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Modeling impacts of ventilation and filtration methods on energy use and airborne disease transmission in classrooms

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A R T I C L E   I N F O

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A B S T R A C T

Lowering the potential of airborne disease transmission in school buildings is especially important in the wake of the COVID-19 pandemic. The benefits of increased ventilation and filtration for reducing disease transmission compared to drawbacks of reduced thermal comfort and increased energy consumption and electricity demand are not well described. A comprehensive simulation of outdoor air ventilation rates and filtration methods was performed with a modified Wells-Riley equation and EnergyPlus building simulation to understand the trade-offs between infection probability and energy consumption for a simulated classroom in 13 cities across the US. A packaged heating, ventilation, and air conditioning unit was configured, sized, and simulated for each city to understand the impact of five ventilation flow rates and three filtration systems. Higher ventilation rates increased energy consumption and resulted in a high number of unmet heating and cooling hours in most cities (excluding Los Angeles and San Francisco). On average, across the 13 cities simulated, annual energy consumed by an improved filtration system was 31% lower than the energy consumed by 100% outdoor air ventilation. In addition, the infection probability was 29% lower with improved filtration. An economizer, which activates cooling based on an outdoor temperature setpoint, increased ventilation and reduced both energy consumption and infection probability. It was also concluded that ventilation and filtration measures better reduced absolute infection probability when the quanta generation rate for an infectious disease was higher. Dynamic outdoor airflow rate controls and filtration technologies that consider both health and energy consumption are an important area for further research.

Abbreviations

ASHRAE American Society of Heating, Refrigerating and Air-Conditioning Engineers
CADR Clean air delivery rate
CDPH California Department of Public Health
CO₂ Carbon dioxide
COP Coefficient of performance
DOE Department of Energy
HEPA High efficiency particulate air
IECC International Energy Conservation Code

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1. Introduction

Management of the indoor environment affects transmission rates of infectious diseases spread by transfer of airborne respiratory aerosols that contain pathogens such as coronavirus (including SARS-CoV-2), influenza, rhinovirus, and tuberculosis. Mounting evidence published in the summer of 2020 [1] showed that the SARS-CoV-2 virus was primarily spread through respiratory aerosols generated by people indoors when they breathe, talk, cough, or sneeze and changed the way people use indoor spaces. Documented outbreaks of COVID-19, the disease that results from infection with SARS-CoV-2, have originated from sharing indoor spaces with an infected person [2,3]. Thus, there has been an increased focus on the potential transmission of respiratory diseases in indoor spaces by institutions, facility operators, researchers, workers, and the public.

The main role of a heating, ventilation, and air conditioning (HVAC) system is to maintain a comfortable indoor environment. Such systems have historically focused on temperature and humidity control with generally little emphasis on air quality concerns. However, the HVAC system outdoor air ventilation rate (to replace and exhaust indoor air) plays a key role in indoor air quality and poorly ventilated buildings may contribute to transmission of airborne diseases [4,5]. In addition, increased outdoor airflow rates improve work efficiency and reduce absenteeism in offices [6,7]. Higher outdoor air ventilation rates can also improve student performance [8,9] and reduce absence rates in schools [9]. Even with the widespread recognition of benefits of ventilation, classrooms are frequently under-ventilated relative to building standards [9,10]. Thus, there have been guidelines issued by institutions like American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) in response to the COVID-19 pandemic to increase outdoor airflow rates to as high as 100% of supply air (or to as much as the HVAC system can accommodate) [11]. However, the impacts of increased outdoor air on reducing the probability of transmitting an infection and the trade-off in indoor comfort and HVAC energy consumption remain unclear.

Outdoor air needs to be filtered and conditioned, which generally leads to increased HVAC system energy consumption. Ascione et al. [12] combined energy audit data, calibrated energy models, and computational fluid dynamics simulation of a university classroom in a cold climate in Italy to compute energy consumption of several mechanical systems designed to reduce disease transmission during the COVID-19 pandemic. The study showed that varying outdoor air ventilation from 7 l/(s.person) to 21 l/(s. person) tripled HVAC energy consumption and that the same increase in outdoor air combined with sensible heat recovery resulted in a more modest 40% increase. Ascione et al. also reviewed past studies recommending the use of in-room filtration (IRF), especially with high efficiency particulate air (HEPA) filters, because of their high filtration efficiency. However, the study was very specific to the location, conditioned area, and number of occupants. Dutton et al. determined that increasing outdoor airflow rates in California offices would reduce workplace exposure to formaldehyde and that energy impacts vary by economizer use and climate, where the coldest California climates had the largest energy penalties due to heating requirements [13]. Economizers increase outdoor air from the minimum ventilation rate to up to 100% outdoor air for “free cooling” when outdoor air conditions (temperature or enthalpy) are favorable from an energy perspective. Although there is a general agreement that increasing outdoor airflow rates has many benefits, including reducing the probability of infectious disease transmission in buildings, the energy impacts of conditioning outdoor air vary by location and time of year.

Azimi and Stephens [14] conducted a study in 2013 using a modified version of Wells-Riley model for predicting probability of infectious disease transmission. They considered HVAC filtration and outdoor air ventilation to estimate risk reductions and associated operational costs for an office building. The study concluded that HVAC system filtration, especially minimum efficiency reporting value (MERV) 13–16 filters, would achieve risk reductions from infectious diseases at lower operational costs than equivalent levels of outdoor air ventilation, since removing respiratory aerosols by filtration is generally more energy efficient than replacing indoor air with conditioned outdoor air. Azimi and Stephens [14] was limited to office buildings and did not consider hourly changing climates and varying efficiency of the heating or cooling equipment as a function of outdoor air temperature. Risbeck et al. [15] investigated the tradeoff between lowering the probability of airborne disease transmission and energy consumption of the HVAC system for classrooms, gyms, cafeteria and other miscellaneous buildings. The study modeled a hypothetical concentration of infectious particles in the air using the Wells-Riley model and determined the expected infection rate based on occupancy level, occupant behavior and HVAC system operation. The study used a simplified model for HVAC energy use and was limited to analyzing high outdoor airflow

