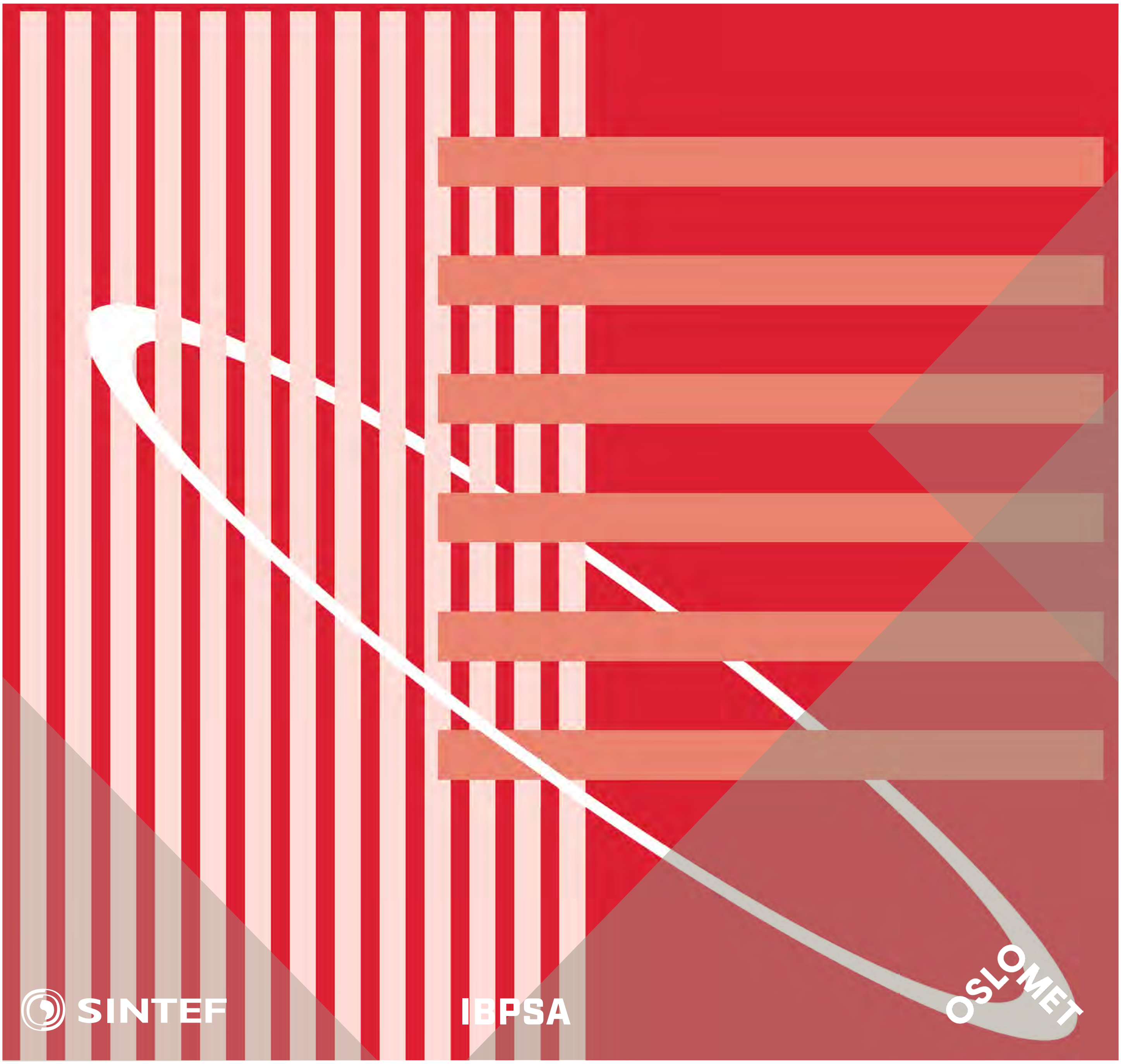
International Conference Organised by
IBPSA-Nordic, 13th–14th October 2020,
OsloMet

SINTEF
PROCEEDINGS
5

BuildSIM-Nordic 2020

Selected papers



SINTEF Proceedings

Editors:
Laurent Georges, Matthias Haase, Vojislav Novakovic and Peter G. Schild

BuildSIM-Nordic 2020

Selected papers

International Conference Organised by IBPSA-Nordic,
13th–14th October 2020, OsloMet

SINTEF Academic Press

SINTEF Proceedings no 5
Editors:
Laurent Georges, Matthias Haase, Vojislav Novakovic and Peter G. Schild
BuildSIM-Nordic 2020
Selected papers
International Conference Organised by IBPSA-Nordic,
13th–14th October 2020, OsloMet

Keywords:
Building acoustics, Building Information Modelling (BIM), Building physics, CFD and air flow, Commissioning and control, Daylighting and lighting, Developments in simulation, Education in building performance simulation, Energy storage, Heating, Ventilation and Air Conditioning (HVAC), Human behavior in simulation, Indoor Environmental Quality (IEQ), New software developments, Optimization, Simulation at urban scale, Simulation to support regulations, Simulation vs reality, Solar energy systems, Validation, calibration and uncertainty, Weather data & Climate adaptation, Fenestration (windows & shading), Zero Energy Buildings (ZEB), Emissions and Life Cycle Analysis

Cover illustration: IBPSA-logo

ISSN 2387-4295 (online)
ISBN 978-82-536-1679-7 (pdf)

SINTEF Academic Press
Address: Børrestuveien 3
PO Box 124 Blindern
N-0314 OSLO
Tel: +47 40 00 51 00

www.sintef.no/community
www.sintefbok.no

SINTEF Proceedings
SINTEF Proceedings is a serial publication for peer-reviewed conference proceedings on a variety of scientific topics.
The processes of peer-reviewing of papers published in SINTEF Proceedings are administered by the conference organizers and proceedings editors. Detailed procedures will vary according to custom and practice in each scientific community.

