SEALING OF ROTARY DRUMS FOR OPERATION UNDER PRESSURIZED CONDITIONS

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Abstract In practice, rotary drums are always designed for operation under vacuum conditions. In this paper, a novel technique is proposed for sealing the rotary drums under pressurized conditions. The proposed system is based on applying a secondary pressurized volume around the leaking gap of the drum. By controlling the pressure of this volume above the pressure of the drum, it will be possible to prevent from any leakage of gases to the ambient. The objective of a controller in this system is that the pressure of secondary volume be kept above the pressure of the drum in spite of the disturbances which may be exerted on the system by the wind outside the drum. The control system is also required to trace the variations in the drum pressure with the least fluctuations in the pressure difference among the drum and the volume.

Key Words Rotary Drum, Leakage, Controlled Sealing, Pressurized Operation

چکیده در عمل استوانه های دوار همواره برای کار در شرایط خلاء طراحی می‌شوند. در این مقاله تکنیک نوینی برای آببندی دستگاههای فرآیندی استوانه دوار تحت شرایط کاری فشاری عرضه شده است. سیستم عرضه شده بر مبنای ایجاد یک حجم ثانویه تحت فشار در اطراف محل نشتی استوانه می‌باشد. با کنترل کردن فشار این حجم کمی بالاتر از فشار داخل استوانه از هرگونه حداقل نشتی گازها به محیط بیرون می‌توان جلوگیری نمود. هدف کنترل در این سیستم این خواهد بود که فشار محفظه تحت فشار علیرغم اغتشاشاتی که ممکن است از بادهای بیرون استوانه به آن وارد شود همواره بالاتر از فشار استوانه نگاه داشته شود. همچنین لازم است که سیستم کنترل تغییرات فشار داخل استوانه را با کمترین نوسانات در اختلاف فشار بین استوانه و محفظه کنترل نماید.

1. INTRODUCTION

Rotary drums are used in some process equipment such as rotary drum driers, rotary drum coolers, rotary drum calciners, rotary cement kilns and etc. The sealing of rotary drums is one of the most important problems in the operation and maintenance of this equipment. Its effect on energy consumption and operational conditions of the equipment is momentous. As an outstanding industrial case in which the sealing of the equipment should be performed with the most sophisticated techniques, is the rotary cement kilns. One of the ways to seal rotary cement kilns is by using a graphite block air seal [1]. The graphite seal consists of 24 graphite blocks of suitable size (depending on the cement kiln diameter), mounted along the circumference of the kiln. Russian industries have used a triple labyrinth seal for wet process rotary cement kilns which was developed and employed in other processes later [1]. Beigel [2] presented a rotary kiln air seal for the discharge end of the kiln, with air cooling of the kiln nose ring which is exposed to the brunt of high temperatures. The seal is fastened to a cooling ring which is attached to the kiln shell. Cooling air is blown into the space between the outer ring and the kiln shell to prevent excessive temperatures.

In this paper, a new technique is proposed for preventing the leakage of gases from the inside of rotary drums during pressurized operation. In practice, this equipment is always designed for operation under vacuum conditions. The vacuum is provided by a draft fan, which intakes gases or air from inside the drum. Providing suitable gaps between the moving drum and the stationary ducts or parts, located at the two ends of the drum, is an indispensable configuration which creates the problem of leakage which should be prevented for the sake of efficient operation [1]. This problem had always forced industries to run the rotary drums in vacuum conditions, since it is not permitted to let leakage of inner gases to the outside during pressurized operating conditions. On the other hand, creating these conditions provides some benefits for chemical processes. These are the increasing of capacity in the drum, as well as the increasing of heat transfer coefficient, which will appear because of increasing in the density of the gases inside the drum; Moreover, there are some processes in chemical industries such as the steam processing of solid particles or some of the gas-solid reactions that are beneficial when accomplished, in pressurized conditions rather than in vacuum conditions. On the other hand, it may appear in the future other applications of this technique which are unknown to the author at the present time.

Creating a pressurized condition requires the pushing of gases inside the drum by a blower. Also, the loading of solid materials inside the drum may be accomplished by some suitable designs such as a compressed air lift technique.

The proposed sealing system for pressurized rotary drum works on the basis of creating a secondary pressurized volume around the leaking gap of the drum, as shown in Figure 1. The figure depicts the current condition of the rotary drum and the stationary solid input duct which intrudes the drum a few centimeters. In this way, the solid material will not penetrate outside the drum, provided that leakage from the gap among the drum and the duct towards the outside does not occur. The pressurized volume is built in the shape of a ring type around the gap. It is welded to the stationary duct from one side, making a gap with the rotary drum on the other side, so that the drum can rotate freely. The pressure inside the volume can be provided by a suitable compressor. Since the pressure of the drum is more than the outer pressure, the pushed air will find its way from the gap among the drum and the frame of the volume.

The ideal operating conditions for the system is when pressure difference among the drum and the volume becomes zero, so that any leakage will be ceased. But this ideal sustained condition can not be realized for a long period of time. This is due to the fact that there are always some input disturbances on the pressure of the drum and the secondary volume exert some variations on the pressure difference. Actually, there are two sources of disturbance:

1. Pressure variations inside the volume, resulted from variations in the ambient pressure, which may be caused from the wind outside the drum. This kind of disturbance is probable for due to the fact that the equipment is located in an open space.
2. Pressure variations inside the drum, are the results of variations in the operating condition of the drum, as well as during the shut-down and start-up of the equipment.

The existence of these variations brings the pressure controlling problem in the volume, such that it can reject the effect of wind disturbances, (tracing the variations in the pressure of the drum.) Since one of the controlling objectives is tracing and adjusting the pressure of the pressurized volume, therefore the variations in the drum pressure can be considered as the set point for the control loop. However, the ambient air pressure changes or those resulting from the wind are considered as disturbances to the control loop.

