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Control of Space Pressurization for Sealed or Tight Rooms

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Abstract

Space pressurization is well-known as a tool for control of air contaminants and a standard building block of laboratory ventilation systems. Several methods are in common use; steps to design, implement and commission them are widely published. Among those familiar lessons is the fact that the tightness or leakiness of the room envelope significantly affects the operation of the pressurization system.

When the room is sealed, or nearly sealed, mechanical coupling between air flows in and out of the room complicates control loop dynamics Flow and pressure loops that are ordinarily almost independent become tightly coupled with the potential to destabilize one another.

This paper develops the mathematical models that support analysis of a pressurization control system. The effect of mechanical parameters, especially the envelope leakage as it ranges from tight to effectively zero is exposed for study.

INTRODUCTION

Laboratory ventilation systems typically apply space pressurization to limit the spread of air contaminants from the laboratory room to surrounding areas. In most facilities, this is a straight-forward, routine process. In certain special facilities, the rooms are built to very low leakage specifications. This makes the pressurization control system very sensitive, requiring special control techniques. For a number of reasons, there is a trend toward tighter room envelopes in laboratories. making this a good time to analyze the lab pressurization problem from start to finish.

The aim of this analysis is to identify the factors that affect performance of room pressure control systems, and to show the effects of varying mechanical parameters. Control system analysis methods based on linear or linearized models can provide that kind of understanding. This does not mean the non-linearity of the system is unimportant or negligible; it means there is value in looking past it to gain the benefits of analysis.

MECHANICAL MODEL

The mechanical phenomena that characterize the system are modeled with basic equations from fluid mechanics. The equations are combined into a system and parameters are assigned.

Figure 1 represents the basic mechanical system. To keep the analysis manageable, there is one mechanical supply flow and one mechanical exhaust. Air exchange with surrounding spaces is represented by one infiltrating (or exfiltrating) air flow to one ambient pressure.

Room air pressure

The pressure within the lab room is affected by the air flow rates in and out of the room. That relationship is

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approximated by the ideal gas law. Taking the derivative against time, and assuming a constant room air temperature relates the rate of pressure change to the air flow rates.

$$P_R = \frac{mRT}{V}$$

$$\dot{P}_R = \frac{\dot{m}RT}{V}$$

It is convenient to discuss the system in terms of the various volumetric flow rates the make up the net mass flow rate into the room. To derive the room pressure equations, we neglect variations in air density through the system. This simplifying assumption is clearly not true. The air in the supply duct is denser than air in the room or the exhaust duct. The changes in room pressure that we study are directly related to density variations. However, the objective of the analysis does not depend on including density in the calculations. With the assumption in effect, the pressure rate is:

$$\dot{P}_R = \frac{RT\rho\sum Q}{V} = \frac{P_0}{V}\left[Q_s + Q_I - Q_E\right]$$

$$\dot{P}_R = \frac{P_0\sum Q}{V} = \frac{P_0}{V}\left[Q_s + Q_I - Q_E\right]$$

The parameter P_0 is defined as the absolute room pressure at the nominal operating point.