Impact of AC Outdoor Unit Placement on Energy Efficiency

Krishna Patel[1], Rajan Rawal[2]
[1]Faculty of Technology, CEPT University, Ahmedabad, India
[2]Centre for Advanced Research in Building Science and Energy, CEPT University, Ahmedabad, India
corresponding author: rajanrawal@cept.ac.in

Abstract

Growing urbanisation in India has led to the increasing development of high-rise buildings. To maintain the building aesthetics, the outdoor air conditioner (AC) units are stacked withing a recessed space. The heat rejected from these outdoor units (ODU) leads to increase in air temperature of the recessed space. This causes inefficient working of AC units. The paper reports the difference in results when on-site measurement data were compared to CFD simulation results. The on-site measurements were taken for six storeys of an eleven-storey building. The different turbulence models were studied, and it was concluded that the NK turbulence model had the least percentage difference of 8.89% when compared to the site data. Further, NK model was used to study the placement of ODUs in recessed spaces having varying depth and width. When the horizontal distance between the ODUs increased by 1m, maximum condenser on-coil temperatures reduced by 6°C. The studies concluded that, even by allocating lesser area to the recessed space, but by optimizing the depth and width of the space, the temperature increase of the microclimate can be avoided and AC efficiency can be maintained.

Introduction

Currently, India has 6% share in total global primary energy demand which is expected to increase to 11% by 2040. Easy access to electricity and increased prosperity has increased the demand for luxury and better standard of living. As a result, the use of AC has become very common in India. A 40% reduction in electricity demand can be achieved in India if efficient air conditioning systems are adopted (Phadke et al., 2014).

According to the Global cooling report mini-split ACs or Window ACs constitute about 70% of the total AC used (Campbell et al., 2018). Energy used by the Room ACs from 2016 to 2050 is expected to increase five folds in developing countries. Cooling energy demand in India from RACs in 2016 was 94 TWh which is expected to rise to 1890 TWh in 2050. The growth rate of sales of AC unit is 10%-15% in one year. The energy use per-capita for space cooling in India is expected to rise from 72 kWh to 1140 kWh making it one of the world's largest consumer by 2050.

To cater to this, constant efforts are made to increase the efficiency of the AC. Though the system efficiency has been looked upon and improved to a great extent, not much care has been taken in the installation practices. Stacking ODU is one such wrongly implemented practice that results in inefficient working of AC units. The use of split AC has increased due to ease of installation and its cost effectiveness.

Due to growing population and increase in urbanisation, India focuses more on vertical development. As a result, the number of high-rise apartments is constantly increasing. The ODUs in the high-rise apartments are usually stacked vertically one above the other in a recessed space. If they operate simultaneously, they reject heat in the space. Due to buoyancy effect of hot air, it rises upwards. The hot air accumulates near the ODUs at the top. Due to increasing temperatures near the inlet area of the ODUs on top floors, the efficiency of Air Conditioners decreases. As a result, energy consumption increases.

To understand the impact of stack effect due to heat dissipation by the condenser units, number of studies have been done using CFD simulations.

The relation between building height and condenser on-coil temperature was obtained by studying buildings heights of 70m, 98m and 126m. It was concluded that while increasing the height from 70m to 126m, a temperature increase of 2°C was observed (T. T. Chow & Lin, 1999). The on-coil temperatures for a building re-entrant and lightwells were studied. To verify the simulation results, a laboratory experiment was conducted on the model of 1:100 scale. Similar results were obtained by the simulation method and laboratory experiment (Tin Tai Chow et al., 2000). The different shapes i.e. of re-entrants I, L and T were studied (T. T. Chow et al., 2000). The I-shaped, L-shaped and T-shaped re-entrant had maximum temperatures up to 42.5°C, 39.5°C and 37.5°C respectively. Condenser Group Performance Indicator (CGPI) was used to assess the group performance of AC. The impact of depth of recessed space on on-coil temperatures. Depths of the re-entrants were taken to be 6m and 10m and it was observed that in deeper recessed space, mass flow rate reduced by 9% over 16^{th} floor. The studies showed that for inner condenser units deeper recessed space shows a lower rise in temperature compared to shallow space by 42% at 5^{th} storey. For outer condenser units, the deeper recessed space showed a higher rise in temperature compared to the shallow recessed space by 234% at 5th storey and 53% at the last story (Bojic et al., 2002). As various studies were done for high rise recessed space, the placement configuration of condenser units in a plant room of low-rise residential building were studied. Increment of 1ºC temperature of condenser on-coil temperature, COP of AC unit drops by 3%. The study suggested that the placement of compressors should be such that outlet air should not return to the condenser (T. T. Chow et al., 2002). The impact of building height on the performance of ODUs placed in a plant room with different wind velocity was studied. It was concluded that if condenser on-coil temperatures increases to more than 48°C, the efficiency of AC drops by 9.4% to 25.5% based on entrant area shape and wind velocity (Choi et al., 2005). The impact of louver (angle and spacing) and placement configuration (horizontal-vertical and distance) of ODU, concluded that optimum louver angle and louver spacing was 80° and 50mm respectively. The vertical arrangement proved to be better than horizontal arrangement (Duan et al., 2016).

