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A Non-Iterative Balancing Method for HVAC Duct System

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ABSTRACT

Building Heating, Ventilation and Air Conditioning (HVAC) system maintain comfortable indoor environment by supplying processed air to each terminal precisely through duct system. Testing, Adjusting and Balancing (TAB) plays critical role in achieving desired air distribution. Traditional TAB method is inaccurate and inefficient due to its trail-and-error natural, which forces people to pay high but expect low. Recently, it has been proposed that non-iterative approach to TAB is promising to improve performance and reduce cost. In this paper, a novel non-iterative balancing method is developed and implemented for TAB engineers to adjust dampers systematically and efficiently. Different from other TAB methods, this method is based on modeling and optimization. The mathematical model for duct system is firstly developed from its components including fan, duct segments and dampers to predict flow rates and pressures in the duct system for any damper positions. To identify the parameters in the model, flow rate measurements are taken for each terminal on real system under different damper positions. With the obtained model, optimal damper positions that gives desired air distribution are calculated by minimizing a specific objective function. To facilitate the adjusting process in real duct system, a sequential tuning instructions are generated which can help engineers to adjust dampers to their proper position using flowmeter as indicators. In this sequential tuning process, each damper only adjusts once to reach balance. Because the pressure and airflow dynamics of the duct system has been modeled, the entire TAB procedure is deterministic and non-iterative. Simulations are performed to validate the effectiveness of this method in Matlab/Simulink environment. Comparison study with existing methods shows that the proposed TAB method significantly shorten the duration of process and reduces balancing error while using easily-accessible equipment like pressure sensor and flowmeter only. It can be expected that the TAB service contractor will apply this method for advanced duct system where accurate air distribution is strictly required.

1. INTRODUCTION

Due to the impending shortage of energy and growing concern on environment, modern heating, ventilation and air conditioning (HVAC) systems in buildings are required to catch up with rising threshold on energy efficiency while pursuing higher indoor air quality (IAQ). Commercial building uses central HVAC system to maintain comfortable indoor environment, which is usually quantified by a combination of temperature, humidity and CO2 concentration level, by supplying a specific amount of process air to each terminal through duct system. Unbalanced duct systems that are unable to deliver desired amount of air to each terminal cause unsatisfactory thermal comfort and bad air circulation pattern. Consequently, occupants may suffer from sick building syndrome. Moreover, unbalanced duct system may give misleading signal to control system, causing the performance and energy efficiency of the system degraded. In fact, it is suggested that balancing should be performed regularly for all constant/variable air volume (CAV/VAV), induction, return air and even toilet and kitchen exhaust systems [5]. Moreover, as the system aging and changing room configurations during the operation, actual flow rates no longer match the demands and re-balancing the system are required. Therefore, testing, adjusting and balancing (TAB) plays critical role in balancing the system to achieve desired air distribution. However, many building owners are reluctant to perform TAB regularly, as the traditional TAB methods are costly and time-consuming, which largely interrupts normal operating. These disadvantages of the conventional TAB methods are due to the lack of solid theoretical analysis. Because of their inefficient trail-and-error nature, the results are largely depending on engineers' skill and experience.

A few researchers proposed new TAB methods to overcome these disadvantages. Federico Pedranzini et al. [3] designed a progressive flow method by utilizing the characteristic of duct resistance and air distribution. This method adjusts dampers progressively from the furthest terminals (largest pressure drop) to the closest terminals (smallest pressure drop). In order to decoupling the interactive effect in the ducts, variable speed drive and a controller for fan speed control at reference terminal are required. It is estimated that about 43% to 33% adjustments can be reduced. However, when the system is rebalanced due to the change of desired air distribution, the whole process must be repeated. Small [4] proposed developing mathematical model and balancing the system based on the obtained model. In order to determine the model parameters, the flow rates of each terminal under two different settings of damper positions are measured. After model identification, the damper positions for balancing are calculated. This method has advantage in rebalancing because the obtained model can be reused. However, due to the inevitable sensor noise and asymmetry of choosing the settings of damper positions, the model accuracy is poor and the sensitivity is biased distributed in the model. Hence the final balancing results is not very accurate.