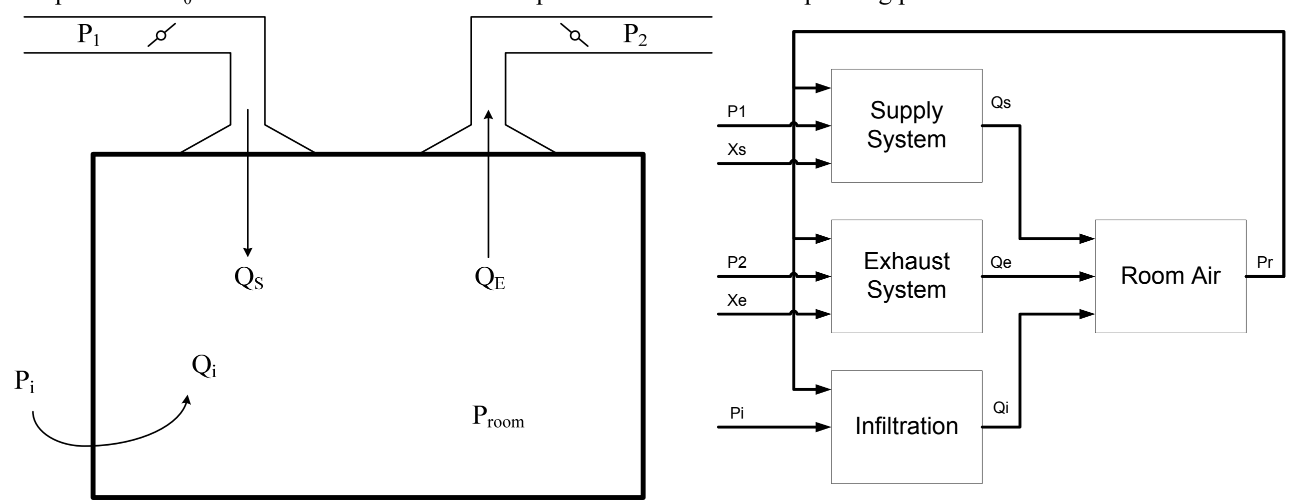

Figure 1: Schematic diagram identifying air flows and pressures in the mechanical model and block diagram illustrating structure of the corresponding mathematical model.

Air flow to and from the room

Supply: The supply system is represented by a high pressure at a reference point in the supply duct, pushing air through a variable flow resistance into the room. Pressure losses along the supply duct upstream and downstream of the supply damper are lumped into one variable resistance parameter (C_s) and included in the non-linear function ($D_s(X_s)$) that represents air flow resistance vs. damper stroke.

$$Q_S = \left(P_1 - P_R\right)^n C_S D_S\left(X_S\right)$$

The flow resistance parameter is a discharge coefficient in the orifice flow equation that models the supply air flow. It models mechanical sizing effects. The non-linear function takes values between 0 (perfectly sealed damper) and 1.0 (fully open damper.) The non-linearity makes it possible to represent characteristics of any type of damper and authority effects.

The pressure exponent (n) in the flow equation is taken as 0.5, meaning the flow is proportional to the square root of the pressure difference.

A real supply system serves more than one terminal. The individual terminals affect each other by changing the pressure drop in common sections of the duct work. The usual practice of controlling the pressure at a point in the duct system tends to reduce the effect, but it does not eliminate it. It is possible to extend this model to study multiple rooms and their effects but that is not within the scope of this paper.

Exhaust: The exhaust system is modeled the same way. Here, room pressure is the high side of the orifice and the suction pressure is in the exhaust duct.

$$Q_E = \left(P_R - P_2\right)^n C_E D_E \left(X_E\right)$$

Infiltration: Air exchange with surrounding spaces is modeled with the orifice flow equation. The parameter (C_I) sizes the air leakage path between the room and adjacent spaces. The exponent on the pressure difference is taken here as 0.5 but values closer to 1.0 can be selected to represent more nearly laminar flow.

$$Q_I = \left(P_I - P_R\right)^n C_I$$

Combined Mechanical Model

Equations for the room air pressure, and the air flows in and out are combined to represent a non-linear system. For purposes of this analysis, we consider the inputs to be the duct pressure and damper position for supply and exhaust, and the pressure in the adjacent space. From these we can calculate the three air flows and the room pressure. Figure 2 illustrates the structure of the model in block diagram form.

CONTROL SYSTEM

The previous section presents a model that can calculate room pressure and air flows from the input values of ambinet pressure and damper positions. In this section we model the other half of the system so we can calculate damper positions from flows and pressures. This enables us to analyze the closed loop. Simple models for the end devices (sensors and actuators) are combined with models of the control algorithms. Laplace Transform based transfer functions describe the dynamics of the control system components.

The damper motors and filtered air flow sensors are each modeled as first order lags. The damper motor model relates positions (x) to controller output values. (y) The flow sensor model relates measured flow values (Q_M) to actual flow values. (Q)

$$Q_{EM} = FQ_E = \frac{Q_E}{\left(\tau_F S + 1\right)} \qquad Q_{SM} = FQ_S = \frac{Q_S}{\left(\tau_F S + 1\right)}$$

$$X_E = MY_E = \frac{Y_E}{\left(\tau_M S + 1\right)} \qquad X_S = MY_S = \frac{Y_S}{\left(\tau_M S + 1\right)}$$

The mechanical model and the control system model fit together. When they are combined, we can analyze closed loop behavior.