### Nomenclature

- **ΔFan**: Additional power consumption for MERV 13 filtration [W]
- **ΔP**\(_{\text{filter}}\)**: Increase in the filter pressure drop for MERV 13 filtration [Pa]
- **Q**\(_{\text{fan}}\)**: System airflow rate [m\(^3\)/s]
- **n**\(_{\text{fan}}\)**: Fan system efficiency for converting electric power to moving air
- **P**\(_{26}\)**: Probability that an infected person transmits infection to at least one of 26 susceptible people
- **P**\(_{1}\)**: Probability that an infected person transmits infection to one susceptible person

### In room filtration

**HVAC**: Heating, ventilation and air conditioning

**MERV**: Minimum efficiency reporting value

**SHGC**: Solar heat gain coefficient

**VOC**: Volatile organic compounds

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rates (above 80% of supply airflow). The Risbeck et al. study illustrated an important concept, namely that hourly weather conditions will play an important role in determining the optimal HVAC strategy. Buildings in milder climates may be able to rely on increased ventilation to reduce infection probability, whereas buildings in extreme climates should favor improved HVAC system filtration instead. Yet, no comprehensive assessment of outdoor air ventilation rates and filtration methods has been carried out to understand how trade-offs will impact infectious disease transmission probability and energy consumption by location.

Thus, the current work builds on previous studies by simulating hourly weather data in 13 cities that represent most U.S. climate zones. The current study models five ventilation system configurations (four constant outdoor air ventilation rates as well as one economizer configuration) and three filtration levels (central MERV 8 and MERV 13 and in-room filtration). While many different types of HVAC equipment are used in U.S. commercial buildings, 50% of total floor area is heated by packaged units and 58% of total floor area is cooled by packaged units [16]. A packaged unit contains all HVAC components and is built off-site and is then delivered and connected to the building’s air duct system. For this study, a packaged unit representative of a commercially available system was configured, sized, and modeled for each city to understand the effect different ventilation and filtration strategies have on energy consumption, peak demand, indoor comfort, and infectious disease transmission probability. Analyzing the details of a realistic HVAC system and providing quantitative results is key to providing stakeholders with the information needed to recommend actions that will lower HVAC system energy consumption (lowering costs and greenhouse gas emissions) while providing a reasonable combination of outdoor air ventilation and filtration to maintain occupant wellness. The current study focuses on an HVAC system modeled in a classroom, which is expected to be representative for any building type with a similar occupant density (e.g. conference room). Results may differ for other building types, however the modeling methodology described in this study is applicable to other building types. Classrooms are modeled in the current study owing to the importance of prioritizing the education system in the pandemic recovery.

The current work uses an EnergyPlus model of a representative classroom across the U.S. and analyzes energy impacts of reducing HVAC system energy consumption, peak demand, indoor comfort, and infectious disease transmission probability. The analysis of Risbeck et al. [15] illustrated that the optimum solution for HVAC operation differs significantly building to building based on the equipment and weather conditions. Our study addresses this variability by considering various HVAC equipment sizes, ventilation rates and methods, filtration methods, and weather conditions.

The remainder of the paper is organized as follows: In Section 2, methods to assess infection transmission probability and energy consumption are discussed. In Section 3, the models from the previous section are simulated to view the results and examine key trends. In Section 4, study limitations and potential future work are discussed. Finally, study results are concluded in Section 5.

2. Methods

2.1. Geographic locations and weather

Simulations were conducted for weather representative of 13 large cities across the continental U.S. (Table 1). The cities were selected based on building models available from the Department of Energy’s (DOE) Commercial Reference Buildings [17], which represent a range of climate zones and moisture regimes, as defined by the International Energy Conservation Code (IECC) [18].

Typical meteorological year weather files (TMY3) for the selected cities were obtained from White Box Technologies [19]. While the design day conditions from the weather files were not used for HVAC system sizing, they are provided as a reference for minimum and maximum ambient conditions for the cities simulated (Table 1). The building model for Boulder, CO was simulated with weather from Denver, CO since a weather file for Boulder was not available. While Los Angeles is within climate zone 3B, which is a hot-dry climate, it is on the border of climate zones 3A and 3C.

2.2. Building model

An EnergyPlus version 9.4 [20] model was created to represent an individual classroom in a row of classrooms with 89.23 m² of floor area and a volume of 325 m³. The north and south walls were modeled as exterior walls and had one window each (representative

| IECC Climate Zone | IECC Moisture Regime | State | City     | Heating Design Day Mean (99th Percentile) Temperature (°C) | Cooling Design Day Mean (1st Percentile) Temperature (°C) | Mean Coincident Wet Bulb Temperature for Cooling Design Day (°C) |
|-------------------|---------------------|-------|----------|------------------------------------------------------------|----------------------------------------------------------|---------------------------------------------------------------|
| 1                 | A                   | FL    | Miami    | 11.1                                                       | 32.7                                                      | 25.3                                                          |
| 2                 | A                   | TX    | Houston  | 2.2                                                        | 34.7                                                      | 25.7                                                          |
| 2                 | B                   | AZ    | Phoenix  | 5.3                                                        | 42.4                                                      | 20.8                                                          |
| 3                 | A                   | GA    | Atlanta  | −3.1                                                       | 33.1                                                      | 23.3                                                          |
| 3                 | B-AC                | CA    | Los Angeles | 8                                                        | 26.9                                                      | 17.5                                                          |
| 3                 | C                   | CA    | San Francisco | 5.2                                                       | 25.6                                                      | 16.6                                                          |
| 4                 | B                   | NV    | Las Vegas | 1                                                         | 41.3                                                      | 19.5                                                          |
| 4                 | A                   | MD    | Baltimore | −7.8                                                      | 32.9                                                      | 23.4                                                          |
| 4                 | B                   | NM    | Albuquerque | −5.8                                                      | 33.9                                                      | 15.4                                                          |
| 5                 | C                   | WA    | Seattle  | −1.3                                                       | 27.6                                                      | 17.6                                                          |
| 5                 | A                   | IL    | Chicago  | −15.7                                                     | 31.5                                                      | 22.9                                                          |
| 5                 | B                   | CO    | Denver   | −14.1                                                     | 33.2                                                      | 15.5                                                          |
| 6                 | A                   | MN    | Minneapolis | −21.2                                                    | 31.1                                                      | 22.2                                                          |
of a typical wall/window ratio) with dimensions shown in Fig. 1 and properties shown in Table 2. The east and west walls were modeled as adiabatic, representing a shared wall with an adjacent classroom. The foundation was a 0.10 m thick concrete slab and the EnergyPlus object (Construction:FactorGroundFloor) was used to simulate the temperature of the perimeter of the concrete slab. This object uses ground temperatures from the weather file and the properties of the slab to model the heat exchange from ground to slab to the conditioned space.