In this paper, a novel non-iterative balancing method is developed and implemented for TAB engineers to adjust dampers systematically and efficiently. A more accurate model based on Darcy-Weisbach equation is developed to better characterize the duct system. New measuring process is designed to obtain data more accurately while simplify the operation procedure for TAB engineers. Differential and algebraic equation (DAE) software package and maximum a posteriori (MAP) estimation is applied in the model identification algorithm to reduce the uncertainties cause by sensor noise. A sequential tuning procedure using flow meter as indicators for adjusting dampers is implemented. The results are validated by simulation and compared to existing methods. The use of this method is also possible to be extended to balancing other system based on similar physical principles like exhaust duct network.

2. MODEL DEVELOPMENT

A schematic of duct model is shown in Figure 1. To model the air flow distribution behavior in duct system for balancing, the pattern of pressures and flow rates under a given duct system must be analyzed. Duct system consists of many components including duct segments, elbows, dampers and terminal outlets. Therefore, model of duct system must be built up from these components.

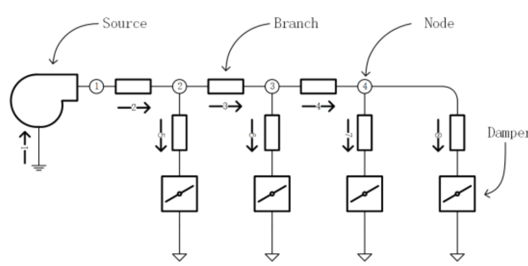


Figure 1: Schematic of duct model

Darcy-Weisbach equation has been used to estimate pressure drop along duct due to friction for fully developed flow in straight conduit for long [2], which is given by:

$$\Delta P = C_f \left(\frac{L}{D}\right)\left(\frac{\rho}{2}V^2\right) \tag{1}$$

where C_f is friction coefficient, L is duct length, D is duct diameter, ρ is air density and V is the average air velocity. The friction coefficient C_f depends on Reynolds Number and roughness of duct wall, the Colebrook's equation estimates C_f implicitly:

$$\frac{1}{\sqrt{C_f}} = -2\log\left(\frac{\varepsilon}{3.7D} + \frac{2.51}{\mathrm{Re}\sqrt{C_f}}\right) \tag{2}$$

where ε is the roughness of duct wall, Re is the Reynolds Number. Since Colebrook's equation is an implicit equation, instead of approximated formulas, DAE toolbox is employed to calculate the values.

For elbows and other fittings in the duct system, the pressure drop is estimated empirically by:

$$\Delta P = \frac{1}{2} C_f \rho V^2 \tag{3}$$

where local friction coefficient C_f is tested in experiments and collected as a look-up table in ASHRAE Duct Fitting Database. Many types of components have been tested including elbows, junctions, reducers and dampers at different positions.

The fan curve which characterize the relationship between pressures and flow rate is approximated by quadratic model:

$$\Delta P = P_{\max}\left(1 - \left(\frac{q}{Q_{\max}}\right)^2\right) \tag{4}$$

The entire duct system model is assembled in Simscape environment, which fully supports physical modeling by DAE approach. In Simscape, the physical domain is defined by the through and across variables. Through variables are subjected to conservation laws that sum up to zero among interconnected components, while across variable shares values for these interconnected components. In this application, flow rates through each branch are defined as through variables and pressures are defined as across variables. By building up the entire duct system in the Simscape environment, simulations can be performed to estimate the pressures and flow rates in the duct system. Different from usual blocks in Simulink, the ports of blocks built in DAE model are non-directional, which means the variables satisfy both sides. The equation (1)-(4) which relate through and cross variables of the two ports are encoded into different blocks including dampers, terminals and fans. Figure 2 demonstrates the Simscape environment to integrate the model of duct system.