The mechanism of control as shown in Figure 2 includes the measuring elements of the pressure both for the drum and the pressurized volume, as well as the controller which receives the measurements, and actuates the flow rate of the blower.

Some small amount of leakage will appear through the gap among the drum and the volume, which depends on the pressure difference among them. It will occur as a result of the variations, affecting the pressure inside the drum and the disturbances affecting the volume pressure, as well as the actuating effects of the controller on the volume pressure. We may suppose that $\Delta P = P_v - P_d$; where ΔP can be positive or negative. As a result of controlling actions on the loop, it would appear some fluctuations in the pressure difference, ΔP. This pressure difference may be positive or negative. Thus, the controller

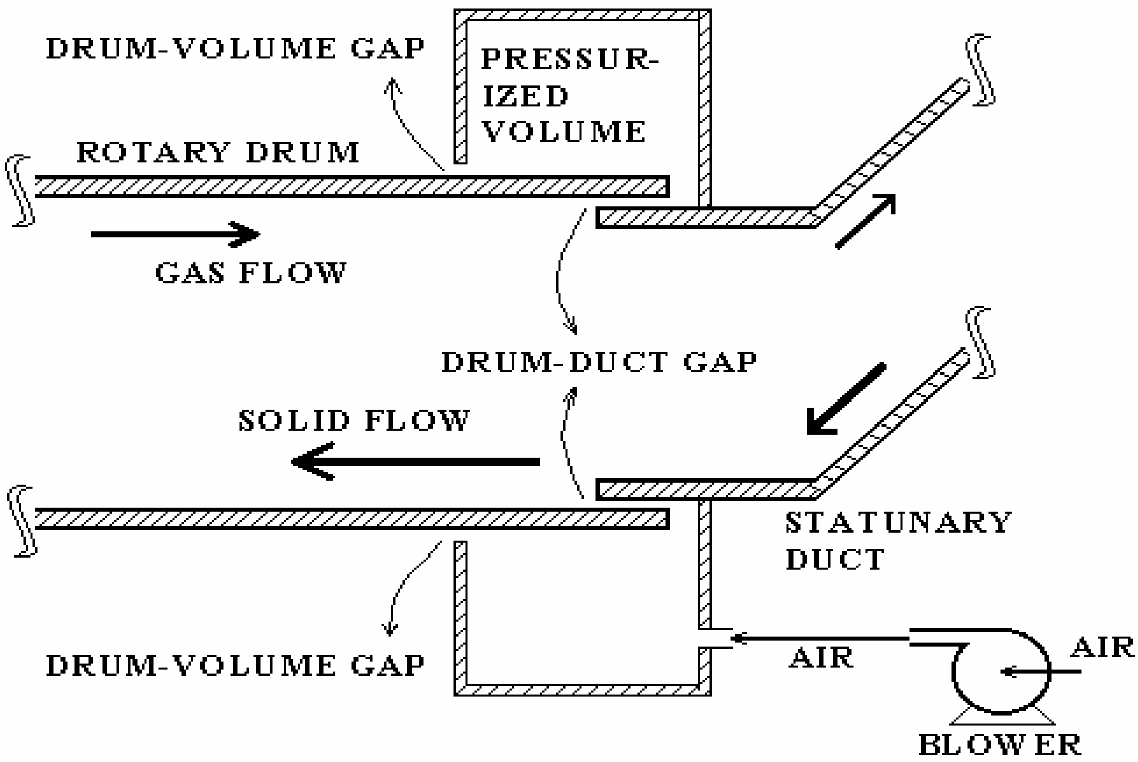

Figure 1. Schematic diagram of the ring type pressurized volume around the leaking gap of a pressurized rotary drum for the prevention of leakage to the outside.

should reject disturbances, tracing the variations in the pressure of drum with the least fluctuation in ΔP. According to the above discussions, it seems that a suitable architecture for the control system is that a small amount, $E_R \geq 0$, be chosen for the set point, such that $E_R = (P_v - P_d)_D$. It means that a controller keeps the pressure of the volume slightly above the drum pressure required, such that the leakage of air into the drum will be negligible in steady state conditions.

DYNAMIC MODEL

The dynamic model of the system is necessary for designing the required controller for it [3,4]. A nonlinear differential model can be obtained by using mass balance of the input-output streams. This model may be used directly for controlling the system by use of some available nonlinear control methods, or it may be used for obtaining the transfer functions describing the input-output relations between the variables of the system. Then, the transfer functions can be used for designing a suitable controller such as adaptive controllers. In this case, adaptive controllers are preferable due to the fact that the original differential model is nonlinear and the parameters of the transfer function model depend on the operating conditions. The dynamic model will include the variables P_a: ambient pressure (disturbance variable), P_d: drum pressure (disturbance or set point variable), P_v: volume

