On a Control Policy that maintains Indoor Air Quality in a Variable-Air-Volume Air-Handling Unit

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Abstract

The air handling unit (AHU) in a variable air volume (VAV) heating, ventilation, and air-conditioning (HVAC) system consists of outside-air, return-air and exhaust-air dampers. The volume flow of air through a damper can be modified by changing the damper angles. In a typical AHU outside air is mixed with recycled-air and this mixture is then circulated to the various rooms in a building. A portion of the net volume flow out of these rooms is exhausted while the remainder is recycled. The net volume flow is usually determined by a supply fan in the duct, whose speed is dependent on the quiescent thermal load. In the absence of any regulation, the volume flow of outside air will depend on the speed of the supply-fan. Additionally, motivated by the fact that air entering the system through the exhaust-air damper will not be appropriately pre-conditioned as that entering the system through the outside-air damper, it is required that air does not enter the AHU through the exhaust-air damper. Using flow-conservation equations we derive a model for the AHU. This model was validated by experimental data collected on a full-scale, HVAC test facility [3]. Using this model we provide a control policy that achieves the objective of maintaining IAQ in a VAV HVAC system. This control policy is validated manually using experimental data.

1 Introduction

A typical air handling unit (AHU) in a heating, ventilation, and air-conditioning (HVAC) system consists of an outside-air, return-air and exhaust-air damper. Air from the outside is drawn into a HVAC system via the outside-air damper. This air is usually mixed with recycled air and circulated to the rooms in a building. A portion of the net volume flow out of the rooms is exhausted, while the remainder is recycled. The quantity of outside-, exhaust- and recycled-air is modified by appropriate changes to the outside-, exhaust- and return-air damper angles. The duct work in an HVAC system is equipped with a supply-fan. The speed of the supply-fan is determined by the current thermal load of the HVAC system, and is therefore not constant. In the absence of any control, varying speeds of the fan will result in different volume flows of outside air into the system. Indoor air quality (IAQ) requirements necessitate a constant intake of outside air independent of the supply-fan speed. In article [2] Cohen suggests the requirement of constant outside-air intake translates to the requirement of pressure-regulation at a portion of the ductwork identified in the HVAC literature as the mixed-air plenum. Using the derived model, we show that the requirement of constant outside-air intake translates to maintaining a constant pressure difference across the outside-air damper. In particular, if the atmospheric pressure remains constant, the above requirement translates to pressure-regulation at the

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mixed-air plenum. In article [4] Seem and House introduce the problem of reverse flow through the exhaust-air damper. The normal operations of the AHU requires air is exhausted through the exhaust-air damper. However, there are conditions under which the there is an intake of air through this damper. There are detrimental effects from this phenomenon. Air that enters the system through the outside-air damper is either pre-heated/pre-cooled appropriately before circulation. This is not true of air that enters the system through the exhaust-air damper. In reference [4], a control-policy that maintains an inverse relationship between the return- and exhaust-air damper angles is shown to prevent flow into the system via the exhaust-air damper. The AHU considered in reference [4] is equipped with a fan known as the return fan, this is in addition to the usual supply fan. The AHU considered in this paper only a supply fan. In this paper we concern ourselves with the issue of maintaining IAQ, while guaranteeing the proper direction of flow across the exhaust-air damper.

Section 2 presents our model of the AHU. Section 3 contains a description of the control-policy that maintains IAQ while guaranteeing the absence of reverse flow across the exhaust-air damper. Section 4 contains the conclusions and suggests directions for future research.

2 Model Derivation

We first derive an expression for the effective cross-sectional area of a parallel-blade damper. Letting \( \theta \) denote the damper angle, \( N \) the number of damper flanges, \( D \) the distance between damper blades, and \( W \) denote the damper width, the effective cross-sectional area \( A \) is given by the expression \( DWN \sin(\theta) \).