On-site experiment to understand the horizontal and vertical profile near the outdoor air condenser units was conducted. According to studies, on the 10th and 20th floor, the temperature rose by 10°C and 19°C respectively. Considering the CGPI metric, 18% drop was observed in the performance of AC units (Bruelisauer et al., 2014).

The existing studies are either simulation-based, or experiment-based. The validation of simulation results by on-site experiment data is yet not achieved. This study aims to compare the CFD simulation results with the measurement data collected at site and further optimize the area of recessed space for ODU placement.

Methodology

Data collection on site

The data was collected for 6 storeys of Casa Riva Hotel, Surat. According to The National Building Code of India, Surat has hot and dry climate. The measurements were taken on the outdoor air conditioner units. The ambient air temperature of the outdoor environment was measured using Onset HOBO data loggers at an interval of 5 minutes for 3 hours. The air velocity at the outdoor environment was measured using Velocicalc TSI at 40 minutes interval for 3 hours. For initial conditions, the average values of ambient outdoor air temperature and air velocity were considered. The specifications of the ODU were noted to use them as inputs for modelling.

Table 1: Initial Conditions of Computational Domain

Parameter	Values
Outdoor Ambient Temperature	28.7°C
Wind Velocity	2.2 m/s

Table 2: ODU Specifications for modelling inputs

Parameters	Values
Capacity	3500 Watts (1 Ton)
Dimensions of ODU	0.72m x 0.56m x 0.26m
Inlet air area from Back	0.6m x 0.4m
Inlet air area from Side	0.15m x 0.3m
Area of Outlet Air	0.4m x 0.4m
Volume flow rate at ODU outlet	0.2 m³/s

The condenser on-coil temperature and outlet air temperature were measured at each compressor unit. These temperatures at the ODUs were further compared to the temperatures given by the CFD simulation model.

CFD Simulation Model

Only one façade was taken for simulation. Based on it building dimensions of 5m x 5m were taken. The building height was kept 18m with each floor being 3 meters high. The ODUs were placed at 0.23m from the wall. The depth and the width of window shades were 0.25m and 2.5m respectively. They were placed at 2.1m height from the floor level at each floor. The vertical distance between the ODUs was 3m.

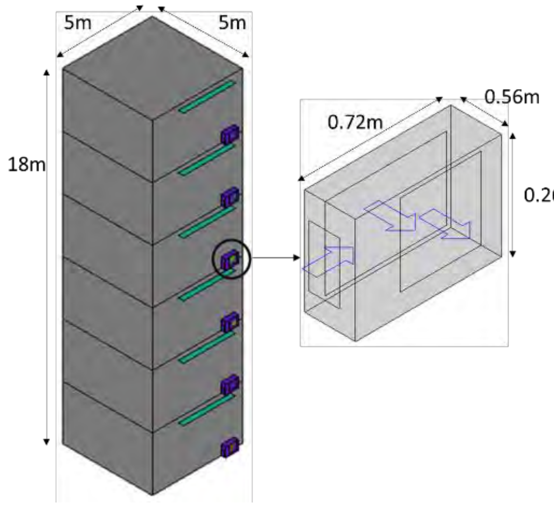


Figure 1: Geometry of the building and ODU

The capacity of the air conditioner was 3.5kW. The measured outlet air flow rate was 0.2 m^3/s. The area out outlet and inlet of condensing unit is shown in Figure 2.

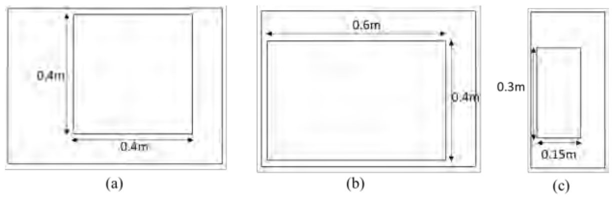


Figure 2: AC outdoor unit dimensions (a) Outlet Area (b) Inlet air (back) (c) Inlet air (side)

Governing Equations:

Mass Conservation Equation:

$$\frac{\partial u_i}{\partial x_i} = 0 \quad (1)$$

Where,

u_i = Velocity in i direction

x_i = Coordinate

Momentum Conservation Equation:

$$\frac{\partial \rho u_i}{\partial t} + \frac{\partial u_i \rho u_i}{\partial x} = -\frac{\partial p}{\partial x_j} + \frac{\partial}{\partial x_j}\mu\frac{\partial u_i}{\partial x_j} - \rho g_i \beta (T - To) \quad (2)$$

t = time

T = temperature

ρ = density of fluid

p = pressure

μ = viscosity

β = Coefficient of Thermal expansion

T_o = reference temperature

Energy Conservation Equation:

$$\frac{\partial \rho CpT}{\partial t} + \frac{\partial u_j \rho CpT}{\partial x_j} = \frac{\partial}{\partial x_j} K \frac{\partial T}{\partial x_j} + q \quad (3)$$

Cp = specific heat at constant pressure

K = Thermal conductivity

q = Heat source

Turbulence Models used:

To predict the best-suited model, four different mathematical models that are used to analyse the turbulence of the fluid. Turbulence models studied here are: Standard k-ε model, Re-Normalization Group analysis k-ε model (RNG), Modified Production k-ε model (MP), improved LK (Launder-Kato) k-ε model and NK (Nagano and Kim) two-equation heat transfer model.