Flow Offset Control

The first control algorithm modeled is the standard flow offset control. PID calculations operate dampers to drive the supply and exhaust flows to their setpoints. The supply flow setpoint is less than the measured exhaust flow by a fixed offset.

$$\frac{Y_E}{(Q_{ES}-Q_{EM})}=G_E=\frac{K_E(\tau_{EI}S+1)(\tau_{ED}S+1)}{\tau_{EI}S}$$
$$Q_{SS}=Q_{EM}-\Delta$$
$$\frac{Y_E}{(Q_{SS}-Q_{SM})}=G_S=\frac{K_S(\tau_{SI}S+1)(\tau_{SD}S+1)}{\tau_{SI}S}$$

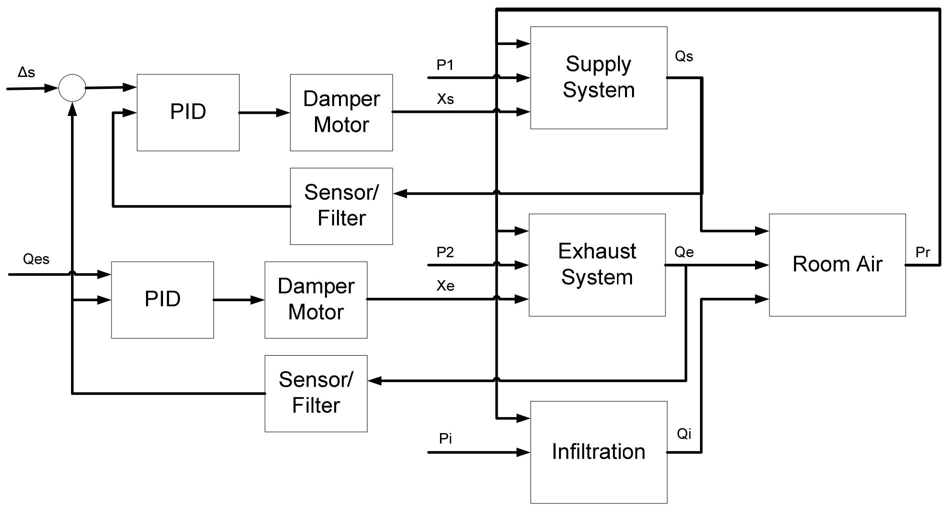

Figure 2: Block diagram of the complete mathematical model of the flow offset control system

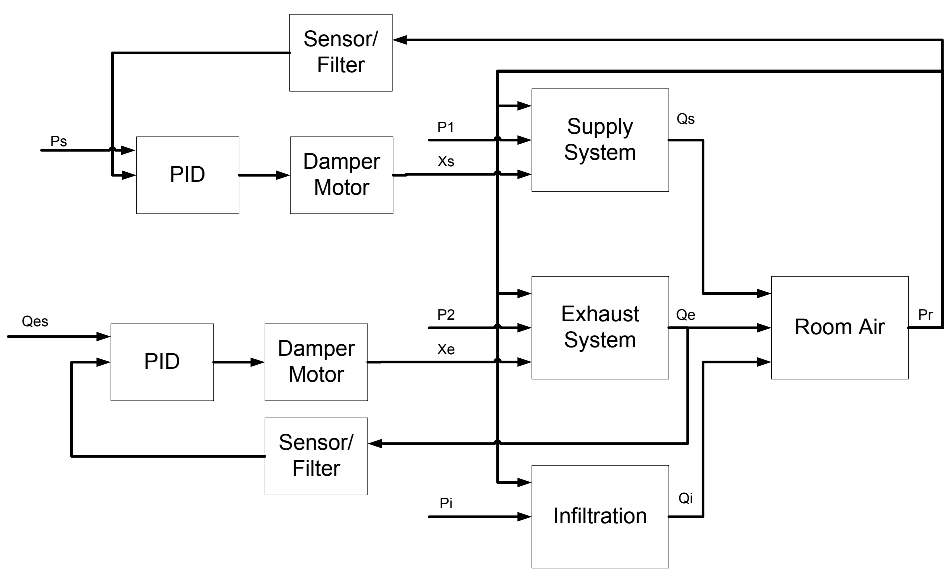

Figure 3: Block diagram of the complete mathematical model of the pressure feedback control system

Pressure Feedback

Pressure feedback is another common control strategy for pressurized rooms. The diagram shows the closed loop with the same mechanical model elements connected to a different control system model. The pressure control loop in the top half of the diagram operates the supply air damper. A flow control loop operates the exhaust damper.

