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Thermal Effectiveness of Wall Indoor Fountain in Warm Humid Climate

J A P Seputra
University of Atma Jaya Yogyakarta, Jl. Babarsari 44, Yogyakarta, 55281, Indonesia

Abstract. Nowadays, many buildings wield indoor water features such as waterfalls, fountains, and water curtains to improve their aesthetical value. Despite the provision of air cooling due to water evaporation, this feature also has adverse effect if applied in warm humid climate since evaporation might increase air humidity beyond the comfort level. Yet, there are no specific researches intended to measure water feature’s effect upon its thermal condition. In response, this research examines the influence of evaporative cooling on indoor wall fountain toward occupant’s thermal comfort in warm humid climate. To achieve this goal, case study is established in Waroeng Steak Restaurant’s dining room in Surakarta-Indonesia. In addition, SNI 03-6572-2001 with comfort range of 20.5–27.1°C and 40-60% of relative humidity is utilized as thermal criterion. Furthermore, Computational Fluid Dynamics (CFD) is employed to process the data and derive conclusions. Research variables are: feature’s height, obstructions, and fan types. As results, Two Bumps Model (ToB) is appropriate when employs natural ventilation. However, if the room is mechanically ventilated, Three Bumps Model (TeB) becomes the best choice. Moreover, application of adaptive ventilation is required to maintain thermal balance.

Keywords: Computational Fluid Dynamics; Evaporative Cooling; Indoor Water Fountain; Room Ventilation; Thermal Comfort

1. Introduction
According to Al-Obaidi et al, warm humid region as Indonesia has uncomfortable climatic condition because it receives huge amount of solar radiation along with long period of clear day resulting in high temperature and humidity all year [1]. Therefore, the effective passive design technique might be used to improve indoor thermal comfort is ‘comfort ventilation’ [2;3;4]. However, naturally ventilated building always has problems with low wind speed which often occurs in equatorial belt countries. In response, addition of ancillary tools to increase wind speed by using ceiling fans are necessary to provide cooling in naturally ventilated buildings [5].

Another comfort strategy which follows ventilation is the evaporative cooling. Analogous with sweat on human's skin, evaporation happens when wind blows on sweat's surface and transform the substance's liquid form into gaseous form. During this process, huge amount of heat, known as heat of vaporization, is absorbed and produces cooling effect toward its surroundings. Although there are many findings stated that evaporative cooling would not be effective in humid environment and could only be applied in a condition which does not require rigorous comfort’s requisite [6;7], in fact, people in humid climates have acclimatized to be accustomed to 45-80% relative humidity and wind speed of 0.2-0.5m/s [8;9;10]. To assess the limitation above, a research was conducted in Thailand by examining occupant’s thermal sensation due to indoor wall fountain inside a conditioned room. This research proves that the occupants feel 0.1–0.3°C cooler when subjected to water feature [11].
Moreover, air cooling by evaporative mist is becoming popular in Japan to cool cities as a solution of Urban Heat Island (UHI). Although Japan has a humid climate, its effective temperature shows that increased humidity is not significant compared to the improvement of thermal comfort due to evaporated water mist [12]. Due to those contentions and the dearth of academic records on evaporative cooling in naturally ventilated indoor space, this research is aimed to figure out the effect of indoor water feature toward occupant’s thermal comfort in warm humid climate. Indoor wall fountain is chosen as the most popular water feature installed inside building in Indonesia.

Over the last decades, numerical studies aided by computer software have been implemented by researchers. For instance, Prianto (2012), employs computer simulation to test balcony effect and room furniture layout in naturally ventilated building toward its thermal comfort condition [13]. Almhafdy et al. (2015), used Computational Fluid Dynamics (CFD) to figure out the performance of natural ventilation occurred at courtyard by experimenting on aspect ratios and cantilevered roofs upon wind speed [14]. Furthermore, a numerical validation on CFD was conducted by developing evaporative cooling system inside a greenhouse. The result was satisfactory and concluded that correct amount of ventilation rate is the main factor to improve the efficiency of evaporative cooling system [15]. Therefore, this research also harnesses CFD to figure out the effectiveness of indoor wall fountain in warm humid climate based on two thermal indices, i.e. air temperature and relative humidity.