We model the dampers in the AHU as nozzles. Letting \( A_1 \) (\( A_2 \)) denote the area of the inlet (outlet), and \( p_1 \) (\( p_2 \)) denote the air pressure at the inlet (outlet) of the nozzle, using a one-dimensional, incompressible flow nozzle model (cf. appendix A.I.4, [5]) the volume flow, \( Q_{out} \), is expressed as

\[
Q_{out} = C_d(\theta) \tan(\theta) A_1 \sqrt{\frac{2(p_1 - p_2)}{\rho}}.
\]

We now derive expressions that relate the volume flow and static pressure at various portions of the duct. Assuming the volume flow through the outer-air (exhaust-air) damper is directed from the building-exterior into the AHU (from the AHU to the building-exterior), we have the following relationships,

\[
Q_{oa} = C_{oa}^d(\theta_{oa}) \tan(\theta_{oa}) A_{oa} \sqrt{\frac{2(p_a - p_{mp})}{\rho}} \quad (1)
\]

\[
Q_{ea} = C_{ea}^d(\theta_{ea}) \tan(\theta_{ea}) A_{ea} \sqrt{\frac{2(p_{rp} - p_a)}{\rho}} \quad (2)
\]

\[
Q_{ra} = C_{ra}^d(\theta_{ra}) \tan(\theta_{ra}) A_{ra} \sqrt{\frac{2(p_{ra} - p_{mp})}{\rho}} \quad (3)
\]

where \( C_{oa}^d(\theta_{oa}) \) is the coefficient of discharge for the outside-air/exhaust-air/return-air damper, \( \theta_{oa/ea/ra} \) is the outside-air/exhaust-air/return-air damper angle, \( Q_{oa/ea/ra} \) is the outer-air/exhaust-air/return-air volume flow, \( A_{oa/ea/ra} \) is the outside-air/exhaust-air/return-air damper's inlet/outlet area, \( p_{mp/rp} \) is the mixed-air (return-air) plenum pressure, \( p_a \) is the pressure at the return-air damper, \( p_{rp} \) is the atmospheric pressure, and \( \rho \) is the density of air (assumed to be constant for a given temperature).

In addition to the assumptions on the direction of volume flow across the exhaust-air and outer-air damper, we assume the volume flow across the return-air damper is directed from the return-air plenum to the mixed-air plenum. Under the flow-direction assumptions made above, the volume flow conservation expressions at the return-air and mixed-air plenums \( Q_{rp} = Q_{ra} + Q_{oa} \), and \( Q_{mp} = Q_{oa} + Q_{ra} \), after some simplifications (cf. [3] for details), results in the following expression for \( Q_{mp} \),

\[
Q_{mp} = C_{oa}^d(\theta_{oa}) \tan(\theta_{oa}) A_{oa} \sqrt{\frac{2(p_a - p_{mp})}{\rho}}
\]

\[
C_{ra}^d(\theta_{ra}) \tan(\theta_{ra}) A_{ra} \sqrt{\frac{2(p_{rp} - p_a)}{\rho}}.
\]

The volume flow \( Q_{mp} \) will be dependent on the fan speed, which in turn will depend on the quiescent thermal load. The volume flow into the return-air plenum, \( Q_{rp} \), together with flow losses in the zones will equal \( Q_{mp} \).

The model derived above requires plots that specify the value of \( C_{oa/ea/ra}^d(\theta_{oa/ea/ra}) \) as a function of \( \theta_{oa/ea/ra} \). This has to be experimentally determined for each AHU. It is worthwhile to note that most
AHUs use packaged HVAC equipment in the interest of cost and space. The vendors of the equipment can provide these plots in most cases.

In the next section we verify equation 4 experimentally, and in the section 3 we synthesize a class of control policies that maintain IAQ.

3 Description of the Control Policy

In the absence of any changes to damper angles, the outside-air volume flow is directly related to the total volume flow, which in turn is determined by the supply fan speed. The speed of the supply fan changes with the quiescent thermal load. Consequently, the outside-air volume flow into the system will vary. We seek a control policy that guarantees a constant outside-air intake. We note that the equations of section 2 assume the speed of sound, we ignore the influence of the dynamics caused by such pressure changes.