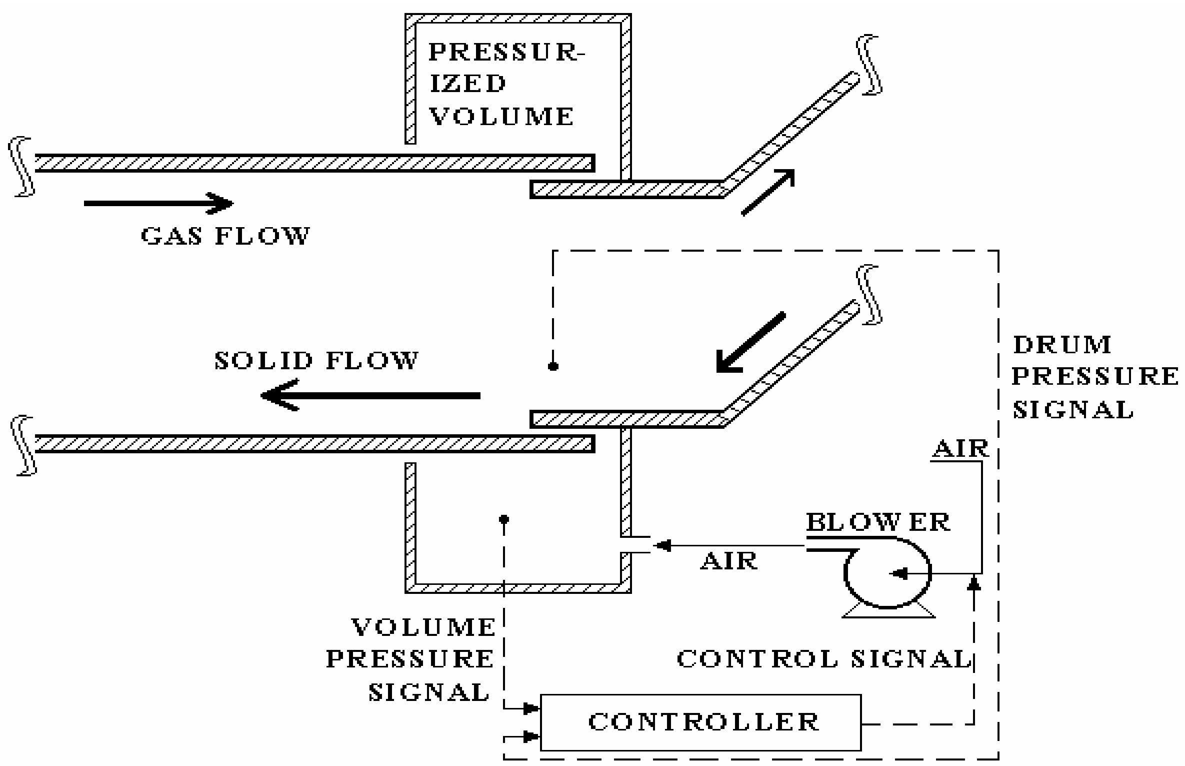

Figure 2. Schematic diagram of the control system for sealing the gaps of pressurized rotary drums.

pressure (controlled variable) and Q_b : blower flow rate (manipulating variable).

In order to obtain the dynamic model for the system, we use the relation for isothermal flow of a gas in the distance l of a channel [3].

$$\frac{M}{2RT}\left(P_1^2 - P_2^2\right) = \frac{G'^2}{g_c} \operatorname{Ln} \frac{P_1}{P_2} + \frac{G'^2 f.l}{2 g_c r_H} \quad (1)$$

The last term in the right hand side of the Equation 1 contributes to the friction effect of the flow. This term is much smaller in comparison with others, to a degree that it could be neglected by assuming $f = 0$. The parameter f is the friction factor of flow in the distance l of the gap depending on Reynolds number [5].

From Equation 1 the mass flux, G', is obtained as:

$$G' = \sqrt{\frac{M g_c \left(P_1^2 - P_2^2\right)}{2RT\left(\ln \frac{p_1}{P_2} + \frac{f.l}{2 r_H}\right)}} \quad (2)$$

For the system of sealing depicted in Figure.1, using mass balance on the volume, the differential model is obtained as:

$$\dot{m}_b(t) - \dot{m}_{va}(t) - \dot{m}_{vd}(t) = \frac{dm_v(t)}{dt} \quad (3)$$

Where $\dot{m}_{b}(t)$ is the mass flux of fresh air entering the volume via the blower, $\dot{m}_{va}(t)$ is the mass flux of air depleting to the ambient from the gap among the drum and frame of the volume and $\dot{m}_{vd}(t)$ is the mass flux of air entering the drum from the gap among the drum and duct.

The term $dm_v(t)/dt$ represents the rate of air accumulation within the volume. All mass fluxes as well as the pressures are time dependent.

The existing mass of the air in the pressurized volume and its differential is written in the terms of volume and pressure as the following:

$$m_v(t) = \frac{P_v(t) V_v M_a}{RT_v} \tag{4}$$

$$\frac{dm_v(t)}{dt} = \frac{V_v M_a}{RT_v} \cdot \frac{dP_v(t)}{dt} \tag{5}$$

The mass flow rate of air depleting from the pressurized volume may be written in the term of mass flux as:

$$\dot{m}_{va}(t) = G'_{va}(t) A_{va} \tag{6}$$

In the same way, the flow rate of air entering the drum from the pressurized volume can be written in terms of mass flux as:

$$\dot{m}_{vd}(t) = G'_{vd}(t) A_{vd} \tag{7}$$

The flow rate of the blower can also be written in terms of the pressure P_v as:

$$\dot{m}_b(t) = \rho_a(t) Q_b(t) = \frac{P_v(t) M_a}{RT_a} Q_b(t) \tag{8}$$

According to Equation 2, the mass fluxes of air from the gaps can be written as:

$$G'_{va}(t) = \sqrt{\frac{M_a g_c \left[P_v^2(t) - P_a^2(t) \right]}{2RT_v \left[\ln \frac{P_v(t)}{P_a(t)} + \frac{f_{va} l_{va}}{2r_{H,va}} \right]}} \tag{9}$$

$$G'_{vd}(t) = \sqrt{\frac{M_a g_c \left[P_v^2(t) - P_d^2(t) \right]}{2RT_v \left[\ln \frac{P_v(t)}{P_d(t)} + \frac{f_{vd} l_{vd}}{2r_{H,vd}} \right]}} \tag{10}$$