Standard k-ε model:

$$\frac{\partial \rho k}{\partial t} + \frac{\partial u_i \rho k}{\partial x_i} = \frac{\partial}{\partial x_i}\left(\frac{\mu_t}{\sigma_k}\frac{\partial k}{\partial x_i}\right) + G_s + G_T - \rho\varepsilon \quad (4)$$

$$\frac{\partial \rho \varepsilon}{\partial t} + \frac{\partial u_i \rho \varepsilon}{\partial x_i} = \frac{\partial}{\partial x_i}\left(\frac{\mu_t}{\sigma_\varepsilon}\frac{\partial \varepsilon}{\partial x_i}\right) + C_1 \frac{\varepsilon}{k}(G_s + G_T)\left(1 + C_3 R_f\right) - C_2 \rho \varepsilon^2 \quad (4.1)$$

Where,

$$G_s = \mu_t \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\frac{\partial u_i}{\partial x_j} \quad (4.2)$$

$$G_T = g_i \beta \frac{\mu_t}{P_{rt}}\left(\frac{\partial T}{\partial x_i}\right) \quad (4.3)$$

$$R_f = -\frac{G_T}{G_S + G_T} \quad (4.4)$$

k = turbulence kinetic energy

ε = turbulence dissipation rate

Re-Normalization Group Analysis (RNG) model: Standard k-ε model consideres the constants based on experimental methods. In the RNG model, the constants have been defined using theoretical methods using Fourier analysis.

Table 3: Constants adopted in RNG model

σ_k	σ_ε	C_1	C_2	C_3	C_μ
0.719	0.719	C_1 (η)	1.68	0	0.085

$$C_1(\eta) = 1.42 - \frac{\eta\left(1 - \frac{\eta}{4.38}\right)}{1 + 0.012\eta^3} \quad (5)$$

$$\eta = \frac{k}{\varepsilon} s \quad (5.1)$$

$$s = \left\{\frac{1}{2}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right\}^{½} \quad (5.2)$$

Modified Production (MP) k-ε model: This model is used to compensate the overestimated turbulent kinetic energy, k, near the stagnation point of the standard k-ε model.

Where, $G_s = \rho \check{C}_\mu \frac{k}{\varepsilon} s\Omega$ (6)

Where,

$$\check{C}_\mu = \min\left(0.09, \frac{0.3}{1\ 0.35S^{1.5}}\right) \quad (6.1)$$

$$S = \min\left(20, \frac{k}{\varepsilon} S\right) \quad (6.2)$$

$$s = \left\{\frac{1}{2}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right\}^{½} \quad (6.3)$$

$$\Omega = \left\{\frac{1}{2}\left(\frac{\partial u_i}{\partial x_j} - \frac{\partial u_j}{\partial x_i}\right)\left(\frac{\partial u_i}{\partial x_j} - \frac{\partial u_j}{\partial x_i}\right)\right\}^{½} \quad (6.4)$$

$$\mu_t = \check{C}_\mu \rho \frac{k^2}{\varepsilon} \quad (6.5)$$

Improved LK (Launder-Kato) k-ε model: The fact that Ω in the above equation gets smaller around the stagnation point is taken into consideration. Based on that G_s is further modified.

$$Gs = \begin{cases} v_t, & \Omega/s \geq 1 \\ v_t S\Omega, & \Omega/s < 1 \end{cases} \quad (7)$$

$$v_t = C_\mu \frac{k^2}{\varepsilon} \quad (7.1)$$

NK model: In the NK turbulence model, instead of Prandtl number and eddy viscosity μ_t, the temperature variance t^2 and its dissipation rate ε_t are used.

Temperature Variance equation:

$$\frac{\partial \rho \overline{t^2}}{\partial t} + \frac{\partial \rho \overline{u_i}\,\overline{t^2}}{\partial x_i} = \frac{\partial}{\partial x_j}\left\{\left(\frac{K}{cp} + \frac{\rho \alpha t}{\sigma h}\right)\frac{\partial t^2}{\partial x_j}\right\} + Pt - 2\rho\varepsilon_t + D_t \quad (8)$$

Dissipation rate equation:

$$\frac{\partial \rho \varepsilon_t}{\partial t} + \frac{\partial \rho \overline{u_i} \varepsilon_t}{\partial x_i} = \frac{\partial}{\partial x_j}\left\{\left(\frac{K}{cp} + \frac{\rho \alpha_t}{\sigma_\varphi}\right)\frac{\partial \varepsilon_t}{\partial x_j}\right\} + \frac{\varepsilon_t}{t^2}\left(C_{p1} f_{p1}\frac{P_t}{2} - C_{D1} f_{D1} \rho \varepsilon_t\right) + \frac{\varepsilon_t}{k}\left(C_{P2} f_{P2} G_s - C_{D2} f_{D2} \rho \varepsilon\right) + E_t \quad (8.1)$$

Where: $P_t = -2\rho\overline{u_j t}\partial\overline{T}/\partial x_j$ denotes the production term of the temperature variance, D_t and E_t are additional terms.

