OPTIMIZATION OF VENTILATION SYSTEMS IN OFFICE ENVIRONMENT, PART I: METHODOLOGY

Liang Zhou†, and Fariborz Haghighat
Department of Building, Civil and Environmental Engineering Concordia University, Montreal, Canada

ABSTRACT
With growing concerns about the impact of indoor environment quality on office workers' well-being and productivity, coupled with the concern over the rising energy costs for space heating and cooling in office building sector, ventilation principles that integrate flexible and responsive elements have grown in popularity in office spaces. Such advanced elements as Underfloor air distribution (UFAD), passive swirl diffusers, and demand control on ventilation rate pose challenges to system design and operation. This paper is concerned with the development and implementation of a practical and robust optimization scheme, with the goal of aiding the office building designers and operators to enhance the thermal comfort and indoor air quality (IAQ) without sacrificing energy costs of ventilation. The path taken is a simulation-based optimization approach by using computational fluid dynamics (CFD) techniques in conjunction with genetic algorithm (GA), with the integration of an artificial neural network (ANN) for response surface approximation (RSA) and for speeding up fitness evaluations inside GA loop. The objective function is constructed in a way attempting to aggregate and weight indices (for thermal comfort, IAQ, and ventilation energy usage assessment) into one indicator.

KEYWORDS
Office indoor environment, ventilation system, CFD simulation, ANN, GA

INTRODUCTION
Within the scope of office spaces, ventilation systems intend to satisfy air quality and thermal comfort requirements by delivering the conditioned air (return air plus outdoor fresh air). Despite the facts that modern ventilation systems strike to offer us the ability for subtle indoor climate control, thermal conditions in office spaces are still far from perfect. From year to year, the international facility management association (IFMA) announces survey data, identifying the fact that the predominant office occupants’ complaints are “it’s too hot and too cold, simultaneously” (IFMA 2003).

Space conditioning contributes a major portion of the energy bill paid by office buildings. In 2004, approximately 60% of the total energy consumed by Canadian commercial/institutional buildings was used for space air conditioning (NRCan 2005). To reduce such a large energy cost, office building constructions have shifted towards high level of insulation and air tightness, and minimal ventilation rate as well. However, this has led to the deterioration in IAQ and raised the problem of sick building syndrome (SBS). Some research (Peijersen et al. 1999, Wargocki et al. 2000) claimed that the risk of SBS and associated sick–leave rates were strongly correlated to building material emission and ventilation rates in office buildings.

It is obvious that trade-offs always exist between the energy consumed by ventilation and the benefits of ventilation to occupants’ comfort and health; therefore, further investigations on indoor environment in ventilated office spaces should orient to a holistic evaluation of thermal comfort, IAQ and system energy efficiency.

† Corresponding Author: Tel: + 1 514 848 2424 ext: 7257
E-mail address: grace_zhouliang@yahoo.com
Also, the above statements reveal that plenty of room remains for improving ventilation design and operation in office spaces and a broader effort should be made to promote more energy effective measures during the operation stage of ventilated office spaces. In response to this, many building designers and researchers have turned to advanced ventilation systems that integrate flexible and responsive elements such as UFAD, passive swirl diffusers, and demand control on ventilation rate. Such systems can supply air directly to where it is required and provide capability for local control, which pose challenges to system performance analysis and advancement due to the non-uniform temperature distributions created within the space. Under such circumstances, highly resolved method and systematic optimization approach are to be developed to guide ventilation design and operation. Although the accomplishments of previous studies on heating, ventilation, and air conditioning (HVAC) system control optimization have been significant, little work has been done to optimize the design configurations and operational states of ventilation systems in office environment, by integrating the thermal and air quality comfort together with energy efficiency into the objective function. The reason is threefold. First of all, in pursuit of improvement in IAQ, thermal comfort, and ventilation energy efficiency in office spaces, it would be necessary to acquire detailed information about the indoor airflow, the pollutant dispersion as well as the temperature variations resulting from various ventilated principles. Therefore, it is necessary to employ a highly resolved approach to look at the problem domain. Secondly, the ventilation performance in a particular office is highly dependent upon a variety of geometric and thermal factors, such as ventilation principles, air supply/return terminals configuration, indoor/outdoor thermal states, contaminant source location and emission rate, supply air conditions, etc.; consequently, a flexible modeling method has to be used to test a large design space. In addition, since the objective function of such an optimization problem is a nonlinear mix-integer one with multi optimum, care should be taken to select an appropriate optimization algorithm.