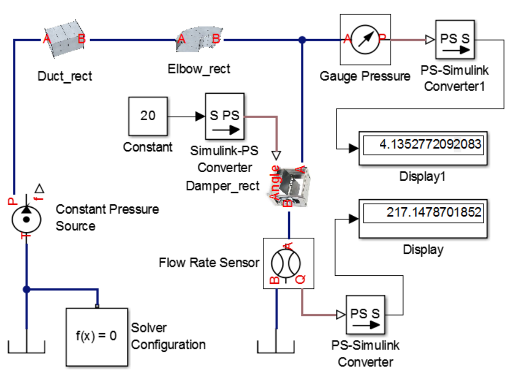


Figure 2: Simulink/Simscape duct system modeling environment

3. MODEL IDENTIFICATION

In order to obtain accurate prediction from this model, the parameters in the model must be identified by measuring data. Measurements are taken in $N+1$ steps for a N terminal duct system. In each step, the flow rates of each terminal are measured and recorded. In the first step, all dampers are fully open. For the following N steps, each damper is closed one time. Therefore, the overall data are symmetrically sampled to avoid biased calibration results. The purpose of keeping exactly one damper closed in each measurement are three-folds. First, cover the largest range of testing by setting to fully closed and opened position. Second, simplify the analysis because the fully closed damper completely stops the airflow. Third, avoid any potential damage to the system due to the high plenum pressure when too many dampers are closed. The detail procedure is shown in Figure 3.

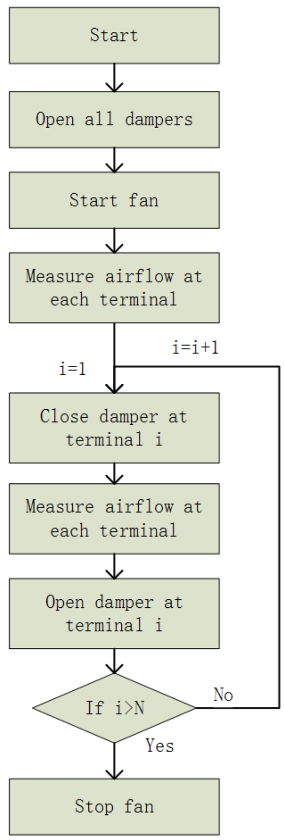


Figure 3: Measuring procedure

To identify the model parameters, Maximum a posteriori (MAP) estimation is used. Define the undetermined parameters $\boldsymbol{\beta}$ and the damper positions $\boldsymbol{\theta}$, then the simulation process can be represented as a function $\mathbf{X} = \begin{bmatrix} \mathbf{q}^T & \mathbf{P}^T \end{bmatrix}^T = f(\boldsymbol{\beta}, \boldsymbol{\theta})$. To derive the formula, the following assumptions are applied:

I) Measurement $\mathbf{Z}$ is conditional independent to parameters $\boldsymbol{\beta}$ given states $\mathbf{X}$ and damper position $\boldsymbol{\theta}$.

II) Measurement $\mathbf{Z}$ follows multi-variable normal distribution with mean $\mathbf{CX}$ and covariance $\boldsymbol{\Sigma}$, where $\mathbf{C}$ is the observation matrix.

III) The probability of states $\mathbf{X}$ given system parameters $\boldsymbol{\beta}$ is a single peak probability mass function:

$$P(\mathbf{X}|\boldsymbol{\beta}) = \begin{cases} 1 & ,\mathbf{X} = f(\boldsymbol{\beta}, \boldsymbol{\theta}) \\ 0 & ,\mathbf{X} \neq f(\boldsymbol{\beta}, \boldsymbol{\theta}) \end{cases} \tag{5}$$

IV) The prior probability of parameters $\boldsymbol{\beta}$ follows logarithmic normal distribution with given mean $\boldsymbol{\beta}_0$ and covariance $\boldsymbol{\Omega}$.