2. Methodology

Research is divided into four main stages; validation, simulation, analysis, and conclusion. Validation is initialized by developing a case study in Waroeng Steak Restaurant’s dining room in Nusukan, Surakarta. Field measurements are done by retrieving the data of air temperature and relative humidity inside and outside the dining room by two HOBO loggers. Each device is set to store the data within 5 minutes interval and activated simultaneously by built-in timer. Data recording is undertaken for 2 weeks to store dynamism of occupant activities and weather conditions. Field results are then compared with the results produced by identical CFD model to figure out the level of agreement.

CFD Simulation is commenced after the validation produces error below 5%. Research variables developed are; feature’s height (low, mid, tall), number of obstructions (one, two, three), and types of fans (forward, backward, ceiling). Each of them is simulated sequentially to obtain best combination of wall indoor fountain components. Thermal performance is measured by investigating the cooling effect and distribution pattern of indoor air temperature and relative humidity.

Analysis stage consists of graphical examination to find the balance of air temperature and relative humidity. Distribution patterns are measured by evenness factor, expressed as a fraction of 0 to 1, 1 represents complete mixing and good uniformity inside the room whereas 0 shows no mixing at all.

SNI 03-6572-2001 is used as criterion in Indonesia since this standard has been contextually adapted for warm humid climate [16]. This standard is commonly used as environmental benchmark, such as a research in Dormitory Building of Universitas Sam Ratulangi Indonesia, it was conducted to measure comfort, ventilation, heat and lighting to confirm its energy efficient design [17]. SNI was also utilized to analyze the role of natural ventilation on seven stories building of Economy and Business Faculty Universitas Brawijaya Indonesia for reducing electric consumption and determine the dimension of fenestration as a recommendation [18]. Furthermore, SNI was the basis of a building assessment in Jember Fashion Carnival Center to achieve thermal comfort for the occupants [19].

| Dry bulb Temperature | Relative Humidity | Air Movement | Surface Radiation |
|----------------------|-------------------|--------------|------------------|
| Cool, 20.5-22.8°C    | Advisable, 40-50% | 0.1m/s at 25°C | Similar with DBT |
| Neutral, 22.8-25.8°C | Tight, 55-60%    | 0.2m/s at 26.8°C |
| Warm, 25.8-27.1°C    |                   | 0.25m/s at 26.9°C |
|                      |                   | 0.3m/s at 27.1°C |
|                      |                   | 0.35m/s at 27.2°C |

Table 1. Comfort criteria of SNI 03-6572-2001
Lastly, the conclusion stage points out important results and findings. These outcomes would be beneficial to provide guidelines for interior designers or room planners to effectively incorporate water features inside building in warm humid climate. Identified research weaknesses and suggestions for future research are also presented in this section.

2.1. Equation of Water Pools' Evaporation Rate
Evaporation rate from a pool of water due to the direct contact with dry air on its surface is expressed by equation below [20],
\[ gh = \Theta A (xs - x) \]
where,
- \( gh \) = amount of evaporated water per hour (kg/h),
- \( \Theta = (25 + 19 v) \) = evaporation coefficient (kg/m²h),
- \( v \) = velocity of air above the water surface (m/s),
- \( A \) = water surface area (m²),
- \( xs \) = humidity ratio saturated air at the same temperature as the water surface (kg/kg) (kg H₂O in kg Dry Air)
- \( x \) = humidity ratio air (kg/kg) (kg H₂O in kg Dry Air)

Therefore, evaporation rate is affected by air velocity on the water surface, contrast between air and water’s humidity ratio, and water surface exposure. This implies that faster wind, drier air, and greater exposure of water to the adjacent air will bolster evaporation as portrayed by Figure 1 below.

