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Ceiling fans: Predicting indoor air speeds based on full scale laboratory measurements

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Abstract

We measured indoor air speeds generated by ceiling fans in 78 full-scale laboratory tests. The factors were the room size, fan diameter, type, speed, direction (up or down), blade height, and mount distance (i.e. blade to ceiling height). We demonstrated the influence of these factors, showing that the most significant are speed, diameter and direction. With other factors fixed, the average room air speed in the occupied zone increases proportionally with fan air speed and diameter. Blowing fans upwards yields lower but far more uniform air speeds than downwards. We show that for the same fan diameter and airflow, fan type has little effect on the air speed distribution in the region outside the fan blades. We developed several new dimensionless representations and demonstrate that they are appropriate for comparisons over a wide range of fan and room characteristics. Dimensionless linear models predict the lowest, average, and highest air speeds in a room with a median (and 90th percentile) absolute error of 0.03 (0.08), 0.05 (0.13), and 0.12 (0.26) m/s respectively over all 56 downwards tests, representing common applications. These models allow designers to quickly and easily estimate the air speeds they can expect for a given fan and room. We include all measured data and analysis code in this paper.

Keywords:

Ceiling fan; Air speed distribution; Full-scale laboratory testing; Rotational speed; Fan diameter; Fan direction

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Highlights:

- Measured air speed distribution in 78 full-scale laboratory tests

- Average air speeds increase in direct proportion to fan rotational speed and diameter

- Blowing fans upwards yields lower but more uniform air speeds than downwards

- At equal diameter and airflow, fan type doesn't affect air speeds in most of the room

- Developed easily applied models to predict indoor air speeds with ceiling fans

Graphical Abstract
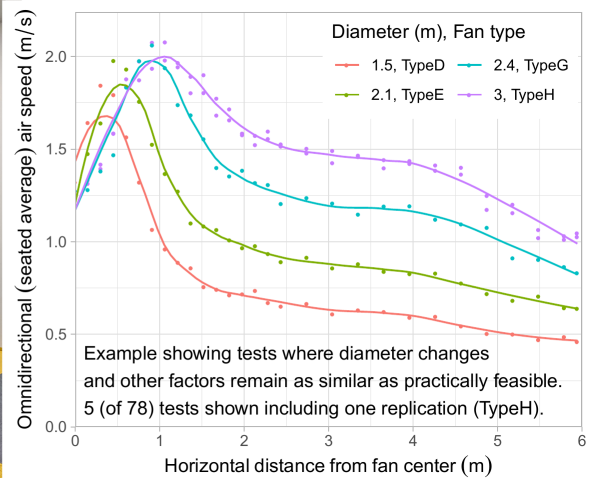
1. Introduction

1.1. Benefits of air movement in buildings

Having the ability to increase the air speed in a room in a controlled manner provides many advantages. It increases the heat transfer from occupants to the environment by convection and evaporation, allowing them to remain comfortable in warmer conditions [1–3]. Many laboratory studies show that air movement provides comfort in warmer conditions [4–7] even at 30°C and 80% RH [8] and this is accepted in existing thermal comfort standards (e.g. [9–11]). A field study intervention adding ceiling fans to an air-conditioned office found that occupants were equally or more comfortable at 26-27°C with increased air movement than at 23°C without [12]. Giving occupants control over air movement provides an instantaneous way to respond to changing thermal comfort needs, responding faster than possible with Heating Ventilation and Air Conditioning (HVAC) equipment designed to condition the whole room [13].

There are significant energy savings from being able to provide comfort in warmer conditions. Estimates range from 5-10%/°C temperature increase [6,14–16]. There are other benefits such as improved perceived air quality [17–19], and destratification (where this is problematic). Finally, thousands of occupant satisfaction surveys with coincident measurements of indoor conditions show that occupants prefer more air movement than they are currently experiencing in buildings [3]. Thus, it is clear that the ability to increase air movement in a room in a controlled way is desirable from many perspectives.

1.2. Terminology and nomenclature

We commonly see different terms used to describe similar, but not identical concepts, which differ between papers and sometimes even within the same paper. We describe each term here and use it throughout.

- Fan rotational speed (N): Physical fan rotational speed (rpm).
- Fan airflow (Q): Volumetric airflow rate through the fan blades (m³/s).
- Fan air speed (SF): Average air speed through the area swept by the fan blades (m/s).
- Air speed (SO): Air speed (m/s) at a point in the room, or a summary statistic of the air speed distribution, such as the median or area-weighted average.
- Occupied zone: Volume of the room at or below 1.7 m height.
- Air speed distribution: The full set of measured air speed data in the occupied zone.
- Blade height (H): Distance from floor to blade, measured at hub (m).
- Mount distance (M): Distance from blade to ceiling (m).
- Ceiling height (C): Distance from floor to ceiling (m).
- Seat: The seated average air speed (average of 0.1, 0.6 & 1.1 m measurements).

- Stand: The standing average air speed (average of 0.1, 1.1 & 1.7 m measurements).
- Area-weighted average air speed: the sum of each seated or standing average measurement location weighted by the fraction of floor area which that location best represents (see included analysis code for detail).

We also create several dimensionless variables for the analysis:

- *xd*: horizontal distance from fan center to measurement location (X) divided by fan diameter (D)
- *xr*: X divided by room size (R)
- *zh*, vertical distance from floor to measurement location (Z), divided by blade height (H)
- *dr*: D divided by R
- *do*: D divided by the occupied zone height (1.7 m)
- *hd*: H divided by D
- *md*: mount distance (M), divided by D
- *cd*: ceiling height (C), divided by D
- *so*: omnidirectional air speed (SO), divided by estimated fan air speed (SF)

1.3. Technical barriers to use of increased air movement

Possibly the largest technical barrier to designing for increased air movement is the absence of a simple method for determining the air speed distribution a fan (or fans) will produce in a room. The absence isn't surprising, since the fan design problem is potentially complex and there is an absence of measured air speed data in realistic conditions that might otherwise provide design insight. Literature to date is sparse and in aggregate explores a very small range of parameters that designers need to evaluate. For example, all published experiments to date used fan diameters from 1.1 - 1.5 m, though they are available in diameters from 0.6 - 7.3 m, and measured one-size fan in a one-size room.

More information is becoming available. The US Code of Federal Regulations [20] determines airflow for ceiling fans sold in the USA using standard test-methods. For fans 7 ft (2.13 m) diameter and under, the test-method measures an air speed traverse below the fan [20]. For larger fans, the test-method (AMCA [21]) measures thrust. There are databases containing performance data for thousands of ceiling fans [22,23]. Additionally, there is a proposed standard for measuring the air speed distribution in a specified room size [24,25]. However, these resources fall short of meeting designers' needs. There is no clear, generalized model of the effects that many characteristics - room size, blade height, furniture, etc. - have on air speed distribution. Even basic questions that designers have, such as 'What size fan do I need to achieve this air speed in this room?', currently remain unanswered.