By applying these assumptions, the posteriori probability distribution can be estimated by:

$$P(\mathbf{X}|\boldsymbol{\beta}) = \begin{cases} 1 & ,\mathbf{X} = f(\boldsymbol{\beta},\boldsymbol{\theta}) \\ 0 & ,\mathbf{X} \neq f(\boldsymbol{\beta},\boldsymbol{\theta}) \end{cases} P(\boldsymbol{\beta}|\mathbf{Z}) \propto P(\mathbf{Z}|\mathbf{X},\boldsymbol{\beta}) P(\mathbf{X}|\boldsymbol{\beta}) P(\boldsymbol{\beta}) = P(\mathbf{Z}|f(\boldsymbol{\beta},\boldsymbol{\theta})) P(\boldsymbol{\beta})$$
$$\propto \exp\left(-\frac{1}{2}(\mathbf{Z}-\mathbf{C}f(\boldsymbol{\beta},\boldsymbol{\theta}))^{\mathbf{T}} \boldsymbol{\Sigma}^{-1} (\mathbf{Z}-\mathbf{C}f(\boldsymbol{\beta},\boldsymbol{\theta})) - \frac{1}{2}(\ln\boldsymbol{\beta}-\ln\boldsymbol{\beta}_0)^{\mathbf{T}} \boldsymbol{\Omega}^{-1} (\ln\boldsymbol{\beta}-\ln\boldsymbol{\beta}_0)\right) \tag{6}$$

To find the maximum a posteriori, it is equivalent to minimize the following objective function:

$$\hat{\boldsymbol{\beta}} = \underset{\boldsymbol{\beta}}{\operatorname{argmin}} \frac{1}{2}(\mathbf{Z}-\mathbf{C}f(\boldsymbol{\beta},\boldsymbol{\theta}))^{\mathbf{T}} \boldsymbol{\Sigma}^{-1} (\mathbf{Z}-\mathbf{C}f(\boldsymbol{\beta},\boldsymbol{\theta})) + \frac{1}{2}(\ln\boldsymbol{\beta}-\ln\boldsymbol{\beta}_0)^{\mathbf{T}} \boldsymbol{\Omega}^{-1} (\ln\boldsymbol{\beta}-\ln\boldsymbol{\beta}_0) \tag{7}$$

Since the model is nonlinear and implicit, it is difficult to derive the explicit form for estimated parameters in general. Therefore, global optimization algorithm may apply to estimate the parameters. In this work, particle swarm optimization is used to estimate the best-fit model parameters.

4. DAMPER ADJUSTMENT

For a duct system with N terminals, only N-1 dampers are need to be adjusted and one damper is fully open. This is because the total flow rate supply is controlled by fan speed so the degree of freedom to balance the air distribution is only N-1. To balance the duct system, the fully open damper must be located first. Define the damper position $\boldsymbol{\theta}$ to be 0 for fully open and 90 for fully closed. When all dampers fully open, the pressures and flow rates of the system is:

$$\mathbf{X}_0 = f(\hat{\boldsymbol{\beta}}, \boldsymbol{\theta}=0) \tag{8}$$

In this case, the flow rates on each terminal can be obtained by defining the terminal matrix $\mathbf{C}_T$ such that the flow rate vector for terminals is $\mathbf{q}_0 = \mathbf{C}_T\mathbf{X}_0$. Suppose the desired flow rates are denoted by vector $\mathbf{q}_d$, and the normalized desired flow rates vector $\hat{\mathbf{q}}_d$ is defined as:

$$\hat{\mathbf{q}}_d = \frac{\mathbf{q}_d}{\|\mathbf{q}_d\|_1} = \frac{\mathbf{q}_d}{\sum_i q_{d,i}} \tag{9}$$