![Figure 1. Water evaporation from static pool](image)

3. Results and Discussion
Research presents the utilization of Computational Fluid Dynamics (CFD) in transient mode due to the time dependent case. Accuracy of CFD program is determined by its ability in incorporating heat transfer between two phase of fluids, i.e water and air. This phase change phenomenon or evaporation is exploited throughly in this paper. However, validation on CFD in simulating the case is necessary to figure out CFD's degree of agreements with the real environment.

3.1. CFD Validation
To achieve good level of certainty in using CFD, a validation study is established in Waroeng Steak Restaurant's dining room, Jalan Adi Sumarmo No. 133 Nusukan – Surakarta (see Figure 2). Research is focused on the wall fountain installed inside the dining room as shown in Figure 3 below.

![Figure 2. Dining Room of "Waroeng Steak".](image)

![Figure 3. Indoor Wall Fountain.](image)

Field measurement is conducted to figure out the thermal effect of indoor wall fountain inside the dining room toward its immediate surroundings. Two HOBO data loggers are placed at different locations as depicted by Figure 4 and Figure 5 below. The first device is installed near the fountain to
measure its thermal effect and the second is tied on nearby door to record initial air temperature and relative humidity before being affected by the water fountain.

![Figure 4. Measurement tool near the fountain.](image1)

![Figure 5. Measurement tool on entrance's wall.](image2)

Outdoor air temperature of 33°C and relative humidity of 55% are recorded as the prevailing condition. Water fountain's flow rate is 0.24 m³/s and the incoming wind speed is 0.1 m/s. Figure 6 and Figure 7 below illustrate CFD domain and point probes aimed to extract the values of air temperature and relative humidity.

![Figure 6. CFD model and probe (perspective).](image3)

![Figure 7. CFD model and probe (plan).](image4)

Values taken from field measurement are 32.9°C and 58.4%, whereas simulation shows 31.85°C and 60% respectively. Therefore, they have good agreement since the deviations or errors do not exceed 5% in each parameters, i.e. 3.34% and 2.68%.

### 3.2. CFD Simulation and Results

To obtain numerical values of air temperature and relative humidity, sets of line probes are modeled as depicted by Figure 8 and Figure 9 below.

![Figure 8. CFD model and line probes (perspective).](image5)

![Figure 9. CFD model and line probes (plan).](image6)

First, simulation is begun by creating three kinds of falling distance model; tall, mid, and low height without obstacles. After simulation finished, CFD produces images of propagating air temperature and corresponding relative humidity as shown in Table 2 below.
Table 2. Examples of CFD results depicting air temperature and relative humidity contours.

| Tall Model | Mid Model | Low Model |
|------------|-----------|-----------|
| **Air Temperature Contours** | **Relative Humidity Contours** | **Air Temperature Contours** |
| ![Image](image1.png) | ![Image](image2.png) | ![Image](image3.png) |
| ![Image](image4.png) | ![Image](image5.png) | ![Image](image6.png) |
| ![Image](image7.png) | ![Image](image8.png) | ![Image](image9.png) |
| ![Image](image10.png) | ![Image](image11.png) | ![Image](image12.png) |
| ![Image](image13.png) | ![Image](image14.png) | ![Image](image15.png) |
| ![Image](image16.png) | ![Image](image17.png) | ![Image](image18.png) |

Since showing all the contour images as above is too inefficient, research combines the calculation of air temperature and relative humidity into graphs as portrayed in Table 3 below. Fluctuating lines indicate uneven air temperature and humidity distribution while straight lines show better uniformity.
Table 3. Comparisons of air temperature and relative humidity on height experiments.

| Tall Model | Mid Model | Low Model |
|------------|-----------|-----------|
| **Air Temperature** | **Air Temperature** | **Air Temperature** |
| **Relative Humidity** | **Relative Humidity** | **Relative Humidity** |

Tall Model possesses greatest potential in cooling the room but provides poor distribution. Next, Two and Three Bumps Model are simulated and the results are shown in Table 4 below.