As with the flow tracking system, we manipulate the blocks to eliminate the interconnecting loops before analyzing the system.

LINEARIZED SYSTEM FOR ANALYSIS

To enable control system analysis, this model is linearized at an operating point. The non-linear air flow equations are differentiated against each input and approximated by the first term of a Taylor expansion. To keep the problem simple, the duct pressures are taken as constants; the room pressure, the ambient pressure and the damper positions are taken as variable inputs to the air flow equations. The supply flow equation is linearized as follows:

$$Q_S = (P_1 - P_R)^n C_S D_S (X_S)$$

$$Q_S \approx Q_S\Big|_{P_{RO}, X_{S0}} + \frac{\partial Q_S}{\partial P_R}\Bigg|_{P_{RO}, X_{S0}} (P_R - P_{RO}) + \frac{\partial Q_S}{\partial X_S}\Bigg|_{P} (X_S - X_{S0})$$

$$Q_S \approx (P_1 - P_{RO})^n C_S D_S (X_{SO}) + n(P_1 - P_{RO})^{n-1} C_S D_S (X_{SO})(P_{RO} - P_R) + (P_1 - P_{RO})^n C_S N_S (X_S - X_{SO})$$

The three terms in the expression are the supply flow at nominal operating point, the deviation in supply due to changes in room pressure and the deviation in supply flow due to changes in supply damper position. The value, N that multiplies the deviation in damper position is the local slope of the non-linear damper curve. The damper position, X and the air flow resistance factor D(X) both range from zero to one. The local slope N may take any positive value. It is greater than 1.0 on the steep part of the damper curve, and less on a shallow part of the curve. For analysis purposes, it is not necessary that we identify the slope at any particular point, or determine a particular installed damper curve, as long as we check an appropriate range of values for N, covering steep, average and shallow characteristics.

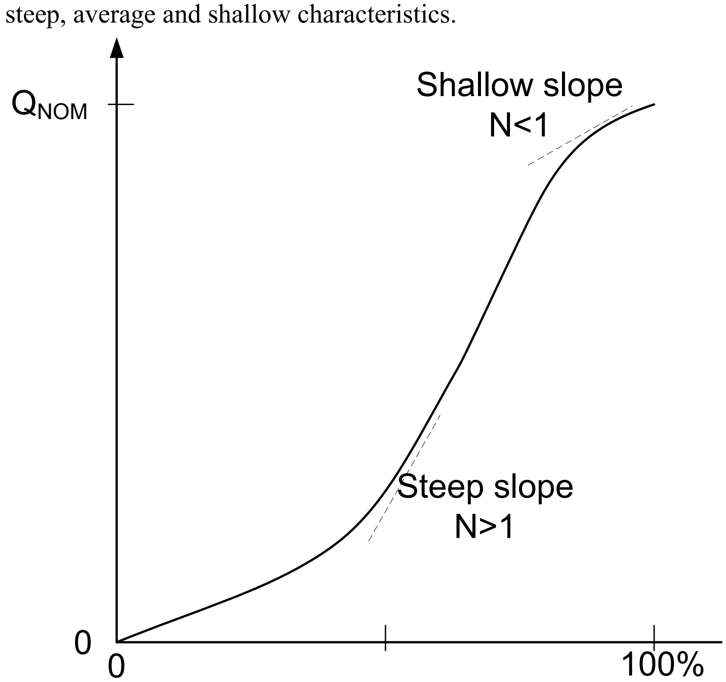

Figure 4: Non-linear damper curve and approach to linearization for analysis

For convenience, the linearized air flow equation is rewritten with a more intuitive set of sizing parameters. The discharge coefficient is replaced with values that represent the air flow at the nominal pressure drop. The nominal supply flow is defined as the supply flow, with the damper fully open (D(x)=1) and the room pressure at the nominal operating point.

$$Q_{SNOM} = C_S \left(P_1 - P_{RO}\right)^n$$

$$C_S = \frac{Q_{SNOM}}{\left(P_1 - P_{RO}\right)^n}$$

When this expression is substituted for the discharge coefficient, all the parameters in the linearized air flow equation are values that describe the desired operating point.

$$Q_S \approx Q_{SO} + Q_{SNOM}\left[\frac{-nD(X_{SO})}{(P_1 - P_{RO})}(P_R - P_{RO}) + N_S(X_S - X_{SO})\right]$$

$$Q_S \approx Q_{SO} - R_S(P_R - P_{RO}) + Q_{SNOM} N_S (X_S - X_{SO})$$

These are the same three terms, but expressed in a handier form for further analysis. The factor, R expresses the effect of room air pressure on supply flow. The factors, Q_{SNOM} and N_S, represent air terminal sizing and local slope of the damper curve. Together they express the effect of a damper movement.