The current study is targeted at addressing such needs by devising and developing an optimization approach that encompasses two essential components: the first one concerns high-resolution indoor airflow and heat transfer investigation so as to capture the distribution of assessment indices regarding thermal comfort, IAQ, and energy usage; the other key component is the integration of an economical and applicable optimization scheme. Such a simulation-based optimization approach should be able to offer greater flexibility in an attempt to predict, evaluate, and compare a wide range of objectives and constraints.

**SIMULATION-BASED OPTIMIZATION APPROACH IN THE CURRENT STUDY**

**Integration of CFD**

Ever since Nielsen’s application of CFD techniques to model room airflow driven by a diffuser (1974), CFD has been routinely used in research to predict the detailed room airflow patterns, highly resolved temperature distributions, and pollutant transportation indoors for more than a quarter century. In contrast to physical measurements, CFD method is relatively inexpensive, applicable to any existing or conceived scenario, and can provide complete information. Consequently, the optimization approach underway is based on the use of CFD techniques to evaluate various ventilation system design configurations and operation states, with the hope that the near-optimal solution could be found thereafter. The platform for implementing and demonstrating CFD simulation in this study will be the Airpak package from Fluent Inc. (Airpak 2002).

**GA as optimization engine**

Generally, optimization algorithms can be sorted into two main categories: conventional gradient-based method and gradient-free direct search. Concerning the former category of method, such as SQP method and Hookes-Jeeves algorithm, some issues pertinent to such methods have proven to discourage the adoption of them in building related studies. For one thing, building phenomena are very
often nonlinear mixed-integer ones (that is, with both continuous and discrete variables), which may lead to discontinuous outputs and thus cause problems for gradient-based methods (Wetter and Wright 2003, Lu et al. 2005). For another, gradient-based methods are generally prone to local optimal (Wang and Jin 2000).

Due to these drawbacks associated with gradient-based methods, the school of methods—referred to as gradient-free global optimization strategies—turned out to be better suited for building applications. GA, as a commonest global optimization method, has gained momentum in the optimization of building thermal system design. The following advantages of GA have been described in previous studies: firstly, GA is capable of dealing with discontinuous variables and multi-modal problems, and is also able to tolerate noisy objective functions (Wright et al. 2002, Huang and Lam 1997); secondly, it can find a sufficiently acceptable solution (near optimal solution) using less computing time, in comparison to other algorithms such as mixed integer programming method (Sakamoto et al. 1999), it can thus be incorporated into on-line optimal control; furthermore, since no derivative information is needed during searching process, GA was also proven to perform well in conjunction with RSA methods (Chow et al. 2002, Lu et al. 2005); the most importantly, GA is essentially stochastic, it can thus have better chance to explore the entire design space and reach the global optimum. Accordingly, GA is chosen as the optimization engine hereafter.

ANN for RSA

One major negative attribute of GA is the relatively large amount of function evaluations involved throughout the search process that may slow down the convergence. In current study, the value of objective function is to be calculated based upon CFD estimates. The full-scale office CFD simulations (with the RNG k-\(\varepsilon\) model) performed in the current study usually take 17 to 20 hours of CPU time on a Pentium IV desktop computer (dual-processor) before reaching convergence. Furthermore, with the current GA set up, it would necessitate 5000 times of CFD program running throughout a GA search. It is thus impractical to directly invoke CFD simulation inside the optimization loop for fitness evaluation. One possible remedy to this problem is to establish a relatively inexpensive low fidelity model for RSA. Such a surrogate model, once built and validated using the inputs and outputs obtained from the high fidelity model (CFD simulation here), can be then used in the place of CFD simulation inside GA loop to account for objective function and thus reduce the computational cost. Though there exist other RSA strategies (e.g. statistical regression and simple curve-fit), ANN technique has gained the greatest popularity in building related studies as a global RSA method. The word ‘global’ denotes that such an approach can represent the response of the system over the entire design space. ANN technique is employed in current study to provide RSAs of the objective indices in response to the variations in the input variables. In order to facilitate automated optimization job, input-output data sets extracted from CFD simulations are to be used to pre-train and test the ANN model. The platform for implementing and demonstrating the ANN-based GA optimization is Matlab 7.1 (2006).