$$-\overline{u_j t} = \alpha_t \frac{\partial \overline{T}}{\partial x_j} \tag{8.2}$$

Table 4: Constants adopted in NK turbulence models

C_{p1}	C_{P2}	C_{D1}	C_{D2}	σ_h	σ_f
1.8	0.72	2.2	0.8	1.0	1.0
f_{p1}	f_{P2}	f_{D1}	f_{D2}	D_t	E_t
1	1	1	1	1	0

The eddy diffusivity for heat is defined as follows:

$$\alpha t = C\lambda\, f\lambda\, k\tau m = C\lambda\, f\lambda\, k\sqrt{\frac{k}{\varepsilon}}\sqrt{\frac{t2}{\varepsilon t}} = C\lambda\, f\lambda \frac{k2}{\varepsilon}\sqrt{2R}$$

Where $R = \tau_t/\tau_u = (\overline{t^2}/\varepsilon_t)(k/\varepsilon)$ is the time-scale ratio, and $\tau_m = \sqrt{k/\varepsilon}\sqrt{\overline{t^2}/\varepsilon_t}$ is the mixed time scale.

The extent of the computational domain was decided based on temperature variations and the required area of analysis. The computational domain extended 4m on the sides, 10m on the front, 3m at the bottom, and 4.5m at the top. Natural outflow conditions were assumed at the five boundaries of the computational domain i.e. top boundary, bottom boundary, X_{min}, Y_{min}, and Y_{max}. On the X_{max} side, that was North of the building according to site data was given fixed velocity condition. The average value of velocity was 2.2m/s from north to south. The ambient temperature of the computational domain was set at 28.7°C based on measurement data on site. Gravitational acceleration of 9.8m/s^2 was considered. The steady-state analysis was done for the simulation model. Pseudo-time step relaxation was considered for the heat balance equation. Under relaxation was used for advection and diffusion term. Convergence criteria was 10^{-4} for temperature, velocity, turbulent kinetic energy, and turbulent dissipation rate. Mesh independent grid test was done to make the model independent of size of the mesh. The optimum mesh size had 10,38,588 elements. At the building geometry, the element size was 100mm X 100mm.

Staking of ODU in different geometries of recessed space

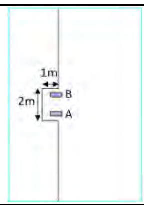
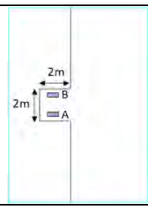
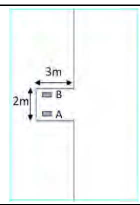
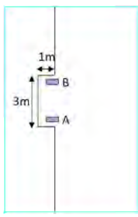
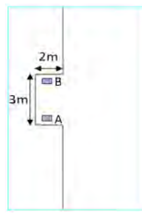
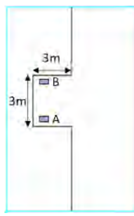
Six different stacking geometries were studied to understand the impact of geometry of the recessed space on performance of AC.

Table 6 represents the depth and width of the recessed space in which the ODUs are placed. 11 storey building was considered. The total height of the building was 33 meters. The ODUs were placed in two columns: Column A and Column B.

Table 5: Placement of ODUs in Recessed Space

Column A	The side inlet area of the ODUs in column A faced the open end of the recessed space
Column B	The side inlet area of the ODUs in column B faced the closed end of the recessed space.

Table 6: Geometries of recessed space

			
Case	1 (W2xD1)	2 (W2xD2)	3 (W2xD3)
Depth	1m	2m	3m
Width	2m	2m	2m
			
Case	4 (W3xD1)	5 (W3xD1)	6 (W3xD1)
Depth	1m	2m	3m
Width	3m	3m	3m

The boundaries of the computational domain were set to have natural outflow conditions. As the main area of analysis was recessed space, the domain boundaries coincided with the building geometry on the sides and back. The extent of computational domain was 5 meters at the front and, 3 meters at the top of the recessed space. No-wind condition was assumed. COP was calculated using the given formula (T T Chow et al., 2006).

$$COP = 4.825 - 0.0687t_o$$

Results

Figure 3 represents the percentage difference of each turbulence model for outlet air temperature, condenser on-coil temperature from the rear, and side of ODU when compared to site data. RMSE (Root Meat Square Error) method was further used to find the mean difference of six floors. NK model shows 8.89% and 4.87% and 3.53% difference for outlet, inlet temperature at back and inlet temperature at side of ODU, respectively.

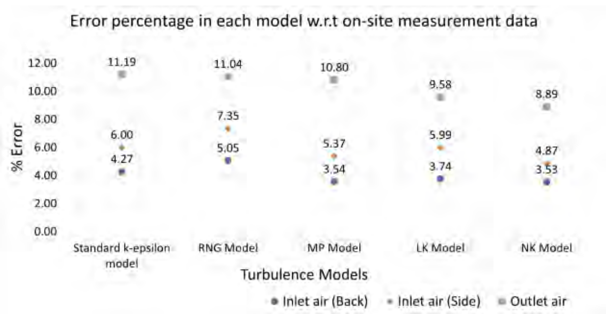


Figure 3: Percentage difference in each model compared to site data

CFD simulations also does not show the effect of stack. Wind velocity is a major reason that stack effect cannot be seen. The magnitude of velocity ranged from 1.5m/s to 3m/s near the ODUs. Thus, it drove away the heat dissipated from the compressors. The placement of ODUs at an open façade caused faster diffusion of the heat rejected from the ODUs in the atmosphere. As a result, the ODUs located at the top floors are not affected by the heat rejection from the ODUs at the lower floors. Similar temperature profile was observed at site.