In both Equations 9 and 10, the temperature of fluid is considered as the temperature of the air inside the volume. Using Equations 9 and 10 in Equations 6 and 7 and also replacing 5, 6, 7 and 8 in 3, results as:

$$\left[\frac{V_v M_a}{RT_v} \right] \frac{dP_v(t)}{dt} = \frac{P_v(t) M_a}{RT_a} Q_b(t) - A_{va} \sqrt{\frac{M_a g_c \left[P_v^2(t) - P_a^2(t) \right]}{2RT_v \left[\ln \frac{P_v(t)}{P_a(t)} + \frac{f_{va} l_{va}}{2r_{H,va}} \right]}}$$
$$- A_{vd} \sqrt{\frac{M_a g_c \left[P_v^2(t) - P_d^2(t) \right]}{2RT_v \left[\ln \frac{P_v(t)}{P_d(t)} + \frac{f_{vd} l_{vd}}{2r_{H,vd}} \right]}} \tag{11}$$

The nonlinear differential Equation in 11 is the differential model of the system which can be used for controlling the sealing system, or it may be used after linearization for obtaining the transfer functions between the input-output variables.

The resulted transfer function matrix represents the input-output relations between the variables of the system, which can be obtained after linearization of the model. Extracted algebraic manipulations are given in Equation 12, while the details of them are presented in appendix A.

$$\Delta P_v(s) = \begin{bmatrix} \frac{K_{Pa}}{\tau s + 1} & \frac{K_{Pd}}{\tau s + 1} & \frac{K_{Qb}}{\tau s + 1} \end{bmatrix} \begin{bmatrix} \Delta P_a \\ \Delta P_d \\ \Delta Q_b \end{bmatrix} \tag{12}$$

The transfer functions in relation 12 represents a three-input system, involving ambient air pressure, rotary drum pressure and fan flow rate which affect the controlling output.

CONTROL

In Figure 3, a block diagram is shown for the

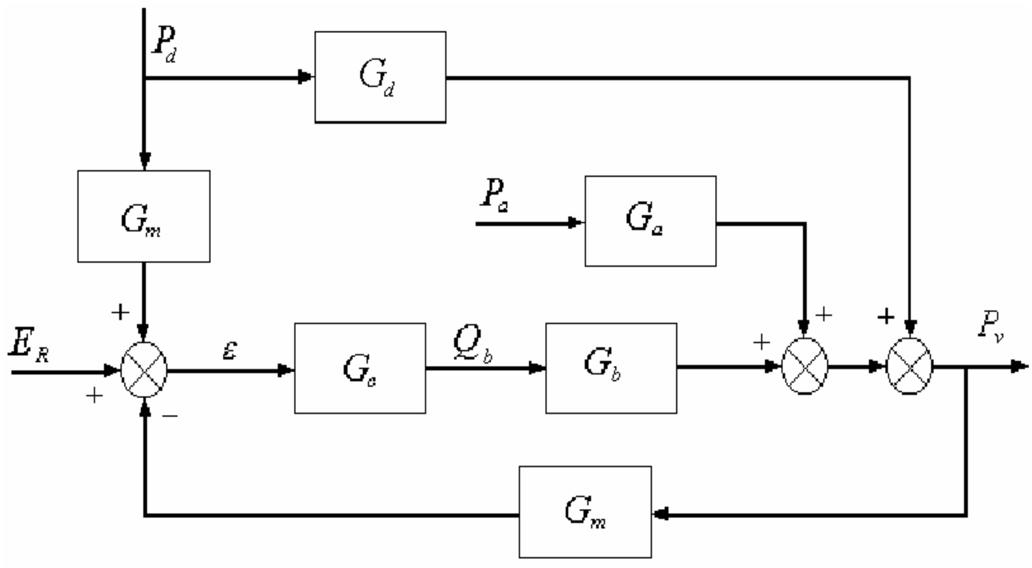

Figure 3. Block diagram for the control system of pressurized drum sealing.

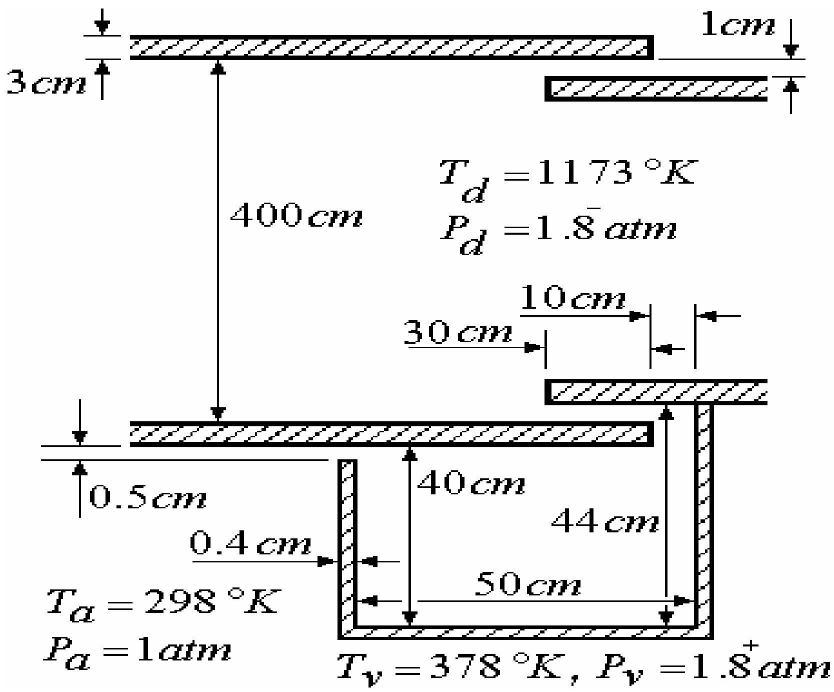

Figure 4. Geometrical dimensions and data used in controller design and simulation.

control system. As it is perceived from this Figure, the drum pressure not only does act, inherently, as a disturbance to the pressurized volume, but it also affects the measured signal entering the controller straightly before passing through process. Thus, the error, ε , entering the controller, is chosen to be

$\varepsilon = E_R - (P_v - P_d)$. Therefore, leakage at steady state conditions should be available to the drum for reducing the error which may appear as a result of fluctuations in the disturbance pressures P_a or P_d.