The simulated classroom had 27 occupants, which is a typical occupancy for a classroom in the U.S. Each occupant was simulated with 120 W of internal gain, with the sensible to total heat ratio determined by an Energy Plus function that considered metabolic rate and room temperature. The design of the building envelope (Table 2) varied by city simulated and was based on properties for a city-specific primary school described in DOE’s existing commercial reference buildings for the building vintage “constructed in or after 1980” [17]. Values for infiltration (1.13 l/s per m² of exterior wall surface), lighting internal gain (21.53 W/m² of floor area), and plug load internal gain (15 W/m²) were the same for each city and sourced from the DOE reference [17]. The infiltration simulated in addition to the mechanical ventilation equates to 3.1 l/(s.person), which is an estimate for classrooms with occasional opening of doors and windows. This is inside the bounds of air exchange rates in classrooms with closed windows [21] and air exchange rates in classrooms reliant on active natural ventilation from doors and windows [22].

The daily schedule (Table 3) was designed to operate the HVAC system 1 h prior to occupancy and to include 5.25 h of instructional time per day. The school year was simulated to start on the third Monday in August and last for 36 weeks (for a total of 945 instructional hours annually), with the following school breaks: one week starting the third Monday in November, three weeks starting the third Monday in December, and one week starting the first Monday in April. Although the specifics on the building model, operation, and number of occupants are important, the focus of the study is on the comparison of ventilation rates and filtration methods in different climates across the US.

2.3. HVAC system model

The HVAC system model represents a packaged system that includes a single-speed compressor heat pump, electric resistance back-up heat, and a blow-through single-speed fan. Fig. 2 shows the components of the wall-mounted HVAC system in a simplified schematic. The system includes: a mixing box where part of the return air is exhausted and the remainder mixes with outdoor air, an air filter, a single-speed blow-through fan, the indoor coil, and the outdoor coil.

In EnergyPlus, the AirLoopHVAC:Unitary System object was used to represent the HVAC system. Normalized heat pump performance curves [23] were applied and scaled to the cooling and heating rated capacities and efficiencies from Bard manufacturer data for the WH Series (5.3–17.6 kW) (Table 4) [24,25], which is a wall-mount packaged HVAC system commonly used in schools. Fan power intensity data, which was not available in the manufacturer data, was collected from a field installation of a similar system [26]. The average value measured in the field study (0.92 W/(l.s)) was used as the basis for the fan power calculations. For each simulation, the unitary system was controlled by a single dual-temperature thermostat. The setpoint between hours 7:30–16:00 on school days was 24.44 °C for cooling and 18.89 °C for heating, which was consistent with typical thermostat settings observed in a field study of 104 classrooms [26]. The HVAC system was shut off outside of these hours.

Heat pump capacity and electric resistance heat sizing for each geographic location (Table 5) was determined by running the annual simulation with a fixed ventilation rate of 7 l/(s.person) while varying the heat pump size from 5.1 to 16.0 kW of rated cooling capacity and the electric resistance back-up heat from 4 kW to 20 kW (operated when the heat pump could not meet the load or when the outdoor air temperature dropped below −20 °C). The size for each location was then determined by selecting the smallest heat pump equipment that resulted in annual unmet cooling hours less than 3% of the total occupied hours (Table 5). Then, the smallest electric resistance back-up heat was selected that similarly resulted in less than 3% unmet heating hours. The most common electric resistance heat element size was 4 kW and the coldest climates, particularly those with the lowest cooling loads and medium capacity heat pumps, required up to 20 kW (Table 5).

The outdoor airflow rate during occupied hours was modeled to be one of four fixed airflow rates (3.5, 7.0, 10.5, or maximum l/(s.person)) or a minimum airflow of 7 l/(s.person) with an economizer increasing outdoor air (up to 100%) when cooling was required and the outdoor air temperature was below the economizer high limit temperature, which varied by city (Table 5). For comparison, the infiltration rate described in the building model equates to a natural ventilation rate of 3.1 l/(s.person) in addition to the mechanical ventilation provided. The economizer logic was based on optimization strategies recommended by Taylor and Cheng [27]. The design
Table 2
Envelope and internal loads for classroom model.

| Location     | IECC Climate Zone | Wall Insulation R Value [m²K/W] | Roof Insulation R Value [m²K/W] | Window Thermal Transmittance U-Value [W/(m²K)] | Window Solar Heat Gain Coefficient (SHGC) |
|--------------|-------------------|---------------------------------|---------------------------------|------------------------------------------------|------------------------------------------|
| Miami        | 1A                | 0.32                            | 2.38                            | 6.93                                           | 0.25                                     |
| Houston      | 2A                | 1.17                            | 2.67                            | 6.93                                           | 0.25                                     |
| Phoenix      | 2B                | 0.73                            | 3.83                            | 6.93                                           | 0.25                                     |
| Atlanta      | 3A                | 1.36                            | 2.44                            | 4.09                                           | 0.25                                     |
| Las Vegas    | 3B                | 1.10                            | 3.66                            | 6.93                                           | 0.25                                     |
| Los Angeles  | 3B-AC             | 0.80                            | 1.76                            | 6.93                                           | 0.44                                     |
| San Francisco| 3C                | 1.36                            | 2.00                            | 4.09                                           | 0.39                                     |
| Baltimore    | 4A                | 1.98                            | 3.03                            | 3.35                                           | 0.36                                     |
| Albuquerque  | 4B                | 1.76                            | 2.99                            | 4.09                                           | 0.36                                     |
| Seattle      | 4C                | 1.92                            | 2.75                            | 4.09                                           | 0.39                                     |
| Chicago      | 5A                | 2.15                            | 3.38                            | 3.35                                           | 0.39                                     |
| Denver       | 5B                | 2.15                            | 3.51                            | 3.35                                           | 0.39                                     |
| Minneapolis  | 6A                | 2.71                            | 3.97                            | 2.95                                           | 0.39                                     |
standard of 7 l/(s.person) is based on California’s Title 24 [28], which generally requires a slightly higher outdoor airflow rate in classrooms than ANSI/ASHRAE Standard 62.1–2019 [29], which bases the outdoor rate requirement on the sum of two factors (maximum design occupancy and floor area). For reference, the ANSI/ASHRAE 62.1–2019 ventilation rate for the modeled classroom is 6.7 l/(s.person). In addition to the design standard, ventilation rates 50% greater (“high”, 10.5 l/(s.person)) and 50% lower (“low”, 3.5 l/(s.person)) were modeled. Finally, the case of maximum 100% outdoor air was modeled. The rate for 100% outdoor air was