We note that the equations of section 2 assume the following relationship between the static-pressure at the mixed-air plenum and return-air plenum relative atmospheric pressure (p_a): p_r > p_a > p_m. We make a few observations on the relationship between the return-, outside-, exhaust-air damper angles, the supply fan speed and the return-air plenum and mixed-air plenum pressures. These observations are validated experimentally. The proofs of these observations are skipped for brevity and can be found in [3].

Observation 3.1 If p_m, p_r, p_a denote the mixed-air plenum, return-air plenum and atmospheric pressure respectively, and if p_r > p_a > p_m, then for a fixed return-air damper angle, \( \theta_{ra} \), fixed exhaust-air damper angle, \( \theta_{ea} \), and a fixed supply-fan speed, increasing (decreasing) the return-air damper angle, \( \theta_{ra} \), will result in (i) an increase (decrease) in the mixed-air plenum pressure, \( p_{mp} \), and (ii) an increase (decrease) in the return-air plenum pressure \( p_{rp} \).

Observation 3.2 If p_m, p_r, p_a denote the mixed-air plenum, return-air plenum and atmospheric pressure respectively, and if p_r > p_a > p_m, then for a fixed return-air damper angle, \( \theta_{ra} \), fixed exhaust-air damper angle, \( \theta_{ea} \), and a fixed supply-fan speed, increasing (decreasing) the outside-air damper angle, \( \theta_{oa} \), will result in (i) an increase (decrease) in the mixed-air plenum pressure, \( p_{mp} \), and (ii) an increase (decrease) in the return-air plenum pressure \( p_{rp} \).

Observation 3.3 If p_m, p_r, p_a denote the mixed-air plenum, return-air plenum and atmospheric pressure respectively, and if p_r > p_a > p_m, then for a fixed return-air damper angle, \( \theta_{ra} \), fixed outside-air damper angle, \( \theta_{oa} \), and a fixed supply-fan speed, increasing (decreasing) the exhaust-air damper angle, \( \theta_{ea} \), will result in (i) an increase (decrease) in the return-air plenum pressure \( p_{rp} \), and (ii) an decrease (increase) in the mixed-air plenum pressure, \( p_{mp} \).

Observation 3.4 If p_m, p_r, p_a denote the mixed-air plenum, return-air plenum and atmospheric pressure respectively, and if p_r > p_a > p_m, then for a fixed return-air damper angle, \( \theta_{ra} \), fixed exhaust-air damper angle, \( \theta_{ea} \), fixed outside-air damper angle, \( \theta_{oa} \), increasing (decreasing) the supply-fan speed will result in (i) a decrease (increase) in the mixed-air plenum pressure, \( p_{mp} \), and (ii) an increase (decrease) in the return-air plenum pressure \( p_{rp} \).

These observations are validated experimentally in [3]. In the remainder we assume the AHU is appropriately instrumented to provide the values of \( p_r - p_a \), \( p_a - p_{mp} \), and the supply-fan speed. We assume there are no wind-gusts, this issue is revisited in the conclusions of this paper. Also, we assume the vendor of the AHU equipment provides the relationship between \( C^{'a}_d(\theta_{oa}/\theta_{oa}/\theta_{oa}) \) and \( \theta_{oa} / \theta_{oa} / \theta_{oa} \) for the outside-air, exhaust-air and return-air dampers respectively. Let \( Q_{oa}^* \) be the desired, constant, outside-air intake. We make the reasonable assumption that the flow losses in the system is (significantly) smaller than \( Q_{oa}^* \) over range of supply-fan speeds. Figure 1 contains a flow chart describing a control policy that guarantees \( Q_{oa} = Q_{oa}^* \).