The linear model developed here is a mechanistic model which is completely deterministic and is suitable for controlling the system in an adaptive mode control. Since the parameters of the linear model are completely specified and correlated to process parameters. Therefore, it is suitable for use in an adaptive algorithm for controlling the system. Thus, the quality of such an adaptive control algorithm will not be inferior, if it is not superior, to any other nonlinear control technique which uses the original nonlinear model of the system. This is due to the fact that the precise model of the system is available and there is no need to any identification technique to be used for modeling the system for use in an adaptive technique for controlling the system.

Now, we design a PI controller for the sealing system, then, the response of this control system will be simulated. The temperature of the frame volume is considered to be $378\ ^{\circ}K$ and its pressure at steady state be 1.8^{+} atm (a little more than the drum). The temperature inside the drum around the sealing region is considered to be $1173\ ^{\circ}K$ and the ambient air condition is considered as standard. The calculated parameters of the model are shown in Table 1.

The selected controller is PI and it is tuned on the basis of Ziegler-Nichols method [3,4]. The calculated parameters of the controller are shown in Table 2.

The simulated results of control system are shown in Figures 5 through 9. In Figure 5, step response of the volume pressure for changes in the drum pressure is shown. In Figure 6, the variations appearing in the manipulated variable (blower flow rate) are shown for the same input of Figure 5. Figures 7 and 8 are drawn similarly for step input of air pressure. As Figure 8 presents, increment of air pressure results in the decreasing of blower flow rate. This is due to the fact that any increment of air pressure results in decreasing the leakage from the volume to the atmosphere. Therefore, the necessary blower flow rate for maintaining the required set point, E_R, will decrease. In Figure 9, step response for a few changes in set point is depicted.

CONCLUSION

Practically speaking, it is impossible to run rotary

TABLE 1. Model Parameters According to the Specifications of Figure 4.

Parameters	Quantity
K_{Pa}	-0.0108
K_{pd}	0.0930
K_{Qb}(atm.sec/m^3)	0.4139
τ(sec)	0.4942

TABLE 2. Controller Parameters.

Parameters	Quantity
K_c	1.738
τ_I	2.63

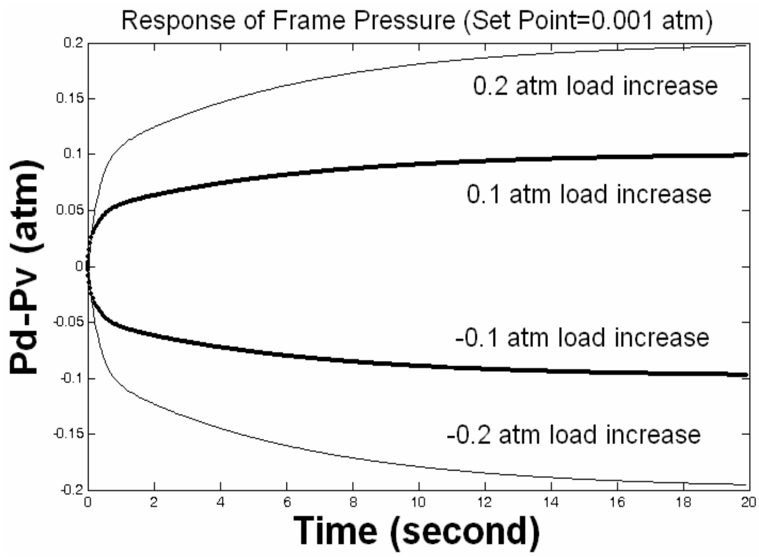

Figure 5. Step response of frame pressure for changes in drum pressure.

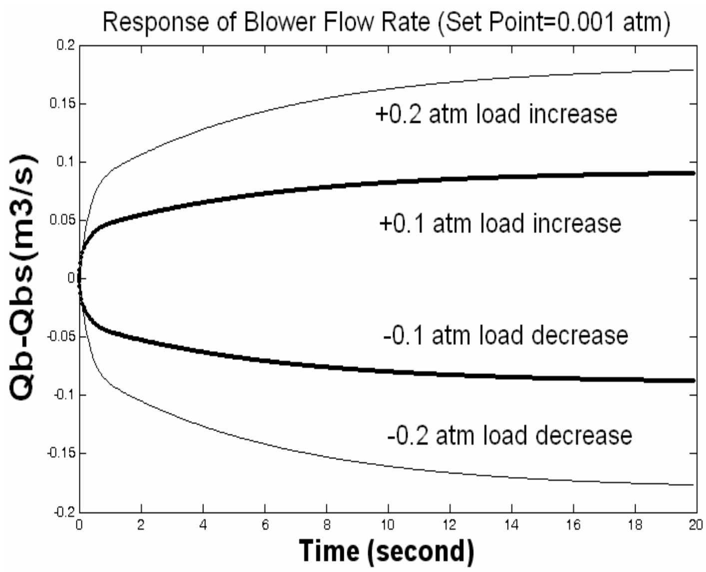

Figure 6. Variations appearing in the blower flow rate for step change in drum pressure.