Similarly, the fully open flow rates on each terminal $\mathbf{q}_0$ can also be normalized as $\hat{\mathbf{q}}_0 = \mathbf{C}_T\mathbf{X}_0 / \|\mathbf{C}_T\mathbf{X}_0\|_1$. By comparing their difference, the largest gap between $\hat{\mathbf{q}}_0$ and $\hat{\mathbf{q}}_d$ indicates the location of fully open dampers:

$$k = \underset{i}{\operatorname{argmax}}\, \hat{q}_{d,i} - \hat{q}_{0,\mathrm{i}} \tag{10}$$

Therefore, damper k is the fully open damper throughout the balancing process. For other dampers, the optimal damper positions are calculated by:

$$\begin{aligned} &\underset{\boldsymbol{\theta}}{\text{Minimize}} && \frac{1}{2}\left(\hat{\mathbf{T}} - \mathbf{C}_T f(\hat{\boldsymbol{\beta}},\boldsymbol{\theta}) \big/ \left\|\mathbf{C}_T f(\hat{\boldsymbol{\beta}},\boldsymbol{\theta})\right\|_1\right)^T \left(\hat{\mathbf{T}} - \mathbf{C}_T f(\hat{\boldsymbol{\beta}},\boldsymbol{\theta}) \big/ \left\|\mathbf{C}_T f(\hat{\boldsymbol{\beta}},\boldsymbol{\theta})\right\|_1\right) \\ &\text{subject to} && 0<\theta_i < 90 \text{ for } i = 1,\cdots,k-1,k+1,\cdots,\mathrm{N} \\ &&& \theta_k = 0 \end{aligned} \tag{11}$$

Again, the particle swarm optimization is applied to solve this nonlinear optimization problem to obtain the optimized damper position $\boldsymbol{\theta}^*$. Once the damper position for all dampers are determined, the engineers are able to adjust dampers in real duct system accordingly. Considering the imprecise damper position indicator in most of dampers, we proposed a sequential tuning procedure with the help of flowmeter as follow:

I) Initialize the settings of damper positions as fully open, and $i = 1$.
II) Set the damper *i* as its optimized position θ_i^* and compute the corresponding target airflow rate q_i^*.
III) Adjust the damper position such that the airflow rate in this terminal is equal to q_i^*.
IV) Inherit the current settings of damper position and set $i = i + 1$.
V) Repeat step II to IV as long as $i \le N_t$.
VI) After the last damper is adjusted, the desired airflow distribution is achieved.

During this process, each damper is only adjusted once. Therefore, no iterative adjusting process is needed. Noticed that during the adjusting process, the target airflow rate q_i^* is not equal to the desired airflow rate $q_{d,i}$ because of the coupling effect in the duct system. Once the last damper is adjusted properly, the airflow rate in all terminals will reach the desired airflow rate $q_{d,i}$ at the same time.

5. VERIFICATION

The ASHRAE has created a duct network in Example 8 of the ASHRAE handbook 2001 Chapter 34 [1] to illustrate the duct calculation procedures. For convenience, the schematic of Example 8 is represented in Figure 4. The duct system consists of two parts: the induced ventilation system under negative pressure before fan consists of three inlet terminals; the supply air system under positive pressure after fan consists of six outlet terminals. Desired air flowrates through all nine terminals are listed in Table 1.