### Table 3

Daily classroom schedule.

| Classroom Schedule  | Time       | Occupancy |
|---------------------|------------|-----------|
| Before school       | 07:30–08:30| 0         |
| Class               | 08:30–11:30| 27        |
| Lunch               | 11:30–12:30| 1         |
| Class               | 12:30–14:45| 27        |
| After School        | 14:45–16:00| 1         |

### Table 4

Rated performance values for Bard WH-B and WH-C systems [24,25]. Fan power data estimated from field measurements of fan power intensity [26]. Supply airflow rates per person assume 27 occupants.

|                      | Units | W18HB | W24HB | W36HB | W48HC | W60HC |
|----------------------|-------|-------|-------|-------|-------|-------|
| Cooling Capacity     | kW    | 5.1   | 6.9   | 10.6  | 13.9  | 16.0  |
| Cooling COP          | –     | 3.31  | 3.31  | 3.25  | 3.22  | 3.22  |
| Heating Capacity     | kW    | 4.9   | 6.6   | 9.3   | 12.1  | 15.0  |
| Heating COP          | –     | 3.5   | 3.5   | 3.3   | 3.24  | 3.25  |
| Supply Airflow       | l/s   | 283.2 | 377.6 | 542.7 | 731.5 | 825.9 |
| Supply Airflow       | l/(s.person) | 10.5 | 14.0 | 20.1 | 27.1 | 30.6 |
| Fan Power            | W     | 261   | 347   | 499   | 673   | 760   |
| Auxiliary Heat       | kW    | 4.0–15.0 | 4.0–15.0 | 4.0–20.0 | 4.0–20.0 | 4.0–20.0 |
| Filter Face Area     | m²    | 0.26  | 0.26  | 0.31  | 0.52  | 0.52  |

**Fig. 2.** Schematic of HVAC system model. Economizer moves dampers to supply 100% outdoor air when the outdoor temperature is below the set point and cooling is required.
dependent upon the size of the modeled system, which was a function of geographic location, and its respective supply fan (Table 5). Note that the 100% outdoor airflow rate for systems modeled in San Francisco and Seattle was only 10.5 l/(s.person), and thus only three fixed outdoor airflow rates were modeled (3.5, 7.0, and 10.5 l/(s.person)). The return airflow rate was calculated using the difference between supply airflow rate and the outdoor airflow rate. The exhaust airflow rate was balanced to equal to the outdoor airflow rate.

2.4. Packaged HVAC filtration system model

As described in the HVAC system model section, an average fan power intensity value of 0.92 W/(l.s) was used as the basis for fan power calculations. This average power intensity measured in a field study [26] included the fan system operation with a pleated 5 cm deep MERV 8 filter. To estimate the energy impact of increasing to MERV 13 filtration, pressure drop versus filter face velocity data were obtained for MERV 8 and MERV 13 filters of 5 cm depth from Airguard [30, 31] and a least-squares regression was used to correlate filter face velocity to pressure drop using the power law (Fig. 3). The HVAC systems modeled have filter face velocities between 1.0 and 1.8 m/s because HVAC manufacturers use discrete component sizes (e.g. cabinets, motors, fans). Because the pressure difference between filters over this range is small, an average filter face velocity of 1.5 m/s was used. The additional power consumption due to the MERV 13 filter in each system was then calculated from Equation (1):

$$\Delta F_{\text{fan}} [W] = \frac{\Delta P_{\text{filter}} [Pa]}{n_{\text{fan}} [-]} \frac{Q [m^3/s]}{n_{\text{fan}} [-]}$$

(1)

where $\Delta P_{\text{filter}}$ [Pa] is the increase in the filter pressure drop for MERV 13 over MERV 8 filtration, $Q$ [m$^3$/s] is the system airflow rate, and $n_{\text{fan}}$ [-] is the fan system efficiency for converting electric power to moving air, which was taken to be 0.35 for a typical blower with electrically commutated motor [32]. When normalized by total airflow, the projected fan power intensity increase was 0.051 W/(l.s). A fan power intensity increase of 0.051 W/(l.s) for the MERV 13 filter is a 6% increase above the 0.92 W/(l.s) used for the fan system that included the MERV 8 filter. For comparison, Zaatari et al. reported an impact of 6–14% increase in fan power for replacing a MERV 8 filter with a MERV 13 filter in a system with a filter face velocity of 1.2 m/s [33]. It is important to note that filter construction and pressure drop can vary significantly among filters brands and models with the same MERV rating.