drums under pressurized conditions by the current mechanical techniques of sealing. They are not capable of sealing the rotary drums under safe conditions without the leaking of gases (almost hot gases and materials) from the drum. But, the proposed controlled system of sealing in this paper is well suited to achieve this object. A suitable controller is provided which can reject the disturbances with the least fluctuations in differential pressure, appearing among the pressure of drum and that of sealing volume. By successfully controlling the pressure of the

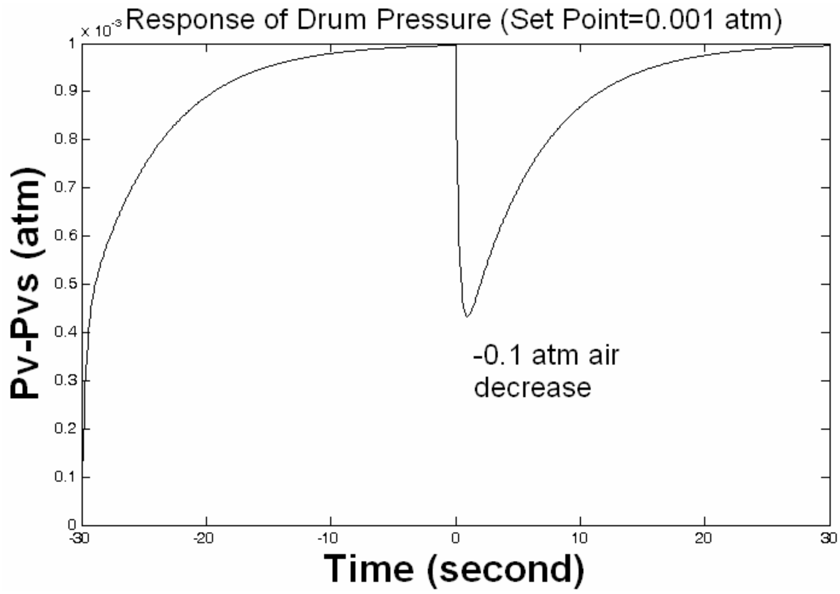

Figure 7. Step response of frame pressure to a step changes in air pressure.

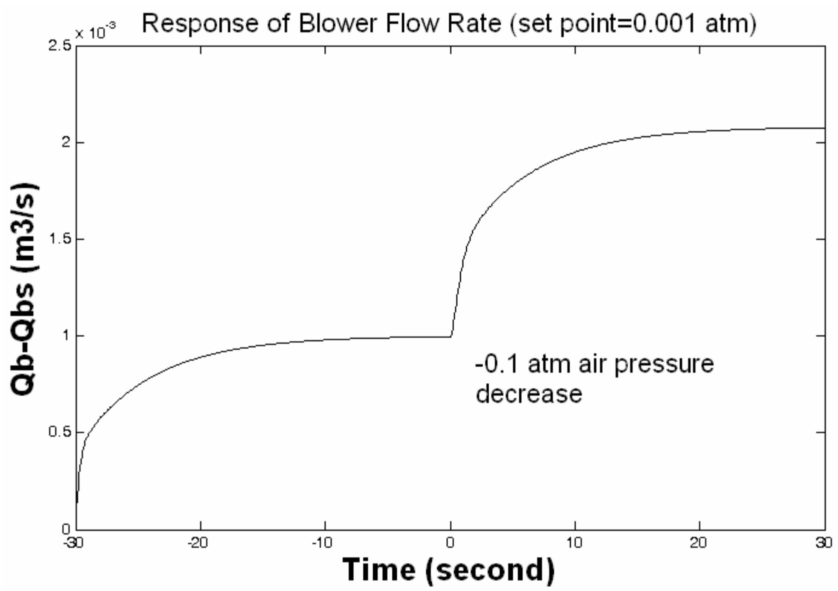

Figure 8. Variations appearing in the blower flow rate for step changes in air pressure.

pressurized sealing volume above the drum pressure, it is possible to prevent any leakage of gases to the ambient, even in high operating pressure conditions of the drum.

APPENDIX A

The differential Equation 11 is the dynamic model of the system which may be used for designing a

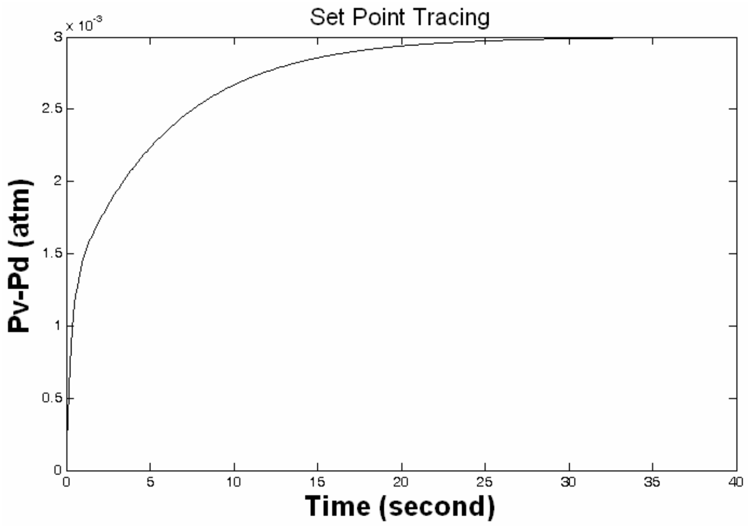

Figure 9. Frame and drum pressure difference responses to a step change in set point.

controller for it. For the purpose of linearization and obtaining the transfer matrix of the system, it can be changed to the following form after being divided by the coefficient of the differential term [6-7].