Additionally, while ceiling fans have an overall cooling effect in the room, they also create a non-uniform thermal comfort environment. Air speeds are higher under the fan than elsewhere in the room, so thermal comfort varies depending on an occupant's location [26]. Gao et. al. [27] showed that when the fan jet impinges directly on furniture, it widens the higher air speed region beyond the fan diameter, however, the majority of the room still has lower air speeds. This difference between higher air speeds in one location than in others may be problematic where there are multiple occupants who cannot freely or easily move about the room, with some too cool and some too warm depending on where they happen to be located. For example, a shared office where one desk is directly under the fan and others are far from the fan. In cases where occupants can move about freely and easily, such as a lobby or cafeteria, this may be beneficial in addressing the natural variability among people - those who desire more cooling can position themselves closer to the fan. This variability affects both steady state (e.g. people who typically prefer cooler temperatures, or are more heavily clothed) and transient scenarios (e.g., one's changing comfort needs directly after commuting to work [28]). A non-uniform thermal environment may be beneficial in both, as long as it is trivial for occupants to relocate. Investigating these scenarios thoroughly - though valuable - is outside this paper's scope.

Last, in addition to the horizontal variability of air speed within the room, there is also vertical variability. Air speeds increase with height while directly under the fan, but this relationship reverses outside the fan jet where they are higher at the foot than the head. Occupants who feel warm tend to prefer cooler heads [29] and people have more surface area in the upper body than the lower body. Thus, this vertical variability may exacerbate the horizontal variability mentioned above. However, current thermal comfort standards ignore this effect, representing air speed using an unweighted average of the measurements at three heights.

Thus, it is clear that information about the air speed distribution in a given scenario is valuable to a designer.

1.4. Review of prior studies investigating ceiling-fan driven air speed distribution in a room

We reviewed previous investigations on air speed distribution induced by a ceiling fan, focusing on the factors which affect that distribution. These factors include fan rotational speed [8,30–33], blade shape and number [31,32,34,35], direction (upward or downward) [30,36–38], mount distance [32,36], ceiling height [32], multiple fans [26,33], and furniture [26,27].

Many prior studies focus on the airflow through a ceiling fan and show that blade geometry and number of blades affect airflow and efficiency [31,35,39–44]. However, these don't focus on the air speed distribution within the room.

Mount distance has received some attention. An empirical study [36] showed that mount distance affected airflow only when the distance between the ceiling and fan (diameter: 1.4 m) was 0.4 m or less. A CFD

study [32] found that when the mount distance was > 0.3 m (diameter: 1.5 m), it does not notably affect air speeds in the room. Chen et al. [32] examined increasing ceiling height and found that with a fixed mount distance, airflow was similar but the air speed decreased in the occupied zone directly below the fan. They also normalized air speeds using the peak air speed at the same height and achieved similar dimensionless profiles at high fan rotational speeds, but not at low speeds.

Several CFD studies [30,37,38] focused on improving disinfection efficacy using fans. Though not the primary focus, they visualize the room air speed distribution and also simulated fans blowing upwards and downwards. Similarly, regarding fan direction, one other study evaluated the effect of fan direction [36]. Although they measured air speed near the fan operating in both directions, the study focused on providing a benchmark for CFD, and made no comparison between the two.

Last, studies have examined other factors that are commonplace in buildings: multiple fans and furniture. Liu et al. [33] measured the effect of single and multiple fans running at different speeds on air speed distribution (fixed fan and room size). Gao et. al. ([27]) measured air speed distribution with different types of furniture directly underneath the fan blades, showing that the furniture deflected higher air speed towards the edge of the table, notably increasing seated average air speeds compared to cases without. Both studies provided extensive data sets and proposed conceptual models of air circulation for the evaluated cases. Mihara et al. [26] used a thermal manikin to measure local cooling effects in a room with furniture and two fans and visualized air speed distribution. The fans had a fixed size and location, and the experiment operated them at three different speeds.

1.5. Objective

This paper's primary goals and their novelty are:

- To measure how different room- and fan-related factors affect air speed distribution. This study is more comprehensive than any in the existing literature in both the number of factors evaluated and the number of tests performed.
- To develop simple-to-use dimensionless models to predict that air speed distribution that only require inputs that are readily available at design stage. The existing literature does not provide a model to make these predictions.

2. Methods

2.1. Factors and levels

We chose factors to include all those identified in the literature review that directly affect the room air speed distribution, and defined the levels based on a wide range of constraints, such as: safety requirements

6

(e.g., the minimum fan heights when the fan does or does not meet UL 507 criteria); existing facilities (the existing test chamber matches the 6.1 m chamber required by ASHRAE 216); commonly available sizes of fans and their mounts; and typical application ranges of fans in practice (e.g., in terms of the ratio of fan diameter to room size).

- Room size (2 levels: 6.1 & 12.2 m)
- Fan diameter (7 levels: 1.22, 1.32, 1.52, 2.13, 2.44, 3.05 & 4.27 m)
- Fan type (9 anonymously reported fan types from 5 different manufacturers, ranging from 3-8 blades/airfoils)
- Fan air speed (from 0.63 to 2.76 m/s, as described later)
- Fan direction (Down or Up)
- Blade height (4 levels: 2.13, 3.05, 3.66 & 4.27 m)
- Mount distance (3 target levels: 0.6, 1.2 & 1.8 m. We report the actual mount distance, which differed slightly from these due to each fan type's mounting constraints.)

2.2. Test description

Figure 1 shows the layout of the experiment and the nomenclature used throughout this paper. For each test, we installed the fan in the center of a square test chamber (6.1 m or 12.2 m wide) at the desired mount distance, then raised the ceiling to achieve the desired blade height. We measured air speed (SO) at fixed locations along one radial line from the center perpendicular to the wall in 15 cm increments, increasing to 30 cm increments at 2.44 m from center (just outside the blades of the largest fan). This yields a higher measurement density near the fan where air speed changes more quickly. We included an additional measurement 0.15 cm from the wall to measure air speed close to this boundary. Using this approach, we assume a symmetrical air speed distribution orthogonally around the fan axis. Preliminary testing showed symmetry along 4 orthogonal traverses, and the close fit between replications in the experimental dataset also demonstrates symmetry.

We took measurements at 8 heights at each location, 4 of which we fixed at 0.1, 0.6, 1.1, and 1.7 m to match with the measurement heights used in existing standards [9,45]. We took 4 other height measurements at fixed fractions of the blade height in increments of 0.1 so that we can compare measurements at the exact same dimensionless fraction of the fan height in different scale tests[1].