Figure 2 Topology of the feed-forward ANN in this study
The feed-forward ANN model to be built here contains 30 hidden neurons (one hidden layer), 7 inputs, and 7 outputs (as shown in Figure 1), and it incorporate hyperbolic tangent sigmoid function in the initial/hidden layers and linear transfer function in the output layer. Levenberg-Marquardt (in conjunction with early stopping technique) and Bayesian regularization are selected as the algorithms when implementing back-propagation training, to improve the generalization ability of the network.

Framework of the current numerical optimization

Due to the space limitation of this paper, the numerical details pertinent to GA and ANN are excluded from the current discussion. Figure 2 briefly outlines the implementation plan of the above numerical optimization methodology. One may notice from this flow chart that the topic of Latin hypercube sampling (LHS) is left open temporarily, which will be presented later on together with ANN training/testing data preparation. Also, the validation treatment of CFD simulation is dropped here. It is straightforward that the reliability of the optimization result here is critically dependent on the accuracy of the prediction of objective indices, that is, the validation of CFD simulations against measurements are required to build the foundation for further optimization. The CFD simulations in this study are pre-validated using experimental data from baseline cases with both UFAD system and ceiling mounted mixing system (MS), as introduced in another paper prepared by the authors for this conference. Good agreements between the measured and the predicted air velocity, temperature, and contaminant concentration profiles provide the justification for the current choice of turbulence model and the present specification of boundary conditions.

ASSESSMENT INDICES

As previously mentioned, the current investigations on ventilated offices would orient to a holistic evaluation of thermal comfort, IAQ and system energy efficiency. Accurate quantitative evaluation of indoor environment can be only realized by considering reasonable independent variables and by selecting appropriate criteria to assess the objectives of interest. The subsequent sections set out to address the topics of what criteria are suitable for the evaluation of comfort level, air quality, and ventilation energy costs.

Based on literature survey, the following indicators are selected in the current study to cover (collectively) all the issues of interest, which are to be calculated based on output quantities from CFD simulation and will be integrated into the objective function for optimization.

PMV for overall thermal comfort assessment
The PMV model (Fanger, 1970) is the most frequently used and best-understood models for quantitative thermal comfort analysis. PMV reflects the mean vote of a large group of occupants who are exposed to a given combination of thermal parameters. PMV index evaluates thermal environment in an indoor space by using a thermal sensation range scale: -3 (cold), -2 (cool), -1 (slightly cool), 0 (neutral), +1 (slightly warm), +2 (warm), +3 (hot). PMV is defined as a function of six thermal variables related to the indoor air conditions and human behaviors, including air temperature, air humidity, air velocity, mean radiant temperature (T_mrt), clothing insulation level, and human activity. The optimizer is attempting to pull the PMV value near the occupant closer to neutral value.

Equivalent temperature for asymmetric thermal sensation

Despite the provision of acceptable whole-body comfort level, UFAD systems are sometimes found to cause asymmetric thermal sensation, if not designed properly. That is, even an individual express global comfort, he/she may still experience discomfort at a particular part of his body. An indicator named as equivalent temperature (ET) was proposed to appraise thermally non-uniform environment. ET was originally introduced to study the highly non-uniform micro-climate encountered in automobiles (Wyon, 1989). ET integrates the independent effects of air temperature, air velocity, mean radiation, and solar load on heat loss/gain from occupant body into a single physical quantity. It is defined as the temperature of a uniform enclosure in which a human body would experience the same rate of heat loss as would in the actual thermally non-uniform environment. It was claimed that if the variations of ET over entire body are controlled in the range of -2 °C to 2 °C, there is no excessive thermal non-uniformity over the entire body. For each part of the human body, ET can be calculated using the following equations (Bauman et al., 2000),
\[ ET = T_s - I_s \times Q_t \]
where Ts is the skin temperature (°C), and Qt is the local heat loss rate from skin surface (W/m²).