$$\frac{dP_v(t)}{dt} = \frac{P_v(t)T_v}{V_v\,T_a} Q_b(t) - A_{va}\sqrt{\frac{RT_v g_c}{2M_a V_v^2}}\sqrt{\frac{P_v^2(t)-P_a^2(t)}{\ln\frac{P_v(t)}{P_a(t)}+\frac{f_{va}l_{va}}{2r_{H,va}}}} - A_{vd}\sqrt{\frac{RT_v g_c}{2M_a V_v^2}}\sqrt{\frac{P_v^2(t)-P_d^2(t)}{\ln\frac{P_v(t)}{P_d(t)}+\frac{f_{vd}l_{vd}}{2r_{H,vd}}}} = g(P_a,P_v,P_d,Q_b) \quad \text{(A-1)}$$

In order to obtain a linear model for the system, tailor expansion series may be used around the steady state point of the variables, $P_{v,s}$ ، $P_{d,s}$ ، $Q_{b,s}$ و $P_{a,s}$, up to the first order term. The tailor series of the multivariable function, g, up to the first order term is [3,4]:

$$g\,(P_a,P_v,P_d,Q_b) = \left(\frac{\partial g}{\partial P_a}\right)_s \Delta P_a(t) + \left(\frac{\partial g}{\partial P_v}\right)_s \Delta P_v(t) + \left(\frac{\partial g}{\partial P_d}\right) \Delta P_d\,(t) + \left(\frac{\partial g}{\partial Q_b}\right)_s \Delta Q_b(t) \quad \text{(A-2)}$$

The, symbol, Δ, is used for describing the deviation of the variable from its steady state value. Such that:

$$\Delta P_a(t) = P_a(t) - P_{a,s}$$

$$\Delta P_v(t) = P_v(t) - P_{v,s}$$

$$\Delta P_d(t) = P_d(t) - P_{d,s}$$

$$\Delta Q_b(t) = Q_b(t) - Q_{b,s}$$

The partial derivatives in Equation A-2 are:

$$\left(\frac{\partial g}{\partial P_a}\right)_s = -\frac{A_{va}}{2}\sqrt{\frac{RT_v\,g_c}{2M_a V_v^2}}\left[\frac{P_{v,s}^2 - P_{a,s}^2}{\ln\frac{P_{v,s}}{P_{a,s}}+\frac{f_{va}l_{va}}{2r_{H,va}}}\right]^{-1/2} \times \left[\frac{-2P_{a,s}\left[\ln\frac{P_{v,s}}{P_{a,s}}+\frac{f_{va}l_{va}}{2r_{H,va}}\right]+\frac{P_{v,s}^2-P_{a,s}^2}{P_{a,s}}}{\left[\ln\frac{P_{v,s}}{P_{a,s}}+\frac{f_{va}\,l_{va}}{2r_{H,va}}\right]^2}\right] = g_{pa,s} \quad \text{(A-3-a)}$$

$$\left(\frac{\partial g}{\partial P_d}\right)_s = -\frac{A_{vd}}{2}\sqrt{\frac{RT_v g_c}{2M_a V_v^2}} \times \left[\frac{P_{v,s}^2 - P_{d,s}^2}{\ln\frac{P_{v,s}}{P_{d,s}} + \frac{f_{vd} l_{vd}}{2r_{H,vd}}}\right]^{-1/2}$$

$$\times \left[\frac{-2P_{d,s}\left[\ln\frac{P_{v,s}}{P_{d,s}} + \frac{f_{vd} l_{vd}}{2r_{H,vd}}\right] + \frac{P_{v,s}^2 - P_{d,s}^2}{P_{d,s}}}{\left[\ln\frac{P_{v,s}}{p_{d,s}} + \frac{f_{vd} l_{vd}}{2r_{H,vd}}\right]^2}\right] = g_{pd,s}$$

(A-3-b)

$$\left(\frac{\partial g}{\partial P_v}\right)_s = \frac{T_v Q_{b,s}}{T_a V_v} - \frac{A_{va}}{2}\sqrt{\frac{RT_v g_c}{2M_a V_v^2}} \left[\frac{P_{v,s}^2 - P_{a,s}^2}{\ln\frac{P_{v,s}}{P_{a,s}} + \frac{f_{va} l_{va}}{2r_{H,va}}}\right]^{-1/2}$$

$$\times \left[\frac{2P_{v,s}\left[\ln\frac{P_{v,s}}{P_{a,s}} + \frac{f_{va} l_{va}}{2r_{H,va}}\right] - \frac{P_{v,s}^2 - P_{a,s}^2}{P_{v,s}}}{\left[\ln\frac{P_{v,s}}{P_{a,s}} + \frac{f_{va} l_{va}}{2r_{H,va}}\right]^2}\right]$$

$$-\frac{A_{vd}}{2}\sqrt{\frac{RT_v g_c}{2M_a V_v^2}} \left[\frac{P_{v,s}^2 - P_{d,s}^2}{\ln\frac{P_{v,s}}{P_{d,s}} + \frac{f_{vd} l_{vd}}{2r_{H,vd}}}\right]^{-1/2}$$

$$\times \left[\frac{2P_{v,s}\left[\ln\frac{P_{v,s}}{P_{d,s}} + \frac{f_{vd} l_{vd}}{2r_{H,vd}}\right] - \frac{P_{v,s}^2 - P_{d,s}^2}{P_{v,s}}}{\left[\ln\frac{P_{v,s}}{P_{d,s}} + \frac{f_{vd} l_{vd}}{2r_{H,vd}}\right]^2}\right] = -g_{pv,s}$$