Table 4. Comparisons of air temperature and relative humidity on bump experiments.

| One Bump (OeB) | Two Bumps (ToB) | Three Bumps (TeB) |
|----------------|----------------|------------------|
| **Air Temperature** | **Air Temperature** | **Air Temperature** |
| **Relative Humidity** | **Relative Humidity** | **Relative Humidity** |
Two Bumps (ToB) Model possesses higher evaporation rate due to faster waterfalls and better uniformity. After that, fans are turned on to enhance wind speed. However, since there are no clear performance differences between two and three bumps model, both of them are simulated again. Fans are created in two types, i.e. two walls and one ceiling mounted fans, and specified as Table 5 below.

| Table 5. Fan’s technical specifications |
|----------------------------------------|
| Fan’s Type | Radius | Hub Radius | Speed  | Thickness | Blade angle |
| Wall       | 0.4 m  | 0.1 m      | 1500 rpm | 0.2 m     | 14 deg     |
| Ceiling    | 0.8 m  | 0.1 m      | 350 rpm  | 0.2 m     | 14 deg     |

Fans are arranged symmetrically as shown by Table 6 below, i.e. Forward Fans blow air to the room, Backward Fans direct air to the fountain, and Ceiling Fans send the air down to the workplanes.

| Table 6. Fans installation inside the examined room along with the trajectories. |
|----------------------------------------|
| Forward Fans (FFans) | Backward Fans (BFans) | Ceiling Fans (CFans) |

Two Bumps Model (ToB) is simulated with mechanical assistance by turning on the fans. The results are shown in Table 7 below.

| Table 7. Comparisons of air temperature and relative humidity on fan experiments in ToB. |
|----------------------------------------|
| FFans | BFans | CFans |

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Image: Two Bumps (ToB) Model possesses higher evaporation rate due to faster waterfalls and better uniformity.

Image: Fans are created in two types, i.e. two walls and one ceiling mounted fans, and specified as Table 5 below.

Image: Fans are arranged symmetrically as shown by Table 6 below, i.e. Forward Fans blow air to the room, Backward Fans direct air to the fountain, and Ceiling Fans send the air down to the workplanes.

Image: Two Bumps Model (ToB) is simulated with mechanical assistance by turning on the fans. The results are shown in Table 7 below.
Better result is produced by ceiling fans due to the closely stacked and straightest lines. Lastly, examination is completed by simulating fans for Three Bumps Model (TeB).

Table 8. Comparisons of air temperature and relative humidity on fan experiments in TeB.

| Fan Type | 20 seconds Avg Tmp (°C) | 20 seconds Avg RH (%) | 40 seconds Avg Tmp (°C) | 40 seconds Avg RH (%) | 60 seconds Avg Tmp (°C) | 60 seconds Avg RH (%) | Evenness at 60 s |
|----------|--------------------------|------------------------|--------------------------|------------------------|--------------------------|------------------------|-----------------|
| FFans    | 32.82                    | 54.95%                 | 31.71                    | 85.20%                 | 32.38                    | 67.04%                 | 0.94            |
| BFans    | 32.99                    | 49.38%                 | 32.91                    | 49.06%                 | 32.85                    | 48.56%                 | 0.78            |
| CFans    | 33.00                    | 49.17%                 | 32.97                    | 48.37%                 | 32.95                    | 47.58%                 | 0.96            |
| Tall     | 32.34                    | 60.60%                 | 31.72                    | 81.66%                 | 30.39                    | 100.00%                | 0.87            |
| Mid      | 32.99                    | 59.00%                 | 31.69                    | 85.70%                 | 30.55                    | 99.25%                 | 0.74            |
| Low      | 33.00                    | 62.38%                 | 31.13                    | 84.15%                 | 30.49                    | 95.95%                 | 0.88            |
| ToB_NoFans | 31.92                | 69.07%                 | 31.59                    | 94.02%                 | 30.62                    | 100.00%                | 0.77            |
| TeB_NoFans | 32.35                | 72.35%                 | 31.02                    | 86.45%                 | 29.96                    | 97.58%                 | 0.80            |

Greater oscillations of air temperature and relative humidity occurred at the beginning of simulation as shown by Table 8 above. It indicates the dominance of airflow exerted by fans over the evaporation process. However, as the time goes by 60 seconds all the thermal conditions have been mixed up and produce less diversities, especially on ceiling fans’ case.