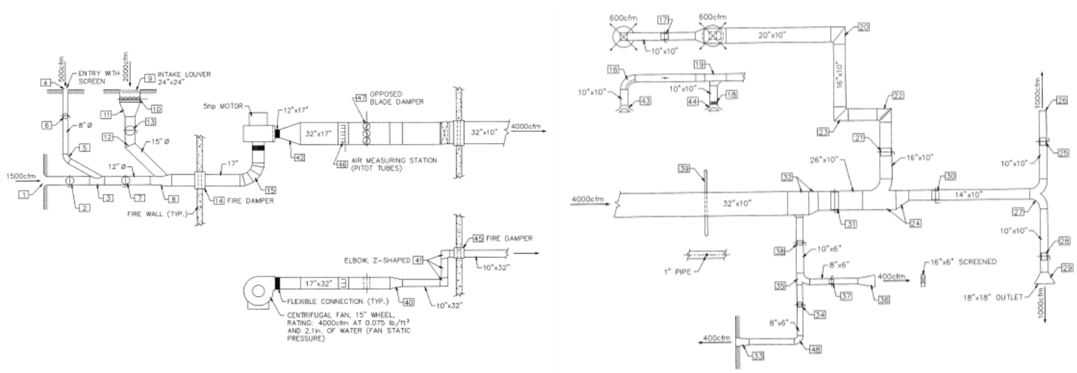


Figure 4: Schematic for Example 8

Table 1: Designed airflow for terminals in Example 8

Induced terminal	1	2	3
Designed flowrate	700 L/s	250 L/s	950 L/s
Supply terminal	4	5	6
Designed flowrate	275 L/s	275 L/s	475 L/s
Supply terminal	7	8	9
Designed flowrate	475 L/s	200 L/s	200 L/s

However, this duct system is unable to achieve desired air flow rate in each terminal without any adjustment of dampers. This fact is revealed by the mismatch of nodal pressure when calculating pressure drop by using designed airflow, which is demonstrated in Figure 5 (blue lines). By modeling this duct system by the proposed duct DAE approach in Simscape, shown in Figure 6 (i and ii), the actual airflow distribution can be calculated and the pressure drop pattern is obtained in the same time, shown in Figure 5 (green lines). In Figure 5, different nodes are distinct from each other by the locations in x-axis. The y-axis is the corresponding nodal pressure. Each line segment represents a part of duct. Air flows through the duct from high pressure to low pressure. Some nodes that have two lines merging together or one line separating apart are the T-junctions.

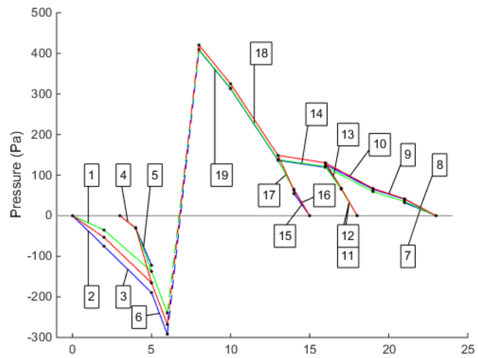
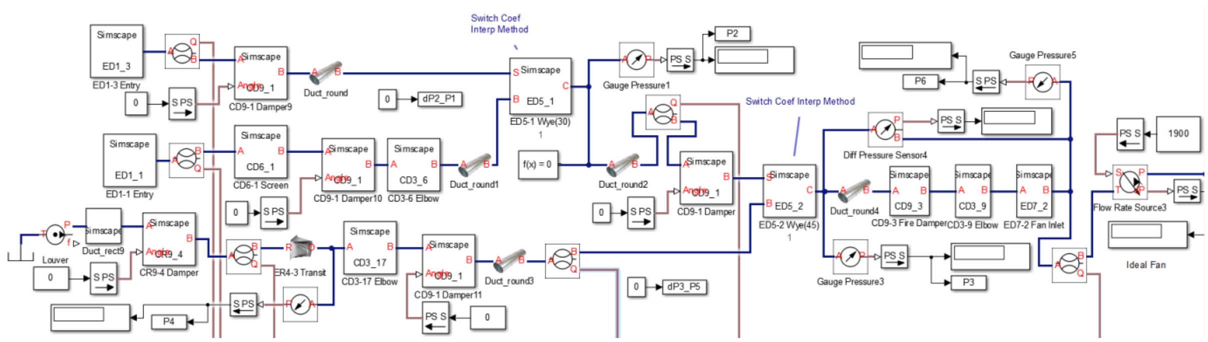
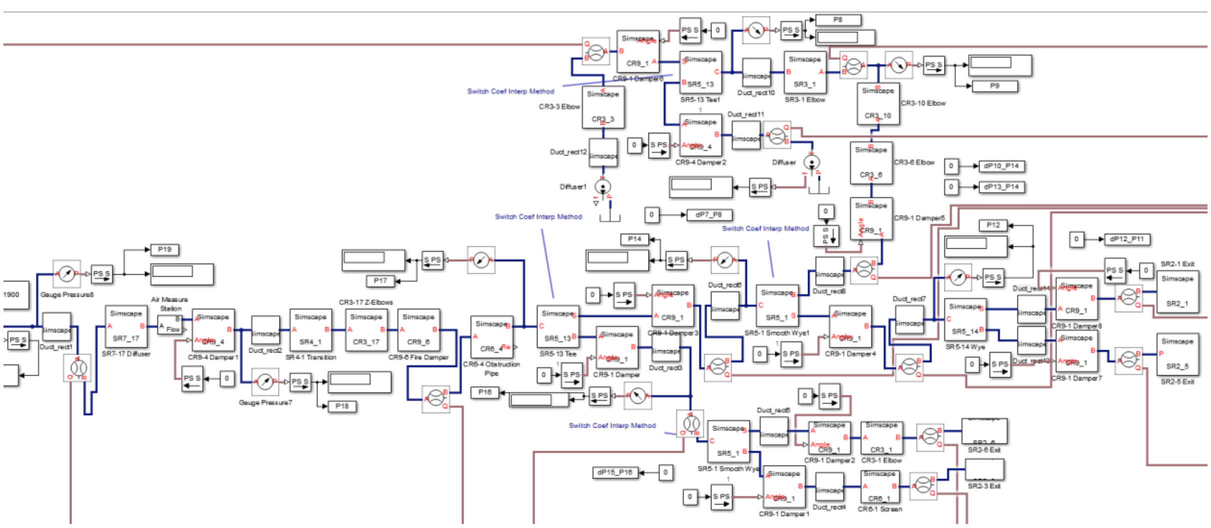


