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Virtual outdoor air flow meter for an existing HVAC system in heating mode

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ABSTRACT

Measurements from the operation of Heating, Ventilation and Air conditioning (HVAC) system are collected by Building Automation System (BAS) mostly for control purpose. In many air handling units (AHUs) the airflow meter is not installed on the outdoor air stream to reduce the initial costs. To provide such a missing information needed for the ongoing commissioning of HVAC system, a virtual flow meter (VFM) should be implemented in the BAS. This paper presents VFMs for the heating season when the heat recovery coils are used. Two models are presented which predict the outdoor air ratio (factor α), in the absence of such an air flow meter, by using BAS trend data. Two energy balance equations, one for the air mixing box and another for the heat recovery coils, are coupled. The results from a case study building show that the two models predict the daily average outdoor air ratio of 0.89 and 0.87, respectively, compared with the reference daily average ratio of 0.86.

1. Introduction

Building Automation Systems (BAS) installed in most institutional and commercial buildings collect and store a huge amount of information from building systems operation that can be used for the ongoing commissioning of Heating, Ventilation and Air conditioning (HVAC) systems if adequate application software is implemented. Schein [1] proposed an information system that connects the building commissioning system with BAS trend data. Trend data from BAS are proposed to be used for Fault Detection and Diagnosis (FDD) and sensors calibration [2–4]. However, due to the complexity of building systems, effective FDD and monitoring strategies might require additional sub-metering [5]. Those additional sensors would come at additional cost. Also, low quality of measurements (e.g., accuracy, missing or abnormal data) could be a barrier to the successful implementation of advanced monitoring strategies. The use of virtual sensors can effectively overcome many of such issues. Virtual sensors, or soft sensors, use mathematical models along with measurements from other physical sensors to predict the value of a variable that is difficult or expensive to measure [6]. The measurements from each sensor that are used as inputs to the virtual sensor model are affected by uncertainty due to the bias and random errors. The uncertainty of sensors then propagate through the virtual sensor formulation and affect the virtual measurement [7].

The measurement of air flow rates in AHUs is of a major significance for control and performance monitoring [8]. Generally, the Pitot traverse or vane anemometer are used for direct measurement of air flow rate in AHUs. Although field applications of physical air flow meters could reach a theoretical accuracy of about ± 1%, factors such as improper installation or gradual drifting might increase the device uncertainty. Yan et al. [9] showed that by using the actual outdoor air ratio, which varies with time, instead of a given minimum value, the estimates of cooling and heating energy consumption could be improved by 17% and 43%, respectively. Although direct measurement of the air flow rate is extremely desirable, practical issues exist: 1) the cost of a physical air flow meter; 2) the additional costs for periodical recalibration; and 3) the installation of a physical air flow meter that respects the minimum distance requirements, which is quite difficult or not possible because AHUs are quite compact [8,9,11].

The development of virtual flow meters (VFM) for AHU applications, along with the assessment of the uncertainty of results, is thus a topic of interest. In the last decade, several models have been proposed to virtually measure the air flow rate in AHU [6,8,10–15]. They are classified as white-box models, based on thermodynamic principles, and grey and black box models, which are inverse models extracted from measurements.

Tan and Dexter [8] developed (1) a relationship between the inlet outdoor air flow rate and the control signal for the inlet damper of a VAV system, and (2) relationships between the supply and extracted airflow rates and the control signals for the fans and dampers. The VFM produced estimates of the recirculated, supply and outdoor air flow rates, with relatively small errors of 8%, 2%, and 3%, respectively, when compared with direct measurements. In Hjortland and Braun [10] the supply air flow rate of a RTU was predicted from the fans VFD signal

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and the pressure difference across the supply fan. The same study proposed linear correlation models to correct: (1) the outdoor air temperature from the dampers position signal, the faulty outdoor air temperature and the return air temperatures as measured by the embedded sensors, and (2) the mixed air temperature from the supply mass air flow rate along with the dampers position signal, the faulty mixed air temperature, and the correct outdoor and return air temperatures as measured by the embedded sensors. Finally, a third order correlation model was proposed to predict the outdoor air fraction from the damper actuator control signal.