We measured air speed using omnidirectional probes designed for low-speed measurements (AirDistSys5000, Sensor Electronics, Poland), accurate to ± 0.02 m/s or 1% of reading from 0.05 to 5 m/s, which meets and

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[1]Where the fixed fraction and fixed height measurements were within 5 cm of each other, we measured exactly at the fixed height and added a fixed fraction measurement. For example, for 3 m blade height, the fixed fraction measurement heights were 0.3, 0.9, 1.2 and 1.5 m, corresponding to fixed height fractions of 0.1, 0.3, 0.4 and 0.5 respectively. The 0.2 fixed fraction equals the fixed height of 0.6 m.
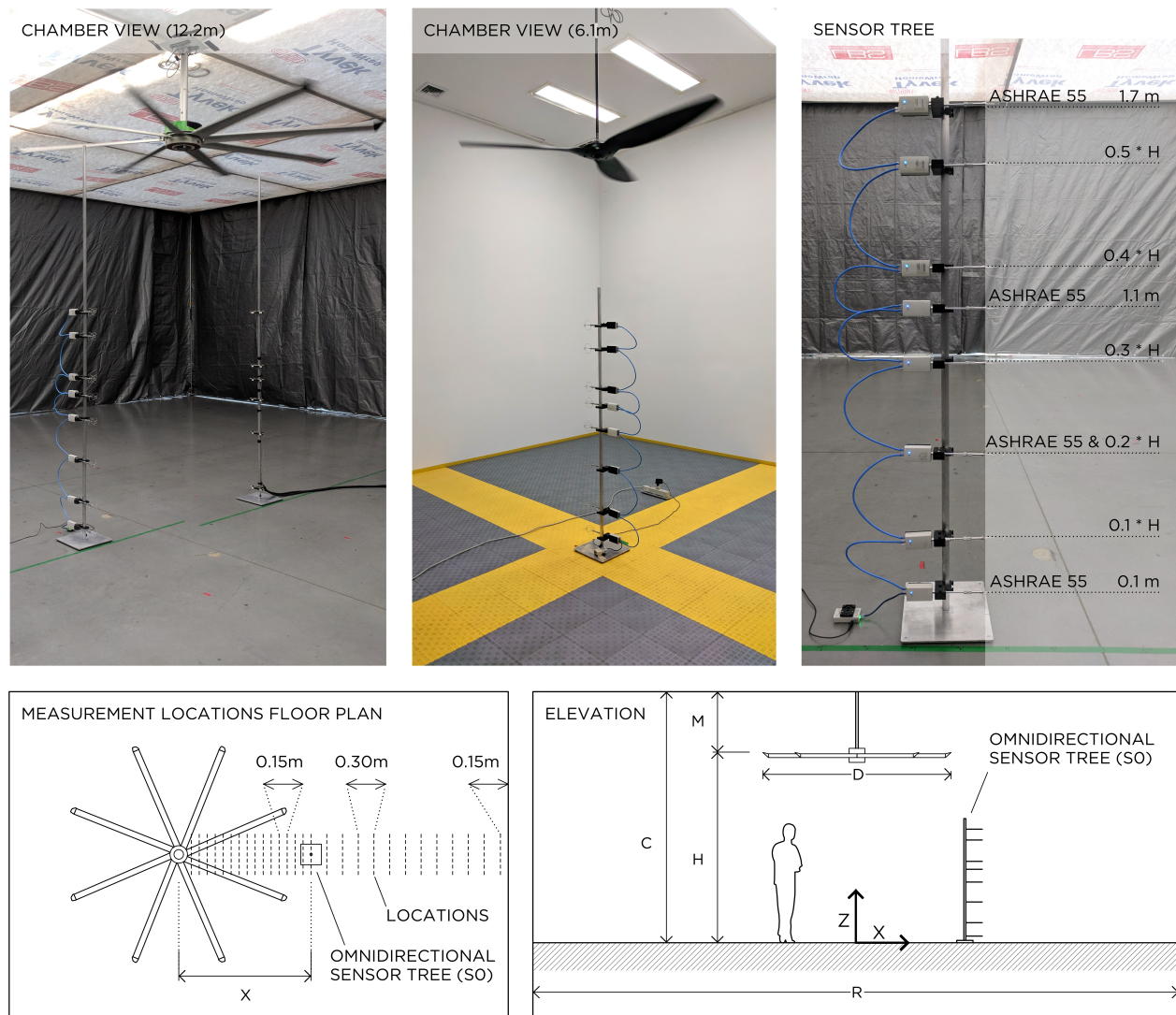
Figure 1: Overview of experimental setup

exceeds the desirable measurement criteria for this application [45]. After moving the measurement tree to each new location, we waited two minutes before recording data. The sensors sample at 8 Hz and output at intervals of 2 seconds. We report each measurement as the average of 90 of these consecutive intervals over three minutes.

2.3. Characterizing fan movement

Ceiling fans sold in the USA are required to have a rated maximum airflow [20]. The rated airflow may be available at other speeds, though the fan affinity laws easily can approximate this - the airflow is linearly proportional to the fan rotational speed[2]. Figure 2 illustrates this by showing rated airflows for each of the 9

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[2]The linear fit intercept is below 0 in all cases. When only a single airflow datapoint is available (typically at the maximum fan rotational speed), a linear fit must assume the intercept is zero. This overestimates the airflow at fan rotational speeds other

8

fan types in this experiment. Following another fan affinity law, with all other design parameters identical, airflow is proportional to the diameter cubed. Aside from the affinity law relationships, in practice there are other reasons why the maximum airflow for a given diameter varies based on the fan type. For example, there are constraints related to the maximum fan rotational speed, blade geometry, number of blades, and UL 507 blade tip speed constraints. This last safety constraint applies to any fan that can be mounted under 10 ft (3.05 m) blade height.

In order to compare different fans, we must characterize how fast a fan moves in a way that is generally applicable across a range of fan types, rotational speeds, and diameters. We compared several potential approaches based on the rated airflow divided by different powers of the diameter or the rotational speed. Divided by the diameter squared shows the smallest relative difference between the minimum and maximum across all fan diameters and types, taking into account the combined effects of the affinity laws and fan type-dependent differences. Thus, we characterize the concept of how fast the fan moves using the 'fan air speed', defined as the airflow divided by the area swept by the blades - a combination of physical measurements representing the average air speed through fan blades.

In the experiment we controlled this factor by setting the fan rotational speed to achieve a target fan air speed (in increments of 0.5 m/s). For fans with discrete speed settings, we chose the setting to minimize the difference between fan air speeds across all of the fan types. For variable speed fans we matched the fan air speed to the average of the discrete speed fans. For example, for a fan air speed level from 1 to 1.5 m/s, two fans with discrete speed settings had fan air speeds of 1.19 and 1.3 m/s respectively, within that range. We set the variable speed fans' rotational speeds to yield the average of these (1.25 m/s). When comparing different fan types to each other, we endeavored to do so under conditions as similar as possible, matching the fan air speeds based on linear regression to the rated airflow data as closely as possible given the available speed settings.

2.4. Design of experiment

There were a large number of factors (7) in this experiment, many with several levels, and certain combinations are not feasible.[3] Thus, a full or fractional factorial design of experiments is not possible. We used the following approach to determine which factor combinations to test, given a total lab time constraint.

- Local sensitivity tests: For each factor we determined a region that is relatively typical, and within

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than the maximum. This does not affect the results in this paper as we tested fans for which we have only one rated airflow (Types A-C) at the maximum rotational speed.