The system is initialized with a fixed value for \( \theta_{za}, \theta_{za} \) and \( \theta_{oa} \). For the purpose of illustration, we choose \( \theta_{za} = 0 \), at initialization. Although it is possible to effect the control-objective via variations in \( \theta_{ra}, \theta_{oa} \) or \( \theta_{ea} \) (cf. observations 3.1, 3.2 and 3.3), our control policy alters the value of \( \theta_{oa} \) and \( \theta_{ea} \) only. To permit the proper direction of flow across the exhaust- and outside-air damper, it is desirable to set \( \theta_{ea} \) and \( \theta_{oa} \) at a mid-range value. The supply-fan speed is gradually increased from zero till \( Q_{oa} = Q_{oa}^* \). The volume-flow \( Q_{oa} \) can be estimated using equation 1, \( C^{'a}_d(\theta_{oa}) \) vs. \( \theta_{oa} \) plot, and the measured value of \( p_r - p_{mp} \). Let \( p_{mp}(\theta_{oa}, p_a) \) be the value of \( p_{mp} \) that satisfies the requirement \( Q_{oa} = Q_{oa}^* \), where \( p_a \) is the ambient atmospheric pressure. It is easy to see that if...
If the fan-speed increases (decreases) as a consequence of a change in thermal load, then from observation 3.4 we infer a reduction (increase) in \( p_{mp} \). Let us suppose that following the increase in fan speed, we have the inequality \( p_{mp} < p_{mp}^*(\theta_{oa}, p_a) \), which in turn implies \( Q_{oa} > Q_{oa}^* \). The control policy of figure 1 would react by increasing \( \theta_{oa} \) by an amount \( \Delta \theta_{oa} \), where \( \Delta \theta_{oa} \) is the resolution of the actuator controlling the return-air damper angle (i.e. \( \Delta \theta_{oa} \) is small). By observation 3.1 this increase in \( \theta_{oa} \) will be accompanied by an increase in \( p_{mp} \) and a decrease in \( p_{rp} \). Let us suppose that in spite of the increase in \( p_{mp} \), the mixed-air plenum pressure is still smaller than \( p_{mp}^*(\theta_{oa}, p_a) \). This in turn implies the new value of \( Q_{oa} \) is still larger than \( Q_{oa}^* \). The system loss is assumed to be less than \( Q_{oa} \) and \( Q_{oa} > Q_{oa}^* \). If the decrease in \( p_{rp} \) results in the satisfaction of the inequality \( p_{rp} - p_a < \epsilon_{min} \), then \( \theta_{oa} \) is reduced by an amount \( \Delta \theta_{oa} \), where \( \Delta \theta_{oa} \) is the resolution of the actuator controlling the exhaust-air damper angle. The quantity \( \epsilon_{min} \) is a design-parameter that corresponds to the minimum value for \( p_{rp} - p_a \). From observation 3.3 we infer an increase in \( p_{rp} \) and a decrease in \( p_{mp} \). The reduction in \( p_{mp} \) will be accompanied by an increase in \( Q_{oa} \), therefore \( Q_{oa} > Q_{oa}^* \). Since the losses are assumed to be less than \( Q_{oa} \), the residual volume flow after the losses must be exhausted. This in turn suggests that \( \theta_{oa} \) cannot be identically equal to zero at any point in time. The abovementioned process is repeated until \( p_{mp} - p_{mp}^*(\theta_{oa}, p_a) \) is almost insignificant. The accuracy of regulation will depend on the value of \( \Delta \theta_{oa} \). Throughout the regulation process, we are guaranteed \( \epsilon_{max} > p_{rp} - p_a \geq \epsilon_{min} \), where \( \epsilon_{max} \) is a design-parameter that corresponds to the maximum value of \( p_{rp} - p_a \). The influence of a reduction in fan-speed can be established by a similar argument.

We are currently in the preliminary stages of implementing this control policy in our test facility. Currently, the damper angles are set manually via pneumatic actuators. The automatic implementation of the policy of figure 1 requires significant additions to the existing setup. However, the policy of figure 1 can be verified in toto by manually altering the damper angles based on \( p_a - p_{mp} \) and \( p_{rp} - p_a \) measurements. For illustration purposes, let \( \epsilon_{min} \approx 0.005 \) inches-H\(_2\)O, and \( \epsilon_{max} \approx 0.1 \) inches-H\(_2\)O. We present two experimental validations of the control policy in figure 1. To the meticulous reader we must mention that since the control is effected manually (i.e. not automatically), due to human error, there are durations in our experiments where we do not strictly meet the control requirements. In spite of this, the experiments provide the necessary validation.