4. Conclusions
To identify the optimum combinations simulated by CFD, numerical values of air temperature and relative humidity are extracted and averaged in the Table 9 below. Evenness is added at the rightmost column as a ratio of minimum to its average value after 60 seconds to determine air uniformity level, value of 1 represents a perfect mixing and value of 0 means no mixing at all.

Table 9. Averaged results of each simulated model by recorded time

| Model Type   | 20 seconds Avg Tmp (°C) | 20 seconds Avg RH (%) | 40 seconds Avg Tmp (°C) | 40 seconds Avg RH (%) | 60 seconds Avg Tmp (°C) | 60 seconds Avg RH (%) | Evenness at 60 s |
|--------------|--------------------------|------------------------|--------------------------|------------------------|--------------------------|------------------------|-----------------|
| Tall         | 32.82                    | 54.95%                 | 31.71                    | 85.20%                 | 32.38                    | 67.04%                 | 0.94            |
| Mid          | 32.99                    | 49.38%                 | 32.91                    | 49.06%                 | 32.85                    | 48.56%                 | 0.78            |
| Low          | 33.00                    | 49.17%                 | 32.97                    | 48.37%                 | 32.95                    | 47.58%                 | 0.96            |
| ToB_NoFans   | 32.34                    | 60.60%                 | 31.72                    | 81.66%                 | 30.39                    | 100.00%                | 0.87            |
| TeB_NoFans   | 32.48                    | 59.00%                 | 31.69                    | 85.70%                 | 30.55                    | 99.25%                 | 0.74            |
| ToB_FFans    | 31.92                    | 73.19%                 | 31.13                    | 84.15%                 | 30.13                    | 98.30%                 | 0.78            |
| ToB_BFans    | 32.35                    | 62.38%                 | 31.23                    | 84.15%                 | 30.49                    | 95.95%                 | 0.88            |
| ToB_CFans    | 32.25                    | 69.07%                 | 31.59                    | 94.02%                 | 30.62                    | 100.00%                | 0.77            |
| TeB_FFans    | 31.95                    | 72.35%                 | 31.02                    | 86.45%                 | 29.96                    | 97.58%                 | 0.80            |
According to the results above, there is not even a single case manages to reach air temperature below 27.1°C since the outside air itself was 33°C, far higher than the expected SNI standard. As response, research is reoriented to find the lowest temperature possible and use the upper limit of 80% relative humidity taken from Chow, et.al. (2010) as requisite to determine the best configuration. Consequently, the highlighted numbers are the cases which fulfill the requirements above.

By applying those new rules, ToB is better than TeB since it produces cooler average air temperature (31.48°C to 31.57°C) and provides better evenness ratio (0.87 to 0.74). Meanwhile, FFans Model reduces air temperature better than the other models by increasing air velocity on water surface temperature (31.48°C to 31.57°C) and provides slightly better uniformity (0.83 to 0.81).

Nevertheless, all simulated cases are having problem with high relative humidity (>80%) after 20 seconds. It suggests that the ventilation rate should be increased to replace old stale air after 20 seconds to lower the humidity hence maintain indoor thermal comfort much longer. Still, a fluctuating outdoor air condition will be a real challenge to solve this problem. Thus, implementation of sophisticated technology such as thermal sensors and actuators is necessary. Additionally, future study to examine adaptive ventilation technology as well as subjective responses from local residents is required to figure out whole thermal effectiveness of evaporating water feature toward indoor thermal comfort in warm humid climate.

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