[3]Due to safety, physics, or fan type constraints. For example, large diameter fans cannot be used with blade heights lower than 3.1 m for safety reasons as they typically do not meet UL 507 requirements on blade velocity and thickness. However, smaller diameter fans are commonly used at lower blade heights.
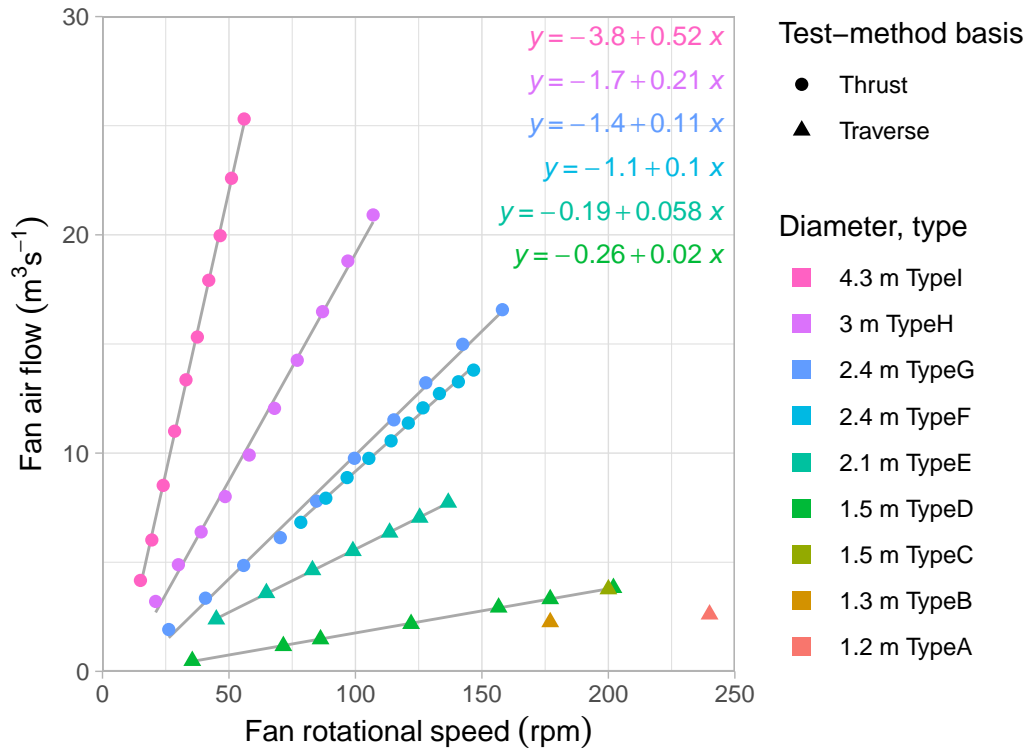
Figure 2: Airflows for each fan in the experiment.

which testing multiple levels was feasible. We tested each level with all other factors held at identical values (or as similar as possible given practical constraints).

- Double scale tests: We performed tests in which the fan diameter, blade height, mount distance, and room size were all twice as large as in otherwise similar tests.

- Similar dimensionless values: We developed some preliminary dimensionless ratios: the ratio of blade height to diameter and the ratio of diameter to room size. We performed several tests in which we held those ratios constant with the ratios in other tests, but at a larger scale, while keeping other factors constant. For example, we matched the 4.3 m diameter fan at 4.3 m blade height with a 2.1 m diameter, 2.1 m blade height test, but other parameters, such as room size, remained the same.

- We performed 12 replications.

- We performed one still air test per chamber.

We used the "AlgDesign" package to optimize the tests ([46]) for the remaining time available after accounting for the tests above. This maximized the value of the remaining tests in creating a mixed effects model. We chose an I- instead of D-optimal design as those perform better in prediction applications when

the model is not known in advance ([47]). We randomized the test order where feasible[4] and performed 78 tests in total. The supplementary material [48] contains a table describing factors and summary statistics of the results for each test.

2.5. Reproducible research

We wrote this paper using R Markdown. All the text, references, bibliography, data analysis and visualization occurs in one file (.Rmd), which automatically builds the document that we submitted to the editor. The supplementary material [48] contains the .Rmd file as well as the entire measurement dataset.

3. Results and discussion

We present results starting with the still air and replication tests, followed by local sensitivity tests for a particular factor. Due to space constraints we display only some of the tests to illustrate a particular concept and we display only the lowest and highest measurement heights in the occupied zone (0.1 m and 1.7 m), and the seated average. The supplementary material [48] contains a figure showing every measurement for each test. We named the figures using a shorthand notation and use similar notation in this paper where appropriate. For example: "R12 D2.4 H3 M0.69 TypeF Down N108 SF2.1 RepA.pdf" corresponds to a test in a 12 m room using a 2.4 m diameter, 3 m blade height, 0.69 m mount distance, TypeF fan blowing downwards at 108 rpm. The fan air speed is 2.1 m/s and this is replication A. The supplementary material figures include measurement uncertainty error bars and overlay the standard error for the smoothing line fits; we omitted these in the paper for visual clarity.

We often report summary statistics using either the lowest, the highest, or the area-weighted average air speed that a seated or standing occupant would experience in the room. We calculate the last by weighting each seated or standing average measurement by the fraction of floor area which it best represents (see analysis code for detail).

3.1. Still air

The upper quartile of the measurements taken in tests without the fans operating was 0.05 m/s, just above the minimum measurement range of the anemometer. Thus, the conditions are effectively still air; well below the threshold of having any appreciable cooling effect on a person. This corresponds with measurements in 5 case study buildings [49], which showed similarly low air speeds in the absence of fans or operable windows, and Rohles' paper [4].

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[4]Changing most factors (e.g. changing a 4.3 m diameter fan, or moving the ceiling) takes longer than performing a single test and thus, full randomization is infeasible as it would vastly reduce the total number of tests. We grouped tests by these difficult factors and randomly selected the sequence of the remaining factors.

3.2. Replications

We performed 11 replicated tests in randomly selected order covering a wide range of factors, and they show very close agreement. The median difference between air speeds measured at the same point in replicated tests is effectively zero (0.006 m/s). The median absolute difference - which represents the typical air speed difference at any given location, ignoring whether one is higher or lower than the other - is 0.03 m/s; the upper quartile is 0.06 m/s across all replications. These differences are close to instrument accuracy (± 0.02 - ± 0.03 m/s over the dataset's range) indicating that the tests are highly replicable. The median absolute difference between replicated tests was slightly higher (by 0.02 m/s) in the larger test room (see figure in supplementary material [48]). Also, we note that the small fraction of the dataset (1.7%) where the absolute difference between tests exceeds 0.2 m/s all occur in the region under the fan blades, and predominantly occur for tests at higher fan air speeds (>2 m/s). This may indicate that air speeds in this region - relatively close to the stagnation point - are less stable than in others.

3.3. Fan air speed

Varying fan air speed with other factors fixed has a directly proportional effect on air speed measured at any location. This applies across the range of diameters we tested where speed was the only modified factor. The proportional relationship becomes less accurate at low fan air speeds (<1.0 m/s). This is likely due to inaccuracies in measuring airflow (and thus the fan air speed) in the test-methods, and momentum effects [32]. Figure 3 shows how changing fan air speed affects the air speed distribution with all other factors fixed. The mean standard deviation at each measurement height and location, expressed as a percentage of the average measurement, decreases from 37% (left column) to 7.7% (right column). For context, the same metric for all replicated tests is 3.4%, indicating that much of the remaining variation is measurement uncertainty and variation between tests. This shows that normalizing the data against fan air speed accounts for most of the difference between tests over a wide range of fan rotational speeds.