| IECC Climate Zone | City          | HVAC System Model | Heat Pump Capacity (Cool/Heat) (kW) | Electric Resistance Heat Capacity (kW) | Maximum Outdoor Airflow rate (l/s) Total (per person) | Economizer High-Limit Outdoor Air Temp (°F) |
|-------------------|---------------|-------------------|------------------------------------|----------------------------------------|--------------------------------------------------------|---------------------------------------------|
| 1A                | Miami         | W60HC             | 16.0/15.0                          | 4.0                                    | 825.9 (30.6)                                            | 20.56                                       |
| 2A                | Houston       | W60HC             | 16.0/15.0                          | 4.0                                    | 825.9 (30.6)                                            | 20.56                                       |
| 2B                | Phoenix       | W60HC             | 16.0/15.0                          | 4.0                                    | 825.9 (30.6)                                            | 22.78                                       |
| 3A                | Atlanta       | W48HC             | 13.9/12.1                          | 4.0                                    | 731.5 (27.1)                                            | 20.56                                       |
| 3B                | Las Vegas     | W60HC             | 16.0/15.0                          | 4.0                                    | 825.9 (30.6)                                            | 22.78                                       |
| 3B-AC             | Los Angeles   | W24HB             | 6.9/6.6                            | 4.0                                    | 377.6 (14.0)                                            | 22.78                                       |
| 3C                | San Francisco | W18HB             | 5.1/4.9                            | 4.0                                    | 283.2 (10.5)                                            | 22.78                                       |
| 4A                | Baltimore     | W36HB             | 10.6/9.3                           | 6.0                                    | 542.7 (20.1)                                            | 20.56                                       |
| 4B                | Albuquerque   | W36HB             | 10.6/9.3                           | 4.0                                    | 542.7 (20.1)                                            | 22.78                                       |
| 4C                | Seattle       | W18HB             | 5.1/4.9                            | 4.0                                    | 283.2 (10.5)                                            | 22.78                                       |
| 5A                | Chicago       | W24HB             | 6.9/6.6                            | 15.0                                   | 377.6 (14.0)                                            | 20.56                                       |
| 5B                | Denver        | W36HB             | 10.6/9.3                           | 5.0                                    | 542.7 (20.1)                                            | 22.78                                       |
| 6A                | Minneapolis   | W36HB             | 10.6/9.3                           | 20.0                                   | 542.7 (20.1)                                            | 21.67                                       |

Fig. 3. Filter resistance as a function of airflow for MERV8 versus MERV13 filter.
2.5. In room filtration model

The simulations were run both with and without IRF that had a clean air delivery rate (CADR) of 302 l/s for particle sizes between 0.09 and 11 μm. This is the rate recommended by the Association of Home Appliance Manufacturers (AHAM) for portable air cleaners for a room of the modeled dimensions [34], resulting in 3.3 filtration air changes per hour (ACH). The CADR metric is based on a laboratory test that measures reduction in particles of a particular type and size distribution in a test chamber when the air cleaner is turned on, as compared to natural decay when the air cleaner is turned off. The IRF simulated was assumed to meet the Energy Star Program Requirements for portable air cleaners [35], which requires that the CADR per Watt for smoke removal (particles 0.09–1.0 μm) meet or exceed 1.37 l/(s-W). Therefore, the in-room filtration delivering a CADR of 302 l/s was modeled with a power consumption of 221 W. The portable air cleaner was modeled to operate from 07:30–16:00 on school days.

2.6. Infectious disease transmission model

A probability of infection transmission calculator, published by the California Department of Public Health (CDPH) [36], was used to estimate the probability of one infected person infecting another susceptible person through long-range airborne transmission in an
enclosed indoor environment. The probability of infection calculated here accounts for room-scale transmission in a well-mixed room and does not account for infection through direct close contact or through potential surface contamination. The well-mixed room model also does not account for airflow patterns created by HVAC and in-room filtration equipment, which could increase transmission based on how infectious aerosols are dispersed [37]. The CDPH calculator applies a modified Wells-Riley equation [36], which is based on a concept of a “quantum of infection” determined for a specific pathogen from epidemiology studies. The modified Wells-Riley equation accounts for removal of airborne infectious particles through ventilation, filtration, and deposition. The CDPH calculator also allows for direct reduction of the quanta generation rate to account for wearing masks. The production rate of infectious particles varies with both the pulmonary ventilation rate of people and the quanta generation rate for the pathogen. The modified Wells-Riley

Fig. 5. Annual electricity consumption per classroom (a), unmet heating and cooling hours (b), and maximum 15-min peak demand (c) by location and ventilation rate of 3.5, 7, and 10.5 l/(s.person), economizer logic (E) (7.0–10.2 l/(s.person)) and 100% outdoor air (10.5–30.6 l/(s.person)).
equation provides an assessment of the impact on relative risk produced by various filtration and ventilation scenarios under the same infectious particle production rate.

The probability that one infected person transmits an infection via respiratory aerosols to one susceptible person was calculated using the CDPH calculator for the following conditions:

- Floor area of 89.2 m² and ceiling height 3.65 m.
- Room occupancy of 27 persons, with one person infected.
- Exposure time of 5.25 h.
- Five ventilation rates per location, which included:
  - 3.5, 7.0, 10.5 l/(s.person)
  - 100% outdoor air - 14.0, 20.1, 27.1, or 30.6 l/(s.person) (location dependent)
  - The average ventilation (7.2–10.2 l/(s.person)) achieved by the economizer for each location, as determined by the EnergyPlus simulation.
- Supply airflow of 377.6, 542.7, 731.5, or 825.9 l/s (location dependent, as determined by the EnergyPlus simulation).
- MERV 8 and MERV 13 central filtration, where filtration efficiency was modeled as 15, 28, and 70% for MERV 8 and 65, 90, and 90% for MERV 13 for particle diameter size bins of 0.3–1, 1–3, and 3–10 μm respectively. The filtration efficiencies for each MERV rating are included in the CDPH calculator and are based on the minimum composite average particle size removal efficiencies specified by ASHRAE Standard 52.2–2017 [38].
- Quanta generation rate of 5 h⁻¹, which is suggested as a reasonable estimate for the original SARS-CoV-2 virus [39] and may be higher for more transmissible variants [40]. The diameter of particles generated was distributed as 20, 30, and 50% among size bins 0.3–1, 1–3, and 3–10 μm respectively.
- Masks worn to reduce quanta generation rate by 75% (to 1.25 h⁻¹), simulating that both the infected and susceptible people are wearing masks with 50% filtration efficiency across all size bins.
- Other defaults, as specified by CDPH [36].