(A-3-c)

$$\left(\frac{\partial g}{\partial Q_b}\right)_s = \frac{P_{v,s} T_v}{V_v T_a} = g_{Q\,b,s} \quad \text{(A-3-d)}$$

Using A-3-a to A-3-d in Equation 2 results in Equation 4.

$$\frac{d\Delta P_v(t)}{dt} = g_{Pa,s}\Delta P_a(t) - g_{Pv,s}\Delta P_v(t) + g_{Pd,s}\Delta P_k(t) + g_{Qb,s}\Delta Q_b(t) \quad \text{(A-4)}$$

After some arrangement we have:

$$\frac{d\Delta P_v(t)}{dt} + g_{Pv,s}\Delta P_f(t) = g_{Pa,s}\Delta P_a(t) + g_{Pd,s}\Delta P_d(t) + g_{Qb,s}\Delta Q_b(t) \quad \text{(A-5)}$$

Equation 6 is Laplace transformed with the initial condition $\Delta P_v = 0$ to give:

$$s\,\Delta P_v(s) + g_{Pv,s}\Delta P_v(s) = g_{Pa,s}\Delta P_a(s) + g_{Pd,s}\Delta P_d(s) + g_{Qb,s}\Delta Q_b(s) \quad \text{(A-6)}$$

Then, after some more arrangement:

$$\Delta P_v(s).(s + g_{Pv,s}) = g_{Pa,s}\Delta P_a(s) + g_{Pd,s}\Delta P_d(s) + g_{Qb,s}\Delta Q_b(s) \quad \text{(A-7)}$$

Equation 7 is divided by $g_{Pv,s}$ to obtain the time constant of the system and the other gain parameters of the transfer functions.

$$\Delta P_v(s)\left(\frac{1}{g_{Pv,s}}s + 1\right) = \quad \text{(A-8)}$$

$$\frac{g_{Pa,s}}{g_{Pv,s}}\Delta P_a(s) + \frac{g_{Pd,s}}{g_{Pv,s}}\Delta P_d(s) + \frac{g_{Qb,s}}{g_{Pv,s}}\Delta Q_b(s)$$

Thus, the time constant and the gain parameters of the system are:

$$\frac{1}{g_{Pv,s}} = \tau \quad \text{(A-9-a)}$$

$$\frac{g_{Pa,s}}{g_{Pv,s}} = K_{Pa} \quad \text{(A-9-b)}$$

$$\frac{g_{Pd,s}}{g_{Pv,s}} = K_{Pd} \quad \text{(A-9-c)}$$

$$\frac{g_{Qb,s}}{g_{Pv,s}} = K_{Qb} \quad \text{(A-9-d)}$$

Therefore, the transfer functions are as the following:

$$G_a(s) = \frac{\Delta P_v(s)}{\Delta P_a(s)} = \frac{K_{Pa}}{\tau s + 1} \quad \text{(A-10)}$$

$$G_d(s) = \frac{\Delta P_v(s)}{\Delta P_d(s)} = \frac{K_{Pd}}{\tau s + 1} \quad \text{(A-11)}$$

$$G_b(s) = \frac{\Delta P_v(s)}{\Delta Q_b(s)} = \frac{K_{Qb}}{\tau s + 1} \quad \text{(A-12)}$$

The pressure variation of the frame volume related to the variations in the outer pressure and the inner pressure, as well as the flow rate of the blower is:

$$\Delta P_v(s) = G_a\,\Delta P_a(s) + G_d\,\Delta P_d(s) + G_b\,\Delta Q_b(s) \quad \text{(A-13)}$$

Thus, the matrix explanation of the relation between inputs and outputs of the system is:

$$\Delta P_v(s) = \begin{bmatrix} \frac{K_{Pa}}{\tau s+1} & \frac{K_{Pd}}{\tau s+1} & \frac{K_{Qb}}{\tau s+1} \end{bmatrix} \begin{bmatrix} \Delta P_a \\ \Delta P_d \\ \Delta Q_b \end{bmatrix} \quad \text{(A-14)}$$

SYMBOLS

A: cross sectional aria for gas flow from the gaps
E_R: set point of control loop
f: friction factor
g: function
g_c: conversion coefficient
G: transfer function
G': mass flux of gas or air from the gaps
K: steady state gain
l: distance of traveling of gas in the gap
m: mass
$\dot{m}(t)$: mass flow rate
M: molecular weight
P: pressure
Q: volume flow rate
r_H: equivalent hydraulic radius
R: gas constant
s: Laplase transform variable
t: time
T: temperature
V: volume

Subscripts

a: air, ambient
b: blower
c: controller
d: drum
m: measurement
v: pressurized volume
1: side one
2: side two

Greek Symbols

ρ: density
τ: time constant
Δ: difference

REFRENCES

1. Duda, W., "Cement Data Book", Bauverlag GmbH, Germany, (1988).
2. Beigel, B., "Abdichtungen Fur Drehofen", ***Zement-Kalk-Gips, (2KG)***, Vol. 5, (1971), 208-215.
3. Seborg, D. E., Edgar, T. F. and Mellichamp, D. A., "Process Dynamics and Control", John Wiley, New York, (1989).
4. Goodwin, G. C., Graebe, S. F. and Salgado, M. E., "Control System Design", Prentice Hall, New York, (1997).
5. McCabe, W. L., Smith, J. C. and Harriott, P., "Unit Operation of Chemical Engineering", McGraw Hill, New York, (2001).
6. Coughanowr, R., "Process Systems Analysis and Control", Second Edition, McGraw Hill, New York (1991).
7. Astrum, K. J. and Hagglund, T., "PID Controllers: Theory, Design, and Tuning", 2nd edition, Instrument Society of America, (1995).