3.4. Fan diameter

Figure 4 shows that increasing the diameter with other factors constant, or approximately constant (i.e. estimated fan air speed, mount distance), has the following effects: 1) increases the width of the high air speed region below the fan in proportion to the diameter without noticeably changing the maximum air speed at head height within that region. 2) increases the air speed in the region outside the blades proportionally to the diameter. The right column shows that SO/D has a similar profile for each test in this region. It is interesting to note the difference between the two smaller and two larger fans, potentially due to the test-method changing between these tests. 3) decreases the range between minimum and maximum seated average air speeds in the room. i.e., it increases the uniformity of the air speed distribution.
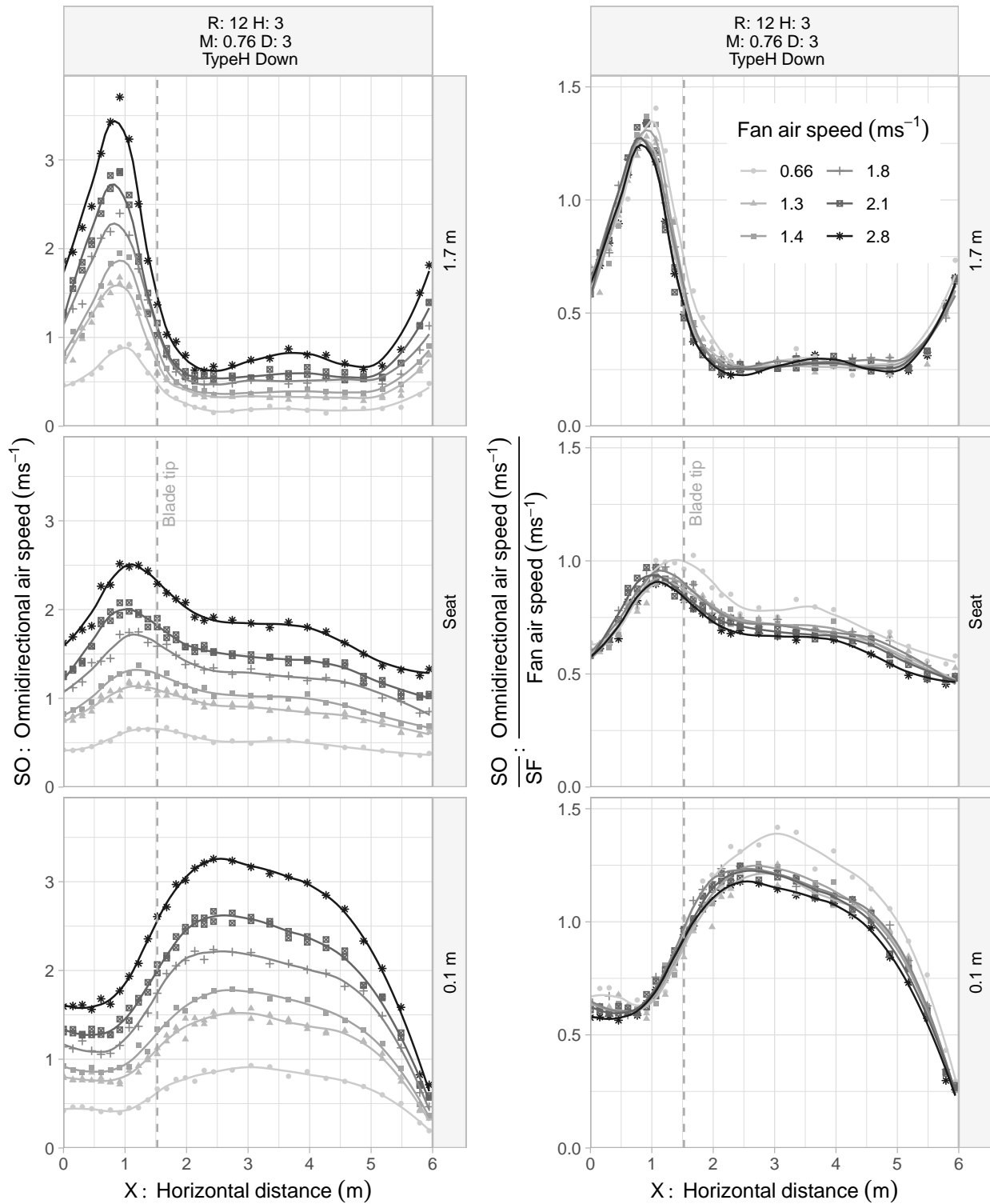
12

Figure 3: Six tests in which the fan air speed (rotational speed) changes but all other factors are identical. Left column: measured air speeds; Right column: normalized against fan air speed.
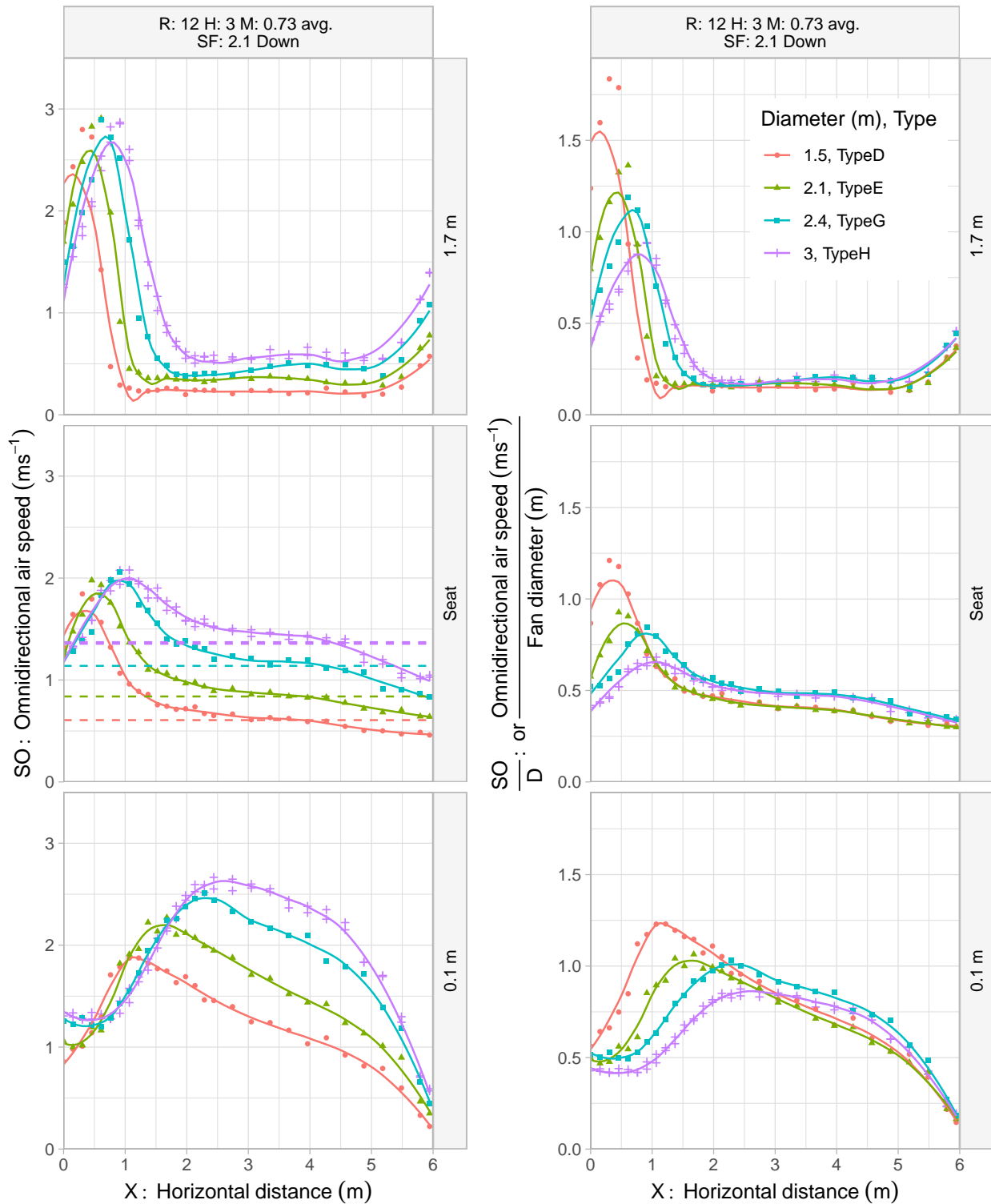
Figure 4: Four different diameters with otherwise similar conditions. Dashed lines indicate the area-weighted average of the seated data. Data for TypeH shows two tests - original and replicated. Left column: measured air speeds; Right column: normalized against fan diameter.