Yu et al. [11] proposed a VFM for RTU which derives the supply air flow rate from the first principle-based model of the heat transfer across the heating and cooling coils. Low cost measurements of the outdoor, mixed and supply air temperatures are used along with manufacturer information. When the RTU is operated in heating mode the proposed VFM predicted the supply air flow rate with a 6.9% uncertainty. In cooling mode, the proposed VFM uncertainty can rise up to 13.8%.

A grey-box based VFM for RTU was presented in [12]. For a given number of heating stages, the VFM was developed using three measurements: the supply and outdoor air temperatures, and the dumpers actuator signal. The model accuracy was estimated to be 6.8%.

Liu developed a VFM for the supply and return air flow rates which uses the fan speed, head and curve to predict the air mass flow rate [13]. The fan curve is given by the manufacturer. In [14] a new method is proposed to derive in-situ the fan curve, and thus improve the model proposed by Liu in [13].

Wang et al. [15] developed a VFM that uses measurements of the fan power input, along with the motor and fan efficiency, to predict the air flow rate. The motor efficiency was expressed as a function of the power input, voltage, and frequency.

In Mishukov and Horyna [16] the supply and exhaust air flow rates inside AHUs are virtually measured with a detailed, first principle-based model of a plate heat exchanger positioned between the supply and exhaust air ducts. The model has been validated on a test bench, showing an average error of 4% for both supply and exhaust flow rates.

The outdoor air flow entering the AHU can be derived from measurements of the supply air flow rate and the outdoor air ratio (Eq. (2)). The outdoor air ratio, the factor α, is the ratio of the outdoor to the supply air mass flow rates. By neglecting the change of latent heat in the air, the factor α can be derived from the air temperatures at the mixing box inlets and outlet (Eq. (2)) [9,17,18].

$$\alpha = \dot{m}_{oa}/\dot{m}_{sa} = \frac{\mathrm{T_{ma}} - \mathrm{T_{ra}}}{\mathrm{T_{oa}} - \mathrm{T_{ra}}} \tag{1}$$

$$\dot{m}_{oa} = \alpha\cdot(\rho_{air}\cdot \mathrm{V_{sa}}) \tag{2}$$

where $\dot{m}_{oa}$ and $\dot{m}_{sa}$ are the outdoor and supply air mass flow rate, ρ_{air} is the air density at the supply conditions, $\mathrm{V_{sa}}$ is the volumetric supply air flow rate, and $\mathrm{T_{ma}}$, $\mathrm{T_{ra}}$, and $\mathrm{T_{oa}}$ are the air temperatures at mixed, return and outdoor conditions, respectively.

In order to avoid additional costs due to the installation of dedicated sensors, VFMs should use measurements from other sensors already installed and connected to the BAS. The uncertainty of model predictions is an important aspect to be considered for the development and use of such a VFM. Cotrufo et al. [19] presented three models that predict the factor α for an AHU, based on: (1) the approximated energy balance equation (Eq. (1)); (2) the energy balance, water mass balance, and air mass balance equations; and (3) the energy balance equation. The factor α from Eq. (1) has the smallest uncertainty, and thus was preferred to the other two more detailed models. Results showed that, because of error propagation, the more complex is the mathematical model, the larger uncertainty should be expected.

This paper presents the case of an AHU that uses a heat recovery loop to pre-heat the outdoor air stream before the mixing box. The factor α from Eq. (1) requires the values of air temperatures at the mixing box inlets ($\mathrm{T_{oa}}$ and $\mathrm{T_{ra}}$) and outlet ($\mathrm{T_{ma}}$). When the heat recovery coil is used, the temperature of the outdoor air stream after the heat recovery coil ($\mathrm{T_{ac}}$) is higher than the outdoor air temperature ($\mathrm{T_{oa}}$). While $\mathrm{T_{oa}}$ is commonly measured by BAS, $\mathrm{T_{ac}}$ is not always available. In order to estimate the factor α with Eq. (1), a new sensor for $\mathrm{T_{ac}}$ should be installed at additional costs. In addition, the compact structure and air stratification inside the AHU prevent the accurate reading with only one sensor of air temperature after the heat recovery coil. Additional costs and practical issues may prevent for the correct calculation of the outdoor air ratio with Eq. (1). In this study two new models are proposed to overcome those issues. The new proposed models predict the factor α, without the need for $\mathrm{T_{ac}}$ measurements, by combining the equations of the energy balance at the mixing box and heat recovery loop. In addition, the uncertainty of predictions of the outdoor air ratio is presented. BAS trend data from an existing case study building are used to 1) identify different patterns of operation, 2) derive missing information, and 3) develop and validate the proposed models.