Head to ankle temperature difference and local air velocity

UFAD strategies supply cooler air into the lower region of the office and produce distinct temperature stratification; however, excessive temperature gradient along vertical direction may result in complaints. According to ASHRAE Standard 55-2004, the temperature difference between the head and the ankle level should not exceed 3 °C, which corresponds to 5% of dissatisfaction. It is necessary to maintain such a vertical temperature difference at or below the recommended value, in order to avoid “cold feet and warm head” complaints. In addition, ASHRAE standard 55-2004 also suggest that the local air speed near an office worker should be controlled at or below 0.25 m/s to avoid annoyance and distraction.

The variant of ET on both sides of individual occupant, the head to ankle temperature difference, and the local air velocity near the occupant will be integrated into the penalty terms in the objective function for optimization. When these three indices exceed the recommended values, the penalty term would be set to a relatively large positive number (since GA intends to minimize the objective function here).

CO₂-based ventilation effectiveness for IAQ assessment

In the baseline experimental cases, CO₂ was injected to the office space as tracer gas. The calculated CO₂ concentration distribution can be further integrated into a dimensionless index—ventilation effectiveness (εv). εv is calculated from the equation below, which is slightly different from the original definition (Awbi, 2003). The CO₂ concentration at breathing level is used here instead of the average concentration throughout the workstation.
\[ \varepsilon_v = \frac{c_{return} - c_{supply}}{c_{supply} - c_{supply, PH}} \]
where \( c_{return} \) is the CO₂ concentration in the return air (ppm), \( c_{supply} \) is the CO₂ concentration in the
supply air (ppm), and \(c_p\) is the CO₂ concentration at breathing level near the occupant (ppm).

The transportation of CO₂ in room air is predicted by CFD program; furthermore, the optimizer intends to search for better system design and operation parameters, so as to decrease the CO₂ concentration at breathing level and improve \(\epsilon\) as much as possible.

**Energy demand for ventilation**

A popular trend within the context of CFD simulation for office environment is towards the prediction of airflow; in contrast, issues with regards to heat transfer and energy usage have not been successfully addressed using CFD techniques. Based on the survey of related previous experimental and numerical, the current study divides the energy usage by ventilation into two main parts—fan power input and cooling energy consumption, which can be derived from the outputs estimated by CFD calculation.

1. Fan energy consumption

Principally, we know that fan power input (W) can be determined using the following expression

\[
E_{fan} = \Delta P \times \dot{V}_{air, total} \times \eta_{fan} / 1000
\]

where \(\Delta P\) is the pressure rise through the supply fan (Pa), \(\dot{V}_{air, total}\) is the overall volumetric flow rate of supply air (L/s), and \(\eta_{fan}\) is the fan efficiency. It is straightforward and understandable that, in the present study, the overall volumetric flow rate can be directly extracted from the CFD model; whereas the pressure rise and fan efficiency remain to be determined.

Based on the design guide for UFAD system (Bauman, 2003), the static pressure rise via the central fan with the MS is assumed to be at 750 Pa in this study while, with the UFAD system, 562.5 Pa is the static pressure rise assigned to the central fans (25% reduction). The efficiency of central supply fan in current study is characterized by relating the efficiency to overall supply flow rate. Since the pressure rise via the supply fan in this study is assumed to be at a constant level, its effect on fan efficiency is thus dropped here. Accordingly, the overall efficiency of the supply fan within the allowable flow rate range (80–160 L/s) is calculated using the following correlation:

\[
\eta_{fan} = \frac{\dot{V}_{air, total}}{400} + 0.25
\]

This correlation is established based on the assumption that when total supply air flow rate is at 160 L/s, the resulting combined fan energy efficiency at 65% while, when supply air flow rate is reduced to 80 L/s, it corresponds to a combined fan energy efficiency at 45%. The intermediate values can be approximated using a linear interpolation between the two extremes.

2. Cooling energy consumption

Cooling coils removes the sensible heat load produced within the conditioned space and offsets the humidity and temperature in the outdoor fresh air. In light of the previous methods used for energy usage prediction with alternate ventilation systems (Bauman, 2003; Xu and Niu, 2006), it is clear that the cooling energy requirement can be subdivided into two portions,

\[
Q_{cooling} = Q_{space} + Q_{vent} = m_{air, total} c_p (T_{supply} - T_{return}) + m_{fresh} (h_{supply} - h_{return})
\]

where \(Q_{cooling}\) is the total cooling load (W), \(Q_{space}\) is the cooling energy portion used to remove sensible heat load in the indoor space (W), \(Q_{vent}\) is the cooling energy portion used to condition the outdoor fresh air to return air states (W), \(m_{air, total}\) is the total mass flow rate of supply air (L/s), \(c_p\) is the specific heat of air (J/kg°C), \(T_{return}\) is the temperature of return air (°C), \(T_{supply}\) is the temperature of supply air (°C), \(m_{fresh}\) is the mass flow rate of outdoor
fresh air (kg/s), \( h_{\text{o}} \) and \( h_{\text{return}} \) are the specific enthalpy of the outdoor air and return air (J/kg), respectively.