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https://escholarship.org/uc/item/4p479663

3.5. Blade height, ceiling height and mount distance

We used three fixed target mount distance levels (0.6, 1.2 & 1.8 m) and fixed blade heights, which means that ceiling height (the sum of blade height and mount distance) varies between tests. Thus, we can't independently assess these factors' effects.

Below a certain mount distance, the proximity of the ceiling to the blades reduces the fan airflow and causes the width of the fan jet to narrow. This relationship accelerates at extremes - often termed the 'starvation' region - causing the airflow to decrease quickly with mount distance. A simple rule is that starvation occurs when the mount distance is significantly less than 0.25 times the diameter - this equates the surface area of the cylinder swept between blade tip and ceiling to the circular area swept by the blades. Most fan manufacturers have a fan-specific minimum mount distance, often short enough to cause some airflow reduction, but without causing starvation. Studies [32,36] have found values over 0.28 and 0.2 times the diameter, respectively, didn't affect airflow. Fan rotational speed also has an effect; slower speeds allow smaller mount distances before affecting airflow. Further research would be beneficial to develop a generalizable model of airflow, rotational speed, and mount distance. In the experiment, we expect the shortest mount distance (0.76 m) to have some effect for the largest two diameter fans (3 m and 4.3 m) as it is 0.25 and 0.18 times the fan diameter respectively. For these tests (visualized in the supplementary material [48]), decreasing the mount distance with other factors fixed significantly reduced the maximum air speed observed at any of the seated or standing average heights, moved the maximum air speed measurement closer to the center of the fan, and reduced the median measured air speed. For small diameter fans, the shortest mount distances don't affect the room air speed distribution as noticeably. The difference between tests with different mount distances, but otherwise comparable, is only slightly more than a typical replication test. This difference may be due to changing ceiling height, instead of an issue related specifically to the mount distance.

The blade height and ceiling height effects are particularly difficult to decouple from one another given our dataset, however, both have smaller effect on the air speed distribution than either fan air speed and diameter. Blade height mainly affects air speeds directly under the fan blades, and less so in the region outside. In tests where we only increased blade height with other factors constant, the maximum air speed directly under the fan decreased and that maximum point's location moved radially outwards from the center. It is likely that for very small fans, very large heights, and or very low fan air speeds, blade height will have a larger effect, but these are not recommended applications.

3.6. Fan type

Figure 5 (upper row) compares the 2.4 m Type G (8 blades with winglets, M: 0.76 m, SF: 2.12 m/s) and Type F (6 airfoils without winglets, M: 0.69 m, SF: 2.14 m/s) to each other under similar conditions
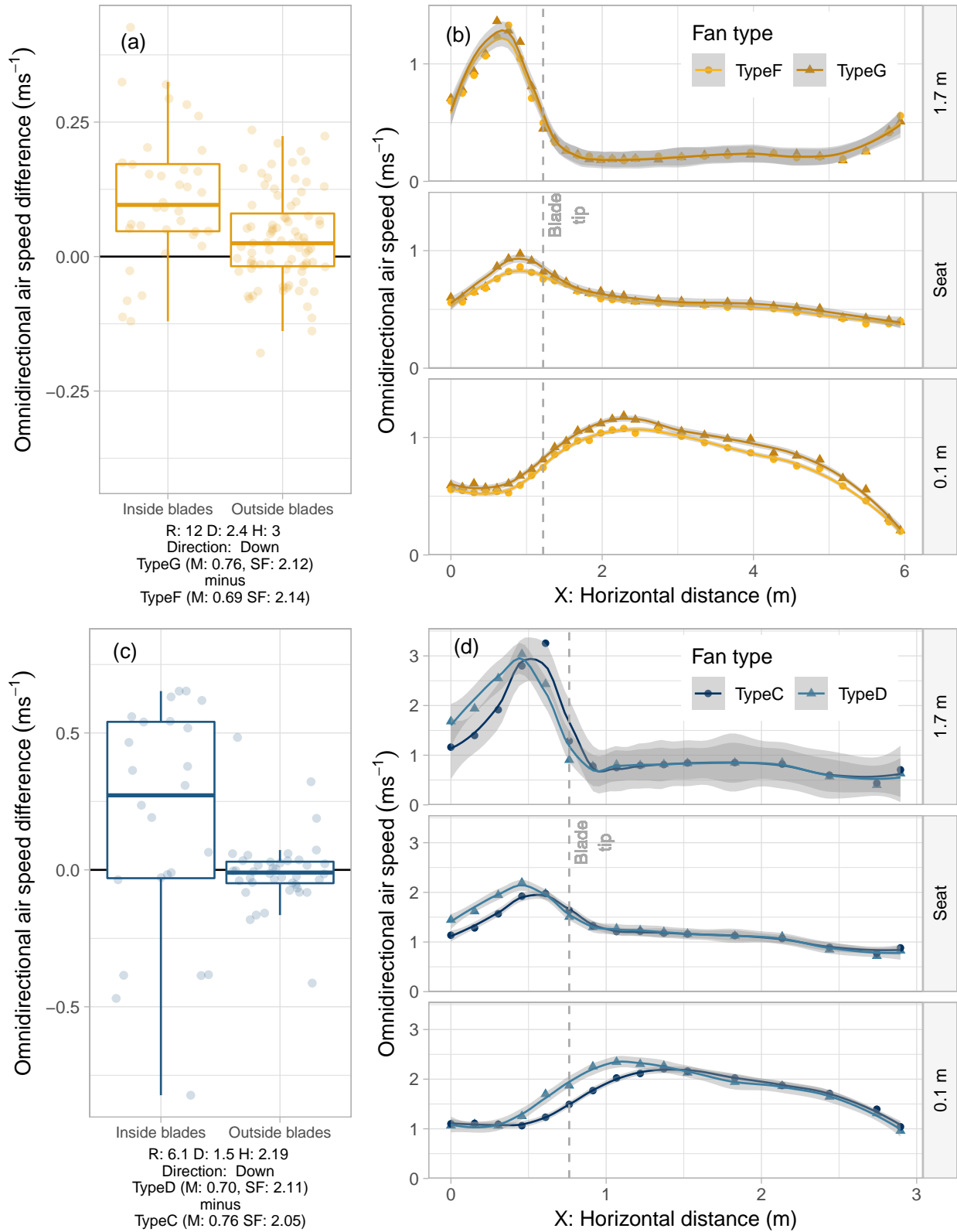
15

Figure 5: Two fan-to-fan comparisons where the fan type changes, but both operate under very similar conditions. Sub-plots (a) and (b) compare TypeG to TypeF, while (c) and (d) compare TypeD and TypeC. Sub-plots (a) and (c) show boxplots of the distribution of the difference between air speeds measured at the same point for both fans, separately for the region inside the fan blades (i.e., directly under the fan) and outside the fan blades.