2. Method

From the energy balance equation at the mixing box, neglecting the latent heat, the factor α can be predicted by using (Eq. (3)), which uses the air temperature after the heat recovery coil ($\mathrm{T_{ac}}$) instead of the outdoor air temperature ($\mathrm{T_{oa}}$) when the system works on heat recovery mode.

$$\alpha = \frac{\mathrm{T_{ma}} - \mathrm{T_{ra}}}{\mathrm{T_{ac}} - \mathrm{T_{ra}}} \tag{3}$$

Eq. (3) is coupled with the energy balance equation of the heat recovery loop (Eq. (4)), and the factor α1 (Eq. (5)) is estimated without the need for the measurements of air temperature after the heat recovery coil ($\mathrm{T_{ac}}$). The factor α1 from Eq. (5) can be used in Eq. (2) to predict the outdoor air flow rate when the heat recovery loop is working.

$$\alpha1\cdot(\rho_{\mathrm{air}}\cdot\mathrm{V_{sa}})\cdot\mathrm{C_{p,air}}\cdot(\mathrm{T_{ac}} - \mathrm{T_{oa}}) = \rho_{\mathrm{glc}}\cdot\mathrm{V_{glc}}\cdot\mathrm{C_{p,glc}}\cdot(\mathrm{T_{hre}}-\mathrm{T_{hra}}) \tag{4}$$

$$\alpha1 = \frac{\mathrm{Q_{hr}} - \rho_{\mathrm{air}}\cdot\mathrm{V_{sa}}\cdot\mathrm{C_{p,air}}\cdot(\mathrm{T_{ma}} - \mathrm{T_{ra}})}{\rho_{\mathrm{air}}\cdot\mathrm{V_{sa}}\cdot\mathrm{C_{p,air}}\cdot(\mathrm{T_{ra}} - \mathrm{T_{oa}})} \tag{5}$$

where $\mathrm{Q_{hr}}$ is the heat recovered by the recovery loop and transferred to the outdoor air stream, $\mathrm{C_{p,air}}$ is the dry air specific heat at the supply air conditions, ρ_{glc} is the glycol density, $\mathrm{V_{glc}}$ is the glycol volumetric flow rate, $\mathrm{C_{p,glc}}$ is the glycol specific heat, and $\mathrm{T_{hre}}$ and $\mathrm{T_{hra}}$ are the glycol temperature before and after the heat recovery coil, respectively.

The return air temperature ($\mathrm{T_{ra}}$), which is measured before the return fans, is used in Eq. (5) as the temperature of the recirculated air stream, at the mixing box inlet. However, the actual temperature at the recirculation mixing box inlet ($\mathrm{T_{rec}}$) may significantly differ from $\mathrm{T_{ra}}$ because of air temperature increase through return fans and heat loss through recirculation ducts. Thus, a second model is proposed (Eq. (6)), in which the return air temperature is replaced by the temperature of the recirculated air. In order to avoid the use of a new sensor to measure $\mathrm{T_{rec}}$, a prediction model is developed to predict $\mathrm{T_{rec}}$ from other BAS available variables.

$$\alpha2 = \frac{\mathrm{Q_{hr}} - \rho_{\mathrm{air}}\cdot\mathrm{V_{sa}}\cdot\mathrm{C_{p,air}}\cdot(\mathrm{T_{ma}} - \mathrm{T_{rec}})}{\rho_{\mathrm{air}}\cdot\mathrm{V_{sa}}\cdot\mathrm{C_{p,air}}\cdot(\mathrm{T_{rec,p}} - \mathrm{T_{oa}})} \tag{6}$$

As the direct measurement of the outdoor air flow rate was not available from the BAS trend data in this case study, for the sake of comparison the reference factor α_{ref} was obtained from short-term measurements of the air temperature at the mixing box inlets ($\mathrm{T_{ac}}$ and $\mathrm{T_{rec}}$) along with BAS measurements of the mixed air temperature ($\mathrm{T_{ma}}$) (Eq. (7)). The short-term measurements were collected with portable data loggers with integrated thermistors [20]. Results from the two α-prediction models (Eqs. (5) and (6)) were compared with the reference value α_{ref} (Eq. (7)).