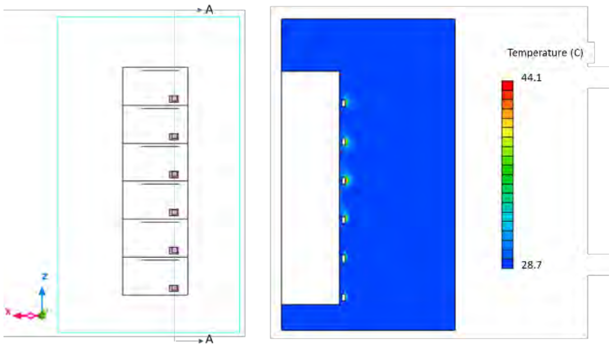


Figure 4: Air temperature variation at section A-A obtained from simulation results

Table 7: CFD simulation results at ODU of Air Conditioner

Floor	Outlet Air Temperature (°C)	Inlet Air Temperature (Back) (°C)	Inlet Air Temperature (Side) (°C)
1	38.73	28.7	28.70
2	41	29.75	28.77
3	41.94	29.8	29.19
4	44.06	30.52	31.97
5	41.45	29.49	31.02
6	39.7	29.22	30.51

Comparison of condenser on-coil temperature at ODUs placed in different geometries of recessed spaces

Condenser on-coil temperatures at back of ODUs placed in Column A increases to 38°C and 34°C for 2m and 3m wide recessed spaces, respectively. The condenser on-coil temperatures at the back of ODUs at Column A does not vary more than 1.5°C.

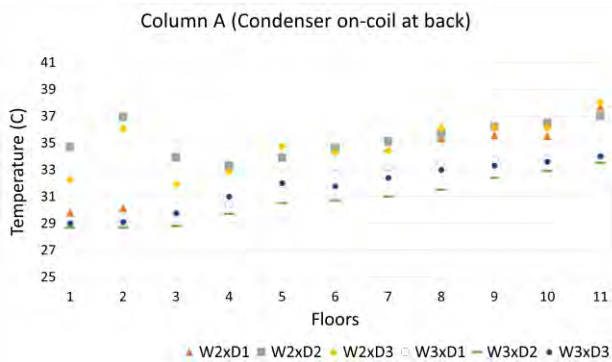


Figure 5: Condenser on-coil temperature at back of ODU placed at Column A for different recessed spaces

For the lower depth of the AC, condenser on-coil temperatures at the sides was minimum. For 2m wide recessed space, the maximum condenser on-coil temperatures were 33.5°C, 35°C and 36.5°C for 1m, 2m and 3m depth, respectively. When the recessed space having width of 3m was considered, the condenser on-coil temperatures further lowered to 31.4°C, 31.6°C and 32.1°C. Thus, for 3m wide space when different depths were considered, the condenser on-coil temperatures did not vary more than 1°C.

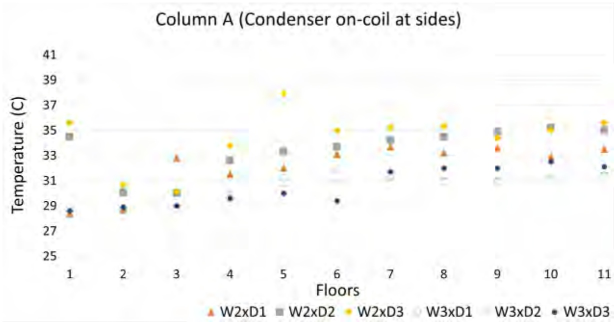


Figure 6: Condenser on-coil temperature at side of ODUs placed at Column A for different recessed space

Figure 7 shows that the condenser on-coil temperatures increase with increasing floor height. At 11th floor, these temperatures increase to 39.5°C and 36°C for 2m and 3m wide recessed space, respectively. The impact of depth of recessed space on condenser on-coil temperatures is seen up to fifth floor. Beyond 5th floor the variation in temperature is mainly caused due to the width of the recessed space rather than depth. From 6th floor to 11th floor, the impact of depth of condenser on coil temperatures at the back of the ODUs at Column B is less 1.5°C and 1°C and for 2m and 3m wide recessed spaces, respectively.

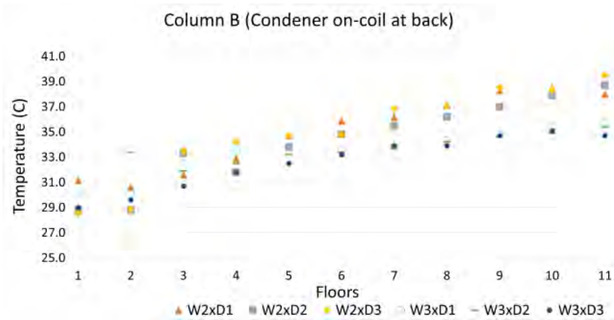


Figure 7: Condenser on-coil temperature at back of ODUs placed at Column B for different recessed spaces

As the side inlet area for Column B lies towards the closed end of the recessed space, the depth does not impact the condenser on-coil temperatures by more than 1°C after 5th floor. Though the width of the recessed space, when increased from 2m to 3m brings down the maximum on-coil temperatures by 3°C. For 2m wide space, the temperature on sides of ODU reaches up to 40°C when the depth is 1m and 3m.