(see Methods section for details). The median, lower and upper quartiles of the difference in air speeds are 0.05, -0.01, 0.13 m/s. Although larger than the replication test difference, considering that not all factors are exactly equal (mount distance, ceiling height, fan air speed), fan type has a minimal effect in this comparison, particularly outside the blades.

Figure 5 (lower row) compares the 1.5 m Type D (3 airfoils, cross-section varying from hub to tip, M: 0.7 m , SF: 2.11 m/s) and Type C (5 blades, uniform cross-section, M: 0.76 m , SF: 2.05 m/s). The median, lower and upper quartiles of the difference in air speeds are 0, -0.05, 0.19 m/s. There is a notable difference in the region directly under the fan blades, likely due to the differing cross-section from hub to tip. Type D has higher air speeds close to the center and Type C has higher air speeds close to the blade tips. Outside the fan diameter, the seated average data is very similar.

These comparisons show that air speeds differ directly under the fan - more so when the blade cross-section varies from hub to tip for one type but not the other - however, there is little difference outside the blades. Overall, the seated average data is similar for a given fan diameter, regardless of type, assuming each type can operate at the same airflow. This allows for simplified comparison of fan types to each other, though types still compete on other factors such as maximum and minimum airflow capabilities, energy efficiency, control and sensing options, reliability, maintenance, noise, cost and aesthetics.

3.7. Dimensionless representation

Figure 6 uses the dimensionless variables defined in the terminology section to represent the air speeds from all 56 downward tests using the same frame of reference. Here, the x-axis normalizes the horizontal distance against the room size (xr), allowing representation and comparison of different scales. The supplementary material [48] contains a similar plot with xd as the x-axis. The y-axis normalizes against fan air speed (so), demonstrating as in Figure 3 that the air speed measured at any point in the room increases proportionally with fan air speed. As discussed in the literature review, [32] noted a similar linear effect with fan rotational speed, noting that the relationship breaks down at very low rotational speeds. They normalized against the maximum air speed measured in the room at each particular height, for one fan. However, this information is not known without performing an experiment, whereas the fan air speed is apriori obtained from publicly available information. Additionally, normalizing by fan air speed (instead of either rotational speed or airflow) has the major advantage that it allows comparison across a broad range of fan types and diameters. This makes a strong case that the concept of 'fan speed' should be presented using fan air speed, especially when discussing more than one fan type. Using so also allows us to easily extract information about that is generalizable about fans and may be relevant to a designer. For example, the maximum air speed at any measured point (i.e. at any height or location) in the occupied zone is a median, lower and upper quartile of
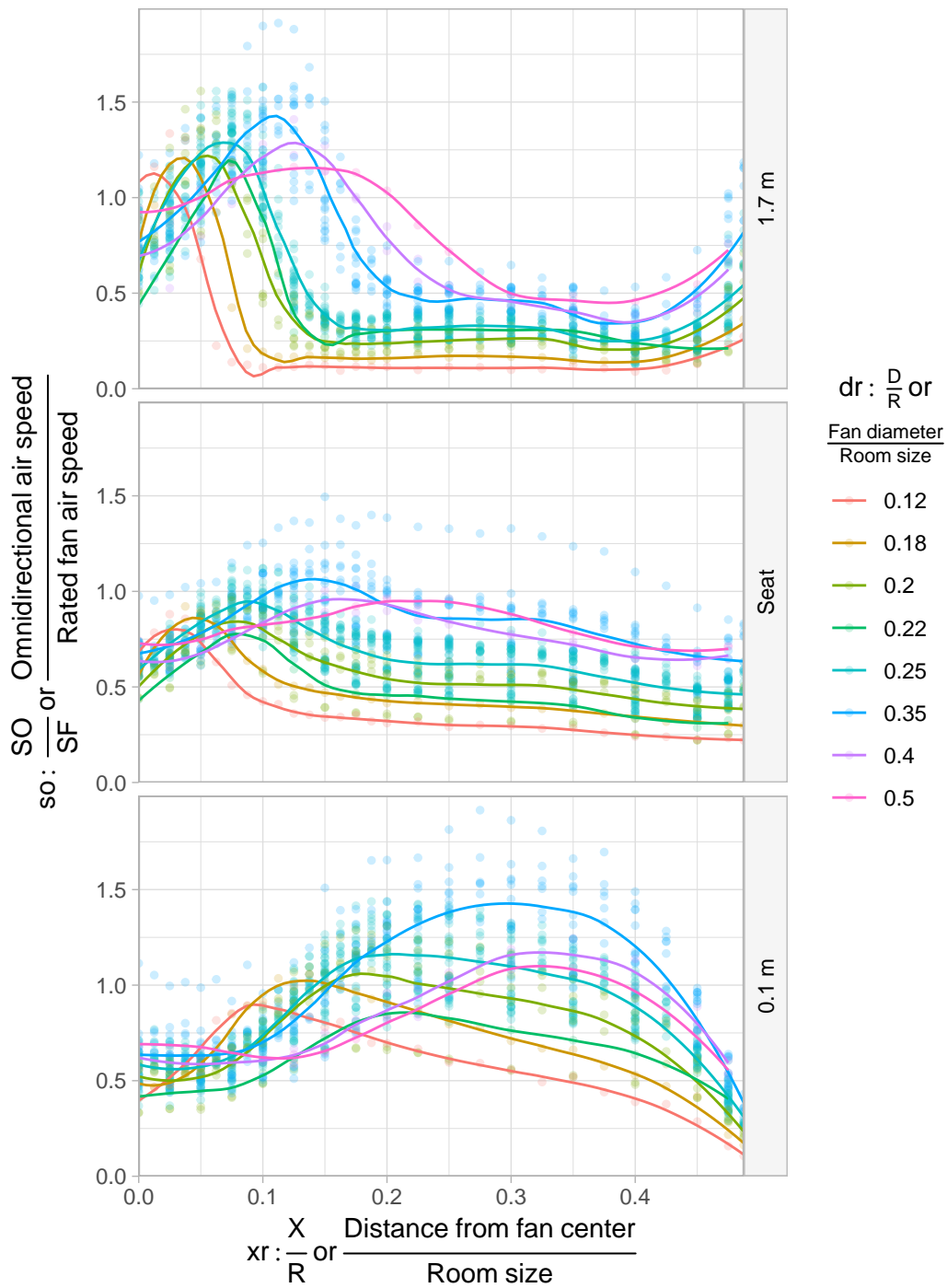
17