Figure 2 presents a time-history of actions for the first experiment. Initially, the fan-speed is held at 72% of its rated maximum, this resulted in \( p_a - p_{mp} \approx 0.325 \) inches-H\(_2\)O, and \( p_{rp} - p_a \approx 0.026 \) inches-H\(_2\)O. After approximately twenty-five seconds, the fan speed was increased to 80% of its rated maximum. This resulted in \( p_a - p_{mp} \approx 0.4 \) inches-H\(_2\)O, and \( p_{rp} - p_a \approx 0.45 \) inches-H\(_2\)O. This confirms observation 3.1. From approximately the one-hundred second point to the one-hundred and seventy five second point, \( \theta_{oa} \) was increased manually in fixed increments. This resulted in a drop in \( p_{rp} - p_a \) and \( p_a - p_{mp} \), as predicted by observation 3.1. Since the process was done manually, due to human error, the value of \( \theta_{oa} \) was made too large, resulting in an extremely small value of \( p_{rp} - p_a \). As a corrective action the value of \( \theta_{oa} \) was decreased by a small amount till \( p_{rp} - p_a \approx \epsilon_{min} \). This is followed by the reduction of \( \theta_{oa} \) at approximately the three-hundred and twenty-five second point. This resulted in an increase in \( p_{rp} - p_a \) and a (negligible) decrease in \( p_a - p_{mp} \), as predicted by observation 3.3. Finally, at approximately the three-hundred and seventy-five second point, the value of \( \theta_{oa} \) is increased till \( p_a - p_{mp} \approx 0.325 \) inches-H\(_2\)O. Clearly, under automatic control, the quality of regulation will be significantly improved. Reference [3] contains other verifications of the control policy that are not reported herein.

4 Conclusions and Future Research

Indoor air quality standards require a constant intake of outside air into a HVAC system. In a variable air volume system, in the absence of any control, the quantity of outside air intake will vary according to the supply fan speed, which in turn varies as the thermal load changes. This is unacceptable. Additionally, the efficiency of a HVAC system can be significantly hampered if the flow of air is reversed at the exhaust-air damper. We develop a control policy that maintains a constant intake of outside air with the additional feature that the flow across the exhaust-air damper is never reversed. The control policy is obtained from a model of the AHU that is experimentally validated. This control policy in effect modulates the position of the return-air damper to regulate the mixed-air plenum pressure to a constant value.
The model assumes the ready availability of discharge coefficients of the various dampers. These parameters vary with the age of an AHU. We suggest investigations into an on-line estimation procedure that computes these parameters. At first-glance it seems an adaptive neural network model might be appropriate. Additionally, in this paper we assumed there are no wind-gusts. Wind-gusts can be represented in our model by an appropriate change to the atmospheric pressure. We conjecture that the objectives can be met by using a control policy similar to the one provided in this paper where the pressure difference between the atmospheric pressure and the mixed-air plenum pressure is regulated to a constant value. We suggest a detailed investigation into this issue as a future research direction. In many HVAC systems consist of a single actuator that is used to concurrently modify the three damper angles. This is usually accomplished by mechanical linkages. Essentially any one of the damper angles uniquely determines the other angles as dictated by the mechanical linkage used. If the presence of wind gusts necessitates the control of the exhaust- and outside-air dampers in addition to the return-air damper, it is of interest to identify the optimal relationship between the damper angles that achieves the objectives of this paper. We suggest investigations to such a commissioning tool as a future research topic.

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References


Figure 1: A control-policy that guarantees $Q_{oa} = Q_{oa}^*$ and $\epsilon_{max} > p_{rp} - p_a \geq \epsilon_{min}$.

Figure 2: A time-history of manual control-actions that validate the control-policy of figure 1, where the fan speed is increased.