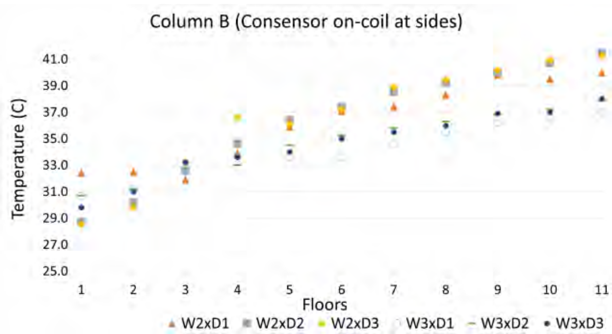


Figure 8: Condenser on-coil temperature at side of ODUs placed at Column B for different recessed space

Figure 9 and Figure 10 shows the temperature profile different cases of recessed space. For no wind conditions, buoyancy drives the motion of fluid. The condenser on-coil temperatures rise by 10°C at last floor for 2m wide recessed space. For 3m wide space, the maximum temperature rises by 5°C at the last floor in the recessed space.

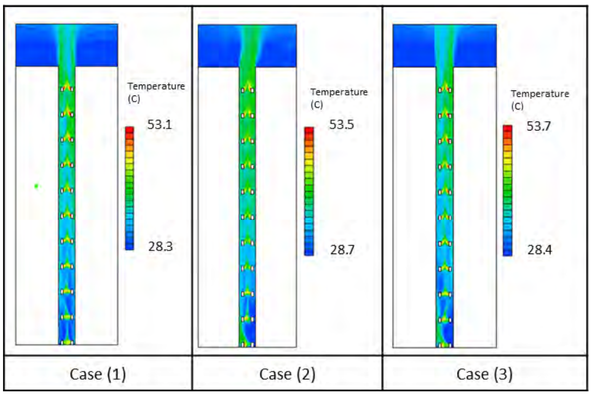


Figure 9: Temperature variation in 2m wide recessed space

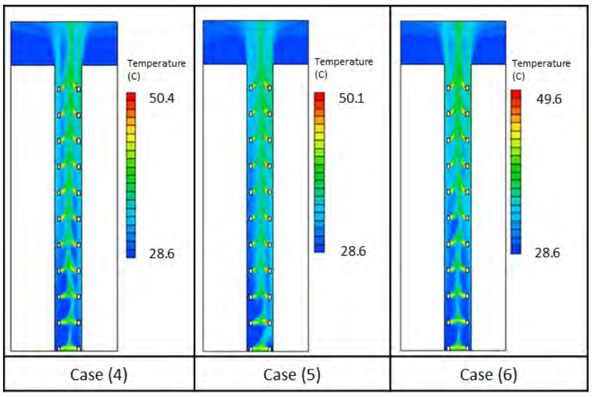


Figure 10: Temperature variation in 3m wide recessed space

Impact on Energy Efficiency of AC

The value of reference COP is taken at ambient temperature of 28.7°C. The calculation for percentage drop of COP is with respect to reference COP value. The maximum drop in COP of AC units was 27% and 22% in Column A and Column B, respectively can be seen in the air conditioner units placed in Case 3. The recessed space having width of 2m shows average 21% and 25% drop in COP of AC at higher floors placed in Column B and Column A respectively. The average drop in the COP of AC placed in 3m wide recessed space was 11% for ODUs placed in Column A and 17% for ODUs placed in Column B. Minimum drop is observed in the ODU of Column A of 3m wide and 1m deep recessed space.

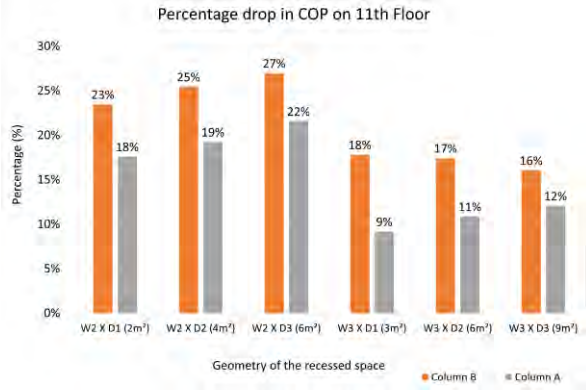


Figure 11: Maximum percentage drop in COP of Air Conditioner units

Discussion

The NK two-equation heat transfer model proves to be the most accurate model when compared to on-site measurement data with minimum difference of 4.58% for condenser on-coil temperatures and 8.89% for outlet air temperature from the ODU.

Wider recessed space is beneficial for reducing the increment in condenser on-coil temperatures. For 2m wide recessed space, the temperature at the 11th floor increased to almost 39.8°C, whereas the condenser on-coil temperatures of ODUs placed in 3m wide recessed space increased to 33.7°C. The condenser on coil temperature at the side inlet area rises as high as 41.3°C for the ODUs placed at Column B. This is because the inlet area at the side of the compressor unit faces the closed end of the recessed space. The heat accumulation is more near the closed space. The condenser on coil temperature at the side inlet area of ODU units placed in Column A rise to 35.5°C. The temperature increment of almost 5-6°C can be avoided by placing the compressor units such that the side inlet area faces the open end of recessed space.