Figure 5: Nodal pressure before and after balancing in Example 8



(i) induced ventilation part before fan



(ii) supply distribution part after fan

Figure 6: DAE model for Example 8 in Simscape

By applying the balancing procedure described in previous sections, the balanced duct system is achieved, shown in Figure 5 (red lines). The detailed flowrates, nodal pressures and damper positions for balancing is listed in Table 2. To evaluate the level of imbalance, we calculate the maximum absolute percentage error (MAPE) of each terminal. The MAPE is calculated by $\max_i \left|q_i - q_{d,i}\right| / \left|q_{d,i}\right|$, where q_i is actual airflow for terminal i and $q_{d,i}$ is the corresponding designed airflow. It can be seen that using balancing method does achieve satisfactory results.

Table 2: Airflow and pressures details before and after balance for Example 8

Branch	Designed Case		Actual Case		Balanced Case		
	Q (L/s)	**dP (Pa)**	**Q (L/s)**	**dP (Pa)**	**Q (L/s)**	**dP (Pa)**	**Damper**
1	700	35.0129	690.1346	35.1671	702.7297	51.7309	5.7139°
2	250	57.0221	212.0255	35.1671	245.7971	51.7309	0.3983°
3	950	111.1274[a]	902.1602	101.9938	948.5268	110.7122	0.0005°
4	950	29.5259	997.8398	29.9911	951.4732	30.6775	3.5831°
5	950	92.5145[b]	997.8398	107.1698	951.4732	131.7657	14.0113°
6	1900	102.4568	1900.0	102.4568	1900.0	102.4568	N.A.
7	275	32.2125	335.6966	34.8530	275.4957	40.8245	15.6997°
8	275	40.7355	218.1797	34.8530	274.6282	40.8245	0.1520°
9	550	24.5789	553.8764	24.9200	550.1239	24.5898	N.A.
10	550	64.5241*	553.8764	61.4334	550.1239	64.5230	0.0444°
11	475	66.3677*	472.5109	65.6802	474.5057	68.3618	2.4564°
12	475	67.9720*	466.6739	65.6802	474.4890	68.3618	0.6127°
13	950	55.5232*	939.1848	55.5261	948.9946	61.5754	3.4866°
14	1500	17.4490	1493.1	17.2650	1499.1	17.4417	N.A.
15	200	55.5232*	216.5671	63.0497	200.4184	67.9792	0.0007°
16	200	66.9949	190.3718	63.0497	200.4630	67.9792	12.5280°
17	400	58.0070[c]	406.9388	75.4217	400.8814	79.3997	1.0215°
18	1900	176.2132*	1900.0	176.2132	1900.0	176.2132	N.A.
19	1900	95.7082*	1900.0	95.7082	1900.0	95.7082	10.6706°
MAPE			22.07%		1.68%		