The EnergyPlus simulation was run first and the parameters and outputs from the simulation (described above) were input into the CDPH calculator using a Python script that automated the data input and tabulated the probability of infection results. The output from the CDPH calculator is the probability that one infected person infects one other susceptible person (P₁). The probability that the infected person transmits an infection to at least one of the 26 susceptible room occupants (P₂₆) was calculated assuming that each potential transmission is an independent event. The probability that at least one transmission occurs is calculated using compound probability of the chance that no transmission will occur 26 times (Equation (2)):

\[
P_{26} = 1 - \left(1 - P_1\right)^{26}
\]

3. Results

The annual electricity consumption per classroom, unmet heating and cooling hours, and maximum 15-min peak demand are presented by ventilation rate for a city representative of each IECC climate zone and moisture regime 1A to 3B (Fig. 4) and 4A–6A (Fig. 5). The results are also tabulated as a supplementary data table (Table S1). While the annual simulation results are presented, the research team reviewed the simulation results at 5-min resolution to validate that the HVAC model was operating as expected. In addition, the total annual HVAC electricity results with ventilation at 7 l/(s.person) (normalized by floor area) for each city were compared to data for average heat pump electricity consumption for educational institutions based on the census division containing that city, as reported by the Energy Information Administration (EIA) [41]. The agreement in total HVAC electricity use for each city was generally in good agreement (within 30%) with the census division data, with notable exceptions being San Francisco, Chicago, and Minneapolis, where the simulated electricity use was respectively 0.5, 1.4, and 1.9 times the average EIA reported HVAC electricity use for the census division. EIA HVAC data is averaged across the entire census divisions, which contain multiple states and climate zones, and is not intended to be representative of specific cities. Chicago and Minneapolis are expected to have higher HVAC loads than census division average and San Francisco lower HVAC loads than census division average. The comparison of the average simulation results to the EIA data averaged for all cities was within 2% agreement, evidence that the magnitude of HVAC electricity consumption predictions are in-line with real world observations. In interpreting the results, it is important to remember that the focus of the study is on the comparison of the impact of ventilation rates and filtration methods on the change in electricity consumption, and not on the absolute magnitude of electricity consumption.

The EnergyPlus results for annual energy consumption (Figs. 4a and 5a) were disaggregated by fan energy (including MERV 8 filter), compressor energy for cooling, compressor energy for heating, and energy for electric resistance heating. The additional energy required for MERV 13 filtration and in-room filtration were calculated separately as described in sections 2.4 and 2.5. The additional energy required for MERV 13 filtration (22–65 kWh) represents the 6% increase in fan energy for all flow rates across all locations. The total annual HVAC electricity consumption for a system operating at a standard ventilation rate of 7 l/(s.person) with MERV 8 filtration, defined as the “baseline”, ranged from 823 kWh in San Francisco (3C) to 6193 kWh in Minneapolis (6A). The replacement of the MERV 8 filter with MERV 13 filter equates to an increase of 0.7–2.7% in annual HVAC system electricity consumption over the baseline. The energy required for the IRF (338 kWh) was more significant and represented a 5.5–41% increase in annual HVAC electricity consumption. The relative increase in energy for improved filtration is greatest in climates with the lowest heating and cooling loads (i.e. Log Angeles and San Francisco).
For all cities other than Los Angeles and San Francisco, increasing the ventilation rate resulted in increased energy consumption (Figs. 4a and 5a). As expected, the increase in annual HVAC energy consumption over the baseline was largest for 100% outdoor air and ranged from 17% in Miami to 92% in Minneapolis. The magnitude of increase in energy consumption is higher in cities which fall within and above IECC climate zones 4 due the greater use of backup electric resistance heat during the winter, which is less efficient than the heat pump. The energy increase to bring under-ventilated classrooms up from 3.5 to 7 l/(s.person) was less than 8% in climate zones 3 and below and was between 10 and 35% in climate zones 4 and above.

Los Angeles and San Francisco were a special case amongst all the cities modeled; an increase in ventilation air up to 100% outdoor air reduced total annual HVAC energy consumed by 16% and 4% over the baseline, respectively. This is because the climate is generally mild during school hours (which exclude June, July, and most of August) and the classroom has high cooling loads. Therefore, increased outdoor air during all occupied hours on an annual basis reduces the annual amount of compressor cooling required. This occurs despite an increase in compressor use on warm afternoons. Therefore, from an energy perspective, increased amounts of outdoor air are favorable for these cities specifically and for other cities having similar climates. Although cities having such mild, coastal climates are limited in the U.S., they are often densely populated, and many students and teachers can benefit from increased ventilation without increased energy costs. However, such benefits may be challenging to fully realize in cities that experience poor outdoor air quality, as occurs especially in regions that experience major wildfire smoke events.

Economizer operation simulated according to the logic described in 2.3 led to lower annual electricity consumption, with up to 200 kWh less consumption in Phoenix and Los Angeles. This energy consumption coincided with an increase in the average ventilation rate above the baseline by up to 3.8 l/(s.person) in Phoenix (for a total of 10.8 l/(s.person)). Because economizers reduce compressor cooling energy, the greatest reductions were seen in cooling-dominated climate zones 1–3. In the cases of Los Angeles and San Francisco, the economizer reduced energy use in comparison to the 100% outdoor air scenario because the economizer only uses 100% outdoor air when it is favorable from an energy perspective (which in Los Angeles and San Francisco, is often, but not always).