Accordingly, the total energy consumed by cooling can be determined provided that the operating states of ventilation system, the return air conditions, and the outdoor air states are specified. Thermo-physical quantities such as the supply air temperature, the flow rate of supply air, the mean humidity ratio indoors, the return air temperature, the outdoor air moisture content, and the temperature would be treated as the independent variables in the analysis here. One thing need to be explained is, the relative humidity throughout the room is assumed to be maintained at a constant level around 40% and the outdoor relative humidity is assumed to be at 70%. Based on the relative humidity and air temperature, the moisture content and enthalpy in return/outdoor air can be determined accordingly.

**SPECIFICATION OF OBJECTIVE FUNCTION FOR OPTIMIZATION**

As previously mentioned, PMV, \( \epsilon_v \), and energy input for cooling load offset and fan driving are selected as the indices to measure comfort, IAQ and energy usage, respectively. Also, the thermal and airflow states of room air should comply with the constraints imposed on the head-to-ankle temperature difference, the variation of ET over the occupant’s body, and the local air velocity. Accordingly, the objective function can be prescribed by aggregating and weighting the above indices into one equation.

\[
J(X) = \min \left[ \sum w_i \left( \frac{\text{ABS}(\text{PMV}_i)}{\text{PMV}_{i,\text{max}}} \right) + w_{\text{fan}} \frac{\text{E}_{\text{fan}}}{\text{E}_{\text{fan},\text{max}}} + w_{\text{cooling}} \frac{\text{E}_{\text{cooling}}}{\text{E}_{\text{cooling},\text{max}}} + PT \right]
\]

where \( X \) is the input vector, consists of all the controlled variables including the design parameters and operating settings of ventilation systems. Subscript \( i \) denotes the number of occupants, as PMV is evaluated separately for individual occupant. \( w_1, w_{\text{fan}}, w_{\text{cooling}} \) represent the weighting factors for thermal comfort index, IAQ index, fan energy requirement index, and cooling energy requirement index, respectively.

It should be mentioned that GA in current case is attempting to minimize the cost function. Therefore, the first term in the cost function heads for pulling occupants’ thermal sensation (represented by PMV index) closer to neutral value; the second term intends to enhance the removal of indoor contaminant; the third and fourth terms are targeted at minimizing the energy consumed by space cooling and supply fan; the last term of the cost function is a penalty term, which accounts for the aforementioned constraints imposed on the flow and thermal conditions. \( \text{PMV}_{\text{max}}, \text{\( \epsilon_v \text{max}}, \text{E}_{\text{fan},\text{max}}, \text{and} \text{E}_{\text{cooling},\text{max}} \) are the maximum values of corresponding objective variables that can be observed from the training data, which are used to scale the objective variables into usable range ([0, 1] in the current case).

The magnitude of the weighting factors are to be specified by the user according to personal preference and based on sensitivity analysis. Different decision-maker will not have the same expectation from the system. For example, when a user concerns more about the overall comfort level than the IAQ issue, \( w_1 \) would be set to a higher value relative to \( w_{\text{fan}} \), whereas \( w_{\text{fan}} \) and \( w_{\text{cooling}} \) can be set to 1, whereas \( w_1 \) and \( w_{\text{cooling}} \) can be set to 0; etc. Results from sensitivity analysis by varying individual weights are presented in PART II of this paper.

**CLOSING REMARKS**

This part of the paper has set the technical stage for the optimization of ventilation system design and operation in office environment. Issues regarding the numerical methods, performance evaluation, and
objective function construction have been covered. In the subsequent parts of this paper, results from LHS (for data base construction), ANN training/testing, sensitivity analysis (for objective function specification), and GA optimization search will be presented.

REFERENCES