The same mathematical steps apply to the exhaust flow.

$$Q_E \approx (P_{RO} - P_2)^n C_E D_E(X_0) + n(P_{RO} - P_2)^{n-1} C_E D_E(X_0)(P_R - P_{RO}) + (P_R - P_2)^n C_E N_E (X - X_0)$$

$$Q_E \approx Q_{EO} + R_E(P_R - P_{RO}) + Q_{ENOM} N_E (X_E - X_{EO})$$

The infiltrating air flow equation is simpler because there is no damper input. The factor, L, expresses the effect of room air pressure on the infiltrating air flow. It describes the leakage characteristics of the room.

$$Q_I \approx (P_{I0} - P_{R0})^n C_I + n(P_{I0} - P_{R0})^{n-1} C_I (P_{RO} - P_R) + n(P_{I0} - P_{R0})^{n-1} C_I (P_I - P_{IO})$$

$$Q_I \approx Q_{INOM} + \frac{nQ_{INOM}}{(P_I - P_{R0})}(P_{RO} - P_R) + \frac{nQ_{INOM}}{(P_I - P_{R0})}(P_I - P_{IO}) = Q_{INOM} + L(P_{RO} - P_R) + L(P_I - P_{IO})$$

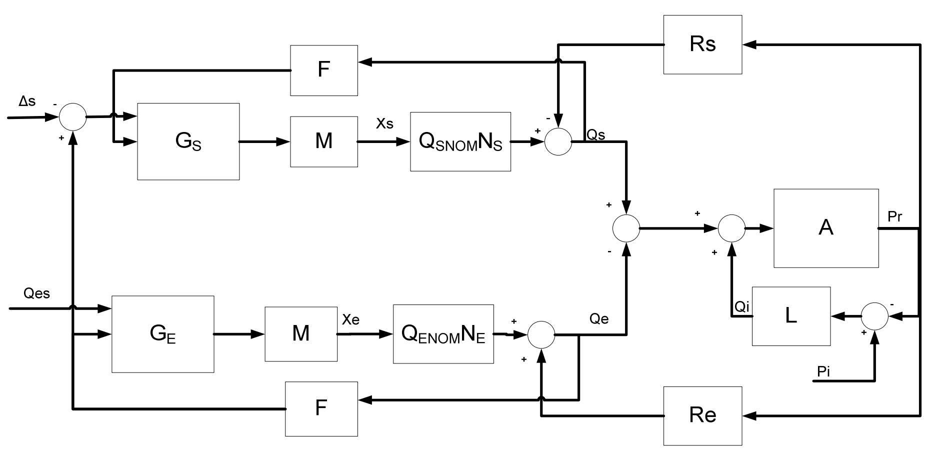

Figure 5: Block diagram of the linearized mathematical model of the flow offset control system

The block diagram of the linear model illustrates the relationship between the leakage parameter (L) and the mechanical flow parameters (Rs and Re) that characterize the ducted flows. Each is expressed as a ratio of flow to pressure. In a room with typical leakage characteristics, the L parameter is much larger than values for Rs and Re. This means a change in room pressure is made up almost entirely by changes in the infiltrating flow, with very little effect on the supply and exhaust flows. If L is small, the infiltration does not compensate so readily for changes, and pressure changes have more effect on the mechanical flows.

This representation also illustrates all the feedback paths in the system, not just the ones associated with the PID controllers. It is the view required to analyze the effects of parameters and groups of parameters on the system dynamics. For example, we can investigate the effect of the leakage parameter in comparison to the room flow parameters, Re and Rs.

The linearized flow equations can be combined with the pressure feedback system for the same purpose. This diagram is a little simpler, with fewer interlocking feedback loops.