[a]Due to different value of loss coefficient, value in Example 8 is 103.
[b]Due to different value of loss coefficient, value in Example 8 is 103
[c]Due to the change of loss coefficient as the database updates, value in Example 8 is 79
*Due to rounding error, the values are different from Example 8

The flow rates for each branch is plotted in the bar chart, shown in Figure 7. It can be seen clearly that the calculated damper position successfully balanced the duct system (yellow bars) to the desired flowrate (blue bars). Therefore, conclusion can be drawn that the proposed methods can calculate the suitable damper position for HVAC duct balancing.

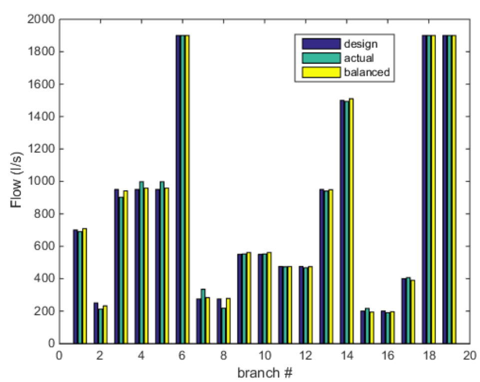


Figure 7: Airflow distribution before and after balancing in Example 8

6. CONCLUSIONS

We present a new non-iterative balancing method for HVAC duct system. The method contains three steps: 1) developing mathematical model for the duct system; 2) measuring pressures and flow rates to identify model parameters; 3) calculating optimal damper positions and adjusting dampers sequentially. This method distinguishes from other non-iterative TAB methods by its DAE model. By using Simscape modeling environment, developing duct system modeling becomes easier, and implementing optimization algorithms to calculate damper position for balancing becomes more convenient. This method has four main advantages: 1) the process is non-iterative and efficient; 2) the balancing result is highly accurate; 3) the procedure is easy to perform for engineers; and 4) the equipment requirement is only capture hood. The performance of this new TAB method is verified by the ASHRAE's example. The MAPE towards desired of the duct system reduces from 22.07% to 1.68%. This method saves the cost and effort of balancing duct systems. Hence this method has strong potential in providing balancing services for building HVAC system. In the future, the feasibility of this method in large scale HVAC system with hundreds of terminals must be studied. Besides, the obtained DAE model by this method can be further exploited. We will seek more applications on control and fault detection and diagnosis.

NOMENCLATURE

ΔP	Pressure drop	(Pa)
C_f	friction coefficient	(Dimensionless)
L	duct length	(m)
D	duct diameter	(m)
ρ	air density	(kg/m^3)
V	average air velocity	(m^3)
ε	wall roughness	(mm)
Re	Reynolds number	(Dimensionless)
$\boldsymbol{\beta}$	Model parameters	(-)
$\boldsymbol{\theta}$	Damper positions	(-)
$\mathbf{Z}$	Flowrate measurements	(m^3/s)
$\mathbf{C}$	Observation matrix	(-)
$\mathbf{X}$	State vector	(-)
$\boldsymbol{\Sigma}$	Measurement covariance	(-)
$\boldsymbol{\Omega}$	Parameter covariance	(-)
$\boldsymbol{\beta}_0$	Prior value of parameter	(-)
$\mathbf{C}_T$	Observation matrix	(-)

Subscript

0	Fully open state
d	Desired valued

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