According to the analysis done for the recessed space, the geometry that has 6m2 area of the recessed space which is 2m wide and 3m deep has the most inefficiently working AC. For the same area of recessed which has 3m width and 2m depth can prove to be 10-15% more efficient. When the recessed space with 3m2 area was studied, an average drop of 13.5% was noted in the COP of AC compared to a 25% drop observed for 2m wide recess space.

As the simulations were done for no-wind conditions, the buoyancy was the only factor that caused the motion of fluid. As a result, the depth did not impact the condenser on-coil temperature at the back of ODU but increasing the depth of recessed space did show increment in the condenser on-coil temperatures at the side of the ODU units for Column A. As the depth of the space was increased, the heat accumulated more at the side inlet area of ODUs in Column A. Average increment of 2°C was observed in condenser on-coil temperatures at the side inlet area of ODUs placed in Column A when the depth was increased from 1m to 3m.

Conclusion

In major Indian cities, urbanisation and increasing population contribute to increasing vertical development. As a result, the cities have many high-rise residential apartments where the outdoor units of Air Conditioners are densely stacked.

This project thus focuses on first comparing the simulation results with on-site measurements and further simulating placements of outdoor units in recessed spaces having varying depth and width. The simulation study was carried out using scSTREAM simulation software.

It was observed that in case 3, the recessed space is the densest, which has 2m width and 3m depth and has 6m² area. The COP drops by 27% for Column B and 22% for Column A. For the recessed space with lesser area in Case lV, where the area is just 3m², minimum drop in COP can be seen. With more width and less depth, only a 9% drop can be seen in the COP of air conditioner at higher floors. Thus, the broader area of the recessed space should be preferred. The width if kept minimum will help in easy dissipation of hot air into the atmosphere rather than accumulating in the space

The study will be beneficial during the pre-design stage to decide the optimum dimensions or recommended ventilation of the recessed space. The study will further help during the installation of ODU to determine the optimum placement of air conditioning units. The building owners, HVAC engineers and installers can decide the optimum placement of ODU units using a defined simulation method. The study will benefit during the pre-design stage to decide the optimum dimensions of the recessed space along with placement of air conditioning units.

Acknowledgment

The authors would like to express gratitude towards CARBSE (Centre for Advanced Research in Building Science and Energy) for being very cooperative and providing the measurement equipment required on-site.

Sincere thanks to Mr. Manoj Patel (Owner of the hotel) for their coperation during on-site building measurement.

References

Bojic, M., Lee, M., & Yik, F. (2002). Influence of a depth of a recessed space to flow due to air-conditioner heat rejection. *Energy and Buildings*, *34*(1), 33–43. https://doi.org/10.1016/S0378-7788(01)00078-0

Bruelisauer, M., Meggers, F., Saber, E., Li, C., & Leibundgut, H. (2014). Stuck in a stack - Temperature measurements of the microclimate around split type condensing units in a high rise building in Singapore. *Energy and Buildings*, *71*(March), 28–37. https://doi.org/10.1016/j.enbuild.2013.11.056

Campbell, I., Kalanki, A., & Sachar, S. (2018). *Solving the Global Cooling Challenge: How to Counter the Climate Threat from Room Air Conditioners*. 50.

Choi, S. H., Lee, K. S., & Kim, B. S. (2005). Effects of stacked condensers in a high-rise apartment building. *Energy*, *30*(7), 968–981. https://doi.org/10.1016/j.energy.2004.08.004

Chow, T. T., & Lin, Z. (1999). Prediction of on-coil temperature of condensers installed at tall building re-entrant. *Applied Thermal Engineering*, *19*(2), 117–132. https://doi.org/10.1016/S1359-4311(98)00042-8

Chow, T. T., Lin, Z., & Wang, Q. W. (2000). Effect of building re-entrant shape on performance of air-cooled condensing units. *Energy and Buildings*, *32*(2), 143–152. https://doi.org/10.1016/S0378-7788(99)00048-1

Chow, T. T., Lin, Z., & Yang, X. Y. (2002). Placement of condensing units of split-type air-conditioners at low-rise residences. *Applied Thermal Engineering*, *22*(13), 1431–1444. https://doi.org/10.1016/S1359-4311(02)00068-6

Chow, T T, Lin, Z., Wang, Q. W., Lu, J. W. Z., Avenue, T. C., Tong, K., & Kong, H. (2006). Split-Type Air-Conditioners At Building Re-Entrant Via Computer Simulation. *City University of Hong Kong ENERGY PERFORMNANCE OF SPLIT- CFD SIMULATION WITH k- ε MODEL*, *23*(1), 1–7.

Chow, Tin Tai, Lin, Z., & Wang, Q. W. (2000). Applying CFD Simulation in Analysing Split-type Air-conditioner Performance at Buildings. *Architectural Science Review*, *43*(3), 133–140. https://doi.org/10.1080/00038628.2000.9696896

Duan, R., Wang, X., Song, Y., & Liu, J. (2016). Influence of Air-conditioning Outdoor Unit Arrangement Strategy on Energy Consumption. *Procedia Engineering*, *146*, 350–358. https://doi.org/10.1016/j.proeng.2016.06.409

Phadke, A., Abhyankar, N., & Shah, N. (2014). Avoiding 100 New Power Plants by Increasing Efficiency of Room Air Conditioners in India : Opportunities and Challenges. *Eedal '13*, *June*, 1–14.