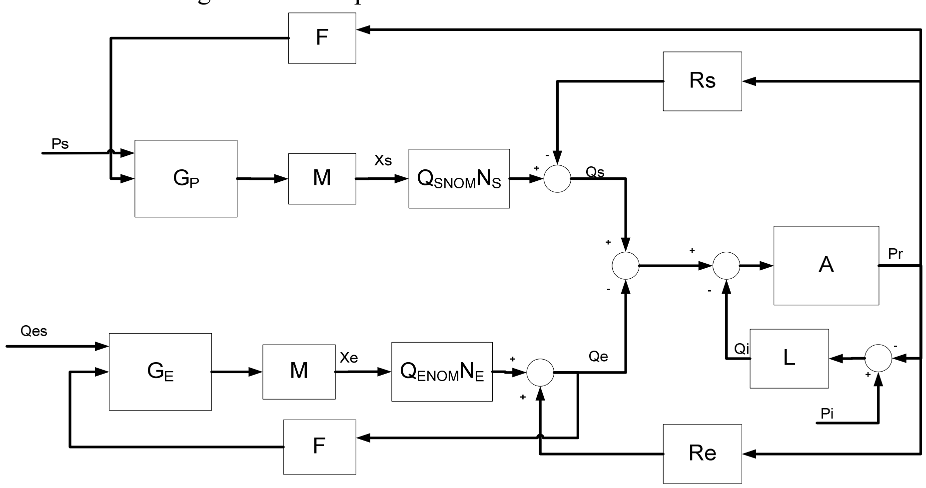

Figure 6: Block diagram of the linearized mathematical model of the room pressure feedback system

CONCLUSION

These complete feedback models can be used to analyze the dynamics of the pressurization control systems and to develop rules for effective design and tuning. Similar models can be developed for other pressurization control systems.

ACKNOWLEDGEMENTS

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NOMENCLATURE

C_E	Discharge coefficient for exhaust flow	Q_S	Supply air flow
C_I	Discharge coefficient for infiltrating air flow	Q_{SS}	Setpoint for supply flow control
C_S	Discharge coefficient for supply flow	Q_{SM}	Measured value of supply flow
D_E	Function expressing exhaust damper curve	Q_{SNOM}	Supply flow at nominal pressures and fully open damper
D_S	Function expressing supply damper curve	R	Gas constant
F	Transfer function representing air flow sensing	R_E	Factor representing effect of room pressure

	equipment
G_E	Transfer function representing exhaust flow controller
G_S	Transfer function representing supply flow controller
K_E	Proportional gain for exhaust flow control
K_S	Proportional gain for supply flow control
L	Factor representing effect of room pressure on infiltrating flow
m	Mass of air in room
M	Transfer function representing damper motor
n	Exponent in orifice flow equation
N_E	Factor representing local slope of non-linear exhaust damper curve
N_S	Factor representing local slope of non-linear supply damper curve
P_1	Air pressure in the supply duct
P_2	Air pressure in the exhaust duct
P_I	Air pressure in space adjacent to room, at the source of infiltrating air
P_{I0}	Air pressure driving the infiltration at the operating point
P_R	Air pressure in room
P_{R0}	Room pressure value at the analyzed operating point
Q	Air flow
Q_E	Exhaust air flow
Q_{EM}	Measured value of exhaust flow
Q_{ES}	Setpoint for exhaust flow control
Q_{ENOM}	Exhaust flow at nominal pressures and fully open damper
Q_I	Infiltrating air flow
Q_{INOM}	Infiltrating flow at the nominal pressures
	on exhaust flow
R_S	Factor representing effect of room pressure on supply flow
S	Laplace Transform operator
T	Temperature of air in room
V	Volume of air in room
X_E	Exhaust damper position
X_{E0}	Exhaust damper position at the analyzed operating point
X_S	Supply damper position
X_{S0}	Supply damper position at the analyzed operating point
Y_E	Output value of the exhaust flow controller
Y_S	Output value of the supply flow controller
ρ	Density of air in room
τ_{ED}	Time constant associated with derivative in exhaust flow controller
τ_{EI}	Time constant associated with integrator in exhaust flow controller
τ_F	Time constant related to flow sensing filter
τ_M	Time constant associated with damper motor
τ_{SD}	Time constant associated with derivative in supply flow controller
τ_{SI}	Time constant associated with integrator in supply flow controller