The heat pump was sized to meet the heating and cooling loads at a ventilation rate of 7 l/(s.person). Therefore, increasing ventilation to 100% outdoor air led to a substantial increase in hours when the cooling temperature setpoint was not met in several of the cities simulated (Figs. 4b and 5b). Out of the total 1750 occupied hours (08:30 to 16:00) in a year, Miami had the highest percentage of unmet hours (29%) followed by Phoenix (19%) and Houston (18%). These cities are all in climate zones 1 and 2. Similarly, increasing ventilation to 100% outdoor air led to a substantial percentage of the 1750 occupied hours where the heating temperature setpoint was not met in several of the cities (Figs. 4b and 5b) including Denver (12%), Baltimore (9%), and Minneapolis (7%), which are in climate zones 4–6. These cities at 100% outdoor air also had a high percentage of total HVAC energy consumed by electric resistance heat: Minneapolis (72%), Baltimore (37%), and Denver (32%). If equipment were sized to meet these heating and cooling loads at 100% outdoor air, annual cooling and heating energy consumption would increase beyond that reflected in Figs. 4a and 5a. This demonstrates that using 100% air as even a short-term pandemic response measure in cold and hot/humid climates may prove challenging in buildings as currently designed because these unmet heating and cooling hours could lead to significant complaints. However, a 50% increase in ventilation rate to 10.5 l/(s.person) had a small increase on unmet heating and cooling hours and is expected to be achievable in all locations modeled using currently installed equipment.

Many commercial building electricity rate structures have a cost associated with peak electricity demand at the facility, analyzed here as the largest amount 15-min peak electricity demand from the HVAC system occurring during summer (May–September) and winter (November–January) seasons. This metric is important for HVAC system operation because HVAC system peak demand generally coincides with facility peak demand [42]. At 100% outdoor air, Phoenix had the highest impact on the maximum winter peak demand with 90% increase over the baseline (Fig. 4c). This was followed by Los Angeles (47%) and Las Vegas (32%). Although Los Angeles had decreased annual energy consumption with increased ventilation, the winter peak demand increases because of increased use of electric resistance heat for short periods. Summer peak demand increases were the highest for Denver (131%) and Minneapolis (116%) for 100% outdoor air relative to the baseline. This is because in early summer the electric resistance heat is running on cool mornings to meet the heating setpoint.

The total HVAC system energy consumption data (Figs. 4a and 5a) is aggregated for all cities and shown in a box and whisker plot with the data categorized by MERV filtration type (i.e., 8 or 13), use of in-room filtration, and ventilation rate (Fig. 6). As described earlier, the additional energy required for in-room filtration is greater than the additional energy required for MERV 13 filtration in the

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Fig. 6. Box and whisker plots of results for all cities for annual HVAC energy consumption (left) and infection probability (right) categorized by MERV filtration type (8 or 13), use of in-room filtration (IRF), and ventilation rate. For each filtration and ventilation combination, the following statistics are shown (outliers excluded and shown as single data points): minimum (bottom bar), 25% percentile (bottom of box), median (bar), mean (X), 75% percentile (top of box), and maximum (top bar).
central HVAC system. The outlier high energy use for 100% outdoor air case represents Minneapolis, which requires substantial heating energy, leading to about 2.6 times the average energy consumptions of all the cities.

The probability of a person transmitting an infection to at least one of 26 susceptible people through the air is aggregated for all cities and shown with the data categorized by MERV filtration type (i.e., 8 or 13), use of in-room filtration, and ventilation rate (Fig. 6b). Increased ventilation rate decreased the probability of transmitting an infection; the correlation is strongest with the lowest level of filtration (MERV 8). Within each bar, the highest infection probability is for locations with HVAC systems that have the lowest supply airflow rate due to the size of the heat pump used (Table 5). The rate at which indoor air is filtered is equal to the supply airflow rate minus the outdoor airflow rate, so lower supply airflow rates result in lower filtration air changes per hour. This means that locations with the largest cooling loads (and therefore largest air handling systems) will have the best central filtration potential and lower infection probability for a given ventilation rate. For example, at a ventilation rate of 7 l/(s.person) in Miami, Houston, Phoenix, and Las Vegas (infection probability with MERV13 of 2.7%) upgrading filtration to MERV 13 notably reduces the magnitude of the correlation between ventilation rate and infection probability. This is a result of the substantially higher filtration efficiency for a MERV-13 filter compared to a MERV-8 filter for particles in the size range relevant to airborne disease transmission. The higher the filtration efficiency the more similar the filtered airflow is to outdoor air in terms of reducing infection probability.

The reduction in infection probability was compared for two scenarios in relation to the baseline: 1) changing from a MERV 8 to a MERV 13 filter, or 2) increasing ventilation by 50% to 10.5 l/(s.person). The average infection probability is lowered by 19% with the filter upgrade, whereas increased ventilation of 10.5 l/(s.person) lowers average infection probability by 8.5% over the baseline case. Although both the methods lower infection probability, MERV 13 filtration results in a lower infection probability with an average 10% lower annual energy consumption than the increased ventilation. Addition of in-room filtration reduced the range of infection probability across all locations since use of IRF reduces the importance of the central HVAC system supply airflow and filtration rate. Given the CADR of 302 l/s, the in-room filtration corresponds to an additional equivalent per-person filtration rate of 11.2 l/(s.person) for 27 people on top of the 7 l/(s.person) ventilation rate baseline. Compared to the baseline, the average infection probability is lowered by 32% for 100% outdoor air with standard filtration and by 46% for outdoor airflow of 7 l/(s.person) with MERV 13 and IRF. Additionally, average annual energy consumed by improved filtration system (MERV 13 + IRF) is 31% lower than the energy consumed by 100% outdoor air. Thus, improved filtration (either through use of MERV 13 filters, IRF, or both) on average leads to lower infection probability and lower energy costs in comparison to increasing ventilation rates above 7.0 l/(s.person).

Increased ventilation due to economizer operation resulted in lower infection probability (Fig. 6 hashed bars, and Fig. 7). Fig. 7 illustrates the relationship between both the annual energy savings and the decrease in the infection probability, relative to the baseline, as a function of the increased ventilation that is achieved by use of the economizer. There is a reasonable linear relationship between increased ventilation from economizer operation and both energy savings ($R^2 = 0.97$) and infection probability reduction ($R^2 = 0.88$). Los Angeles, Phoenix, Albuquerque, and Las Vegas achieve higher energy savings due to a combination of high cooling loads, a significant number of cooling hours with outdoor air temperatures below the economizer limit, and large HVAC system size (which increases maximum ventilation rate possible during economizer operation). Although the infection probability reduction resulting from economizer operation is relatively small, it is an added benefit on top of energy savings. Temperature-based economizer controls are widely available and are a standard option on packaged HVAC systems.

