in the design and operation of dilute- and dense-phase (positive-pressure) pneumatic conveying systems. This leakage can be significant for high-pressure dense-phase systems and even for dilute-phase systems, where relatively small diameters of conveying pipeline are employed.

(2) Comparisons between the existing methods used to determine rotary valve air leakage show marked differences, especially for operating pressures above 100 kPa g.

(3) The majority of rotary valve pressure drop occurs across the last blade of the rotor. Sonic flow conditions occur here when the ratio of absolute pressures reaches the critical value of P_{cr} = 0.5283.

(4) For the 200 to 250 mm diameter rotary valves considered in this paper, rotor speeds up to 30 rpm do not contribute significantly to total air leakage. That is, carry-over leakage is negligible and can be ignored.

(5) Assuming air-only conditions for the theoretical modelling of air leakage is considered a worst-case (maximum leakage) scenario for industrial systems, where: the feed bin is almost empty, but product still needs to be conveyed through the pipeline; the rotary valve is operated as an air-lock (with an upstream pre-feeder or feed rate controller). In the development of the model, rotary valve radial clearances are represented by an arrangement of orifices in series and in parallel, where it is required to know the number of blades in close contact with the housing for each side of the valve.

(6) The new air leakage model predicts well the single-slit and high-pressure rotary valve experimental results of other researchers, as well as the authors' own data.

Further experimental and development work is needed on larger rotary valves (i.e. to investigate further the issue of carry-over leakage), as well as axial clearance leakage. Such efforts will result in a more comprehensive air leakage model that can be applied to most positive-pressure pneumatic conveying systems.

ACKNOWLEDGEMENTS

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