4. Discussion

The infection probability metric presented in Figs. 6 and 7 was calculated with a quanta generation rate of 1.25 h$^{-1}$. Quanta generation rate varies by pathogen, occupant activity, and whether or not masks are worn [36,39]. To consider the impact of a higher quanta generation rate, the probability of one infectious person transmitting to one of 26 remaining susceptible occupants was recalculated for a quanta generation rate increased by 10 times (from 1.25 to 12.5 h$^{-1}$), which reflects, for example, a case where masks are not worn (a 4 times multiplier on quanta generation) and the pathogen is 2.5 times more infectious. The resulting relationship between the infection probabilities is well-fit by a quadratic function (Fig. 8), although use of this particular functional form is valid only over the range shown and does not have physical meaning. This relationship can be used to convert the P26 result shown in Fig. 6 to higher infection probabilities expected for an increase in magnitude of the quanta generation rate. For HVAC scenarios with lower infection probability (i.e. greater filtration levels, 2-3% on the x-axis in Fig. 8), the probability of infection is approximately 10 times greater, scaling 1:1 with the 10 times increase in quanta generation rate. However, for HVAC scenarios with higher infection probability (i.e. lower filtration and ventilation rates, 6–7% on the x-axis in Fig. 8), the probability of infection is approximately 8 times greater for the 10 times multiple of quanta. This is because the probability of transmitting an infection asymptotes to 100% (i.e., this approach is calculating the probability of infecting at least one individual, not the total number of individuals infected).

The scope of the current work does not consider that high ventilation rates increase occupant exposure to outdoor air pollutants. Further work needs to consider and quantify the impacts of ventilation and filtration in the context of time-varying levels of outdoor particulate matter seen across the U.S. Generally, the increased filtration methods considered in this work (MERV 13 and IRF) will also reduce indoor exposure to particulate matter of outdoor origin. While filtration is an effective and energy efficient method to reduce exposure to both respiratory aerosol particles and outdoor particulate matter, standard MERV 13 and IRF do not remove gas-phase pollutants. Increasing outdoor airflow rates to reduce indoor infectious disease transmission is also expected to lower concentrations of volatile organic compounds (VOCs) from indoor sources and reduce carbon dioxide (CO$2$) (exhaled by occupants), a benefit that would not be achieved by mechanical filtration methods. Building standards require a minimum level of outdoor air ventilation to
dilute indoor concentrations of these gas-phase pollutants (VOCs, CO₂) that are not removed by standard filtration methods. As described in the introduction, 7 l/(s.person) is generally considered adequate outdoor air ventilation to achieve this goal [28, 29]. The additional health and productivity benefits of reduced exposure to indoor VOCs and CO₂ from increased ventilation are not considered in this analysis and are an area for future work.

Air cleaning methods are an area that is rapidly changing with both technology and test methods currently under development. For this work, the IRF model assumes a CADR of 302 l/s, which is the flow rate of clean, pathogen-free air delivered by the filter. Although the accepted standard today is HEPA filtration, other technologies could be modeled using the Wells-Riley approach if an equivalent CADR metric can be determined through appropriate test methods and analysis.

5. Conclusion

Our study analyzed the effect of ventilation and filtration on a classroom’s HVAC energy consumption and the probability of airborne disease transmission between occupants. The work incorporated an annual simulation for a classroom and HVAC system with hourly weather data for 13 cities across the U.S. Our study modeled a packaged HVAC system with actual efficiency curves and addressed the variability of locations by changing the design of the building envelope according to DOE reference models. HVAC system sizing for each geographic location was determined by running the annual simulation with a fixed ventilation rate of 7 l/(s.person), while varying the heat pump size and electric resistance back-up heat. Once the system was sized, follow on simulations were run that encompassed five ventilation system configurations (four constant outdoor air ventilation rates as well as one economizer configuration) and three filtration levels (central MERV 8, central MERV 13, and IRF). The probability of infection was calculated according to a CDPH model (based on the Wells-Riley equation), and was used to estimate the probability of an infected person
infecting a susceptible person through airborne transmission with and without masks.

The results showed that replacement of a MERV 8 filter with a MERV 13 filter led to a minimal increase (1.5% average) and IRF led to a moderate increase (10.7% average) in annual HVAC system energy consumption over the baseline. In addition, the results showed that a 50% increase in ventilation flow to 10.5 L/(s.person) led to a 13% average increase and 100% outdoor air led to a 45% average increase in annual energy consumption. Increases in ventilation not only increased energy consumption, but also increased energy peak demand and resulted in a high number of unmet heating and cooling hours, especially for the 100% outdoor air case. HVAC systems originally designed to provide ventilation at 7 L/(s.person) cannot simply be increased to 100% outdoor air without sacrificing thermal comfort in most cities, except for Los Angeles and San Francisco. Furthermore, results of this analysis showed that, on average, switching the baseline central filtration to MERV 13 and adding IRF was superior to switching the baseline to 100% outdoor air in terms of both HVAC energy consumption and infection transmission probability. Additionally, the analysis showed that the magnitude of infection probability reduction from ventilation and filtration mitigations was greater for higher quanta generation rates.

Economizer operation was also studied in all the cities and the results showed that economizers, while developed for the purpose of saving energy, also have potential to reduce infection probability. The economizer logic used in the simulations of this study is basic with a fixed high limit of outdoor air temperature and is a standard option on most packaged units. However, 2018 survey data showed that only 15% of commercial buildings in the U.S. have an economizer cycle enabled in their building HVAC systems [16]. Further improving dynamic outdoor airflow rate controls to consider both health and energy consumption are an area for further research.

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Credit author statement

T. Pistochini and S. Chakraborty are the lead authors that conducted the literature review, developed the methodology, reviewed results, and wrote the publication. All authors collaborated on development of the EnergyPlus model and application of the infectious disease model. C. Mande programmed the EnergyPlus model and script to run and post-process simulations.

Declaration of competing interest

The authors declare the following financial interests/personal relationships which may be considered as potential competing interests: Theresa Pistochini reports financial support was provided by Trane Technologies plc.

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Appendix A. Supplementary data

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