Figure 6: Dimensionless representation of all 56 downward tests at three measurement heights.
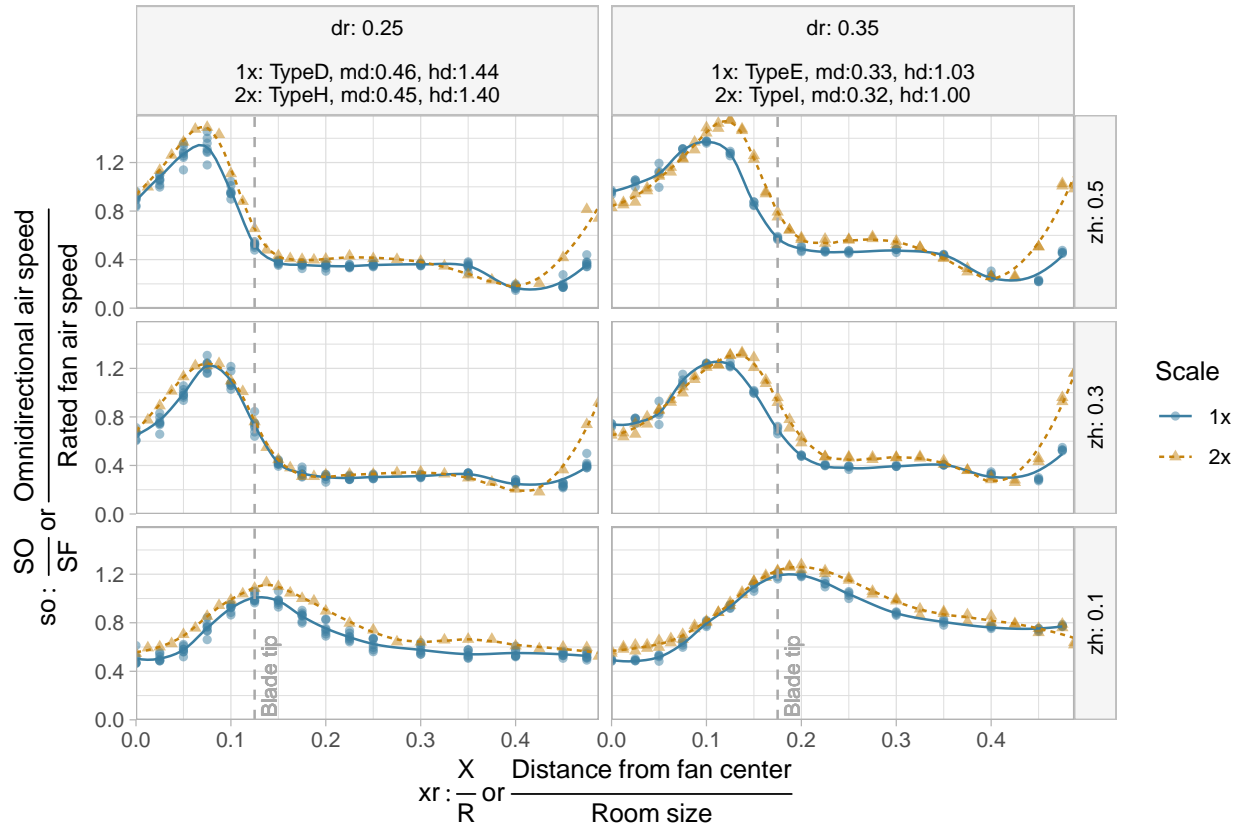
Figure 7: Dimensionless comparison of double scale tests. Results for three dimensionless heights (0.1, 0.3, & 0.5 times the blade height) shown for a total of 14 tests, including multiple different speeds and replications.

1.39, 1.33 & 1.49 times the fan air speed. This applies for all downward direction tests, regardless of fan type or diameter. Lastly, the legend normalized the fan diameter against the room size (dr), clearly demonstrating the effect of a larger fan to room size ratio.

Figure 7 uses dimensionless variables to present a total of 14 double scale tests at different fan air speeds and replicated different numbers of times. Here, every geometry factor is scaled by two, or as close to two as possible due to mount constraints. This includes the measurement height, which we express as a ratio of the blade height. Overall, the profiles are remarkably similar to each other. Note that because of the diameters involved at the two scales, the airflow test-method (and thus the fan air speed and the dimensionless value, *so*) changes. These figures suggest that the traverse test-method reports slightly lower airflows than the traverse. Rating all fans using the same test, or quantifying the difference between test-methods for an identical fan, would be beneficial.

3.8. Dimensionless model

We created models to aid in ceiling fan selection, allowing designers to rapidly estimate the air speeds that an occupant would experience in a room for a given set of fan and room characteristics[5]. The models predict the lowest, area-weighted average, and highest seated and standing air speeds in the room.

We chose to use multiple linear regression so that the resulting model can be explained in text, and the calculation can be performed quickly and easily. Such simply defined models - even if less accurate than other approaches[6] - are valuable to the designer as there is typically limited time available to dedicate to more detailed analyses.[7] The following table describes models for predicting the lowest, area-weighted average, and highest values of *so*, and the associated fit to the results. We include all downwards cases except those in which the mount distance strongly affects the airflow estimate - where $md < 0.20$. We identified these models by searching through all candidate 3-term linear models without interactions (including an intercept and boolean for seated or standing) and selecting the best fitting model using 3-fold cross-validation, repeated 20 times. The linear models for the lowest and area-weighted average data show a slightly better fit using ceiling height instead of blade height. However, when predicting the highest air speed in the room, the opposite was true. The models are notably less accurate in predicting the highest air speed in the room; unsurprising as this occurs in the region directly under the fan blades, the same region in which we showed that fan type has an effect. The ratio of the diameter to the height of the occupied zone accounts for the effect of a fixed height occupied zone in the dimensionless model.

Figure 8 shows a close model fit, with typical absolute error for *so* is within 0.02, 0.04, and 0.05 for the lowest, area-weighted and highest air speed respectively. The 90th percentile and maximum absolute errors for all three models are 0.1 and 0.21 respectively. The largest errors typically occur for tests that are outside the range of expected applications (e.g. a very small diameter fan for the room size, or a very high ceiling height) or in a region where non-linear effects become apparent (e.g. the relationship between fan air speed and measured air speed becomes less accurate below the "Very Low" speed level). For further context, the median (and 90th percentile) absolute error for the lowest, area-weighted average, and highest air speeds (*SO*) in the room are 0.03 (0.08), 0.05 (0.13) and 0.12 (0.26) m/s respectively. These errors seem sufficiently accurate for most applications, particularly when considering the instrument error (0.02 to 0.03 m/s) and that the median replication error between tests was 0.03 m/s.

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[5]Blowing downwards only. Though there are 20 upwards tests, we do not have the data needed to develop a generalizable model for the upwards direction as airflow testing isn't required for fans blowing upwards.

[6]The tests cover a wide range that includes most recommended fan applications. More accurate predictions are possible by performing a regression on a subset of tests that most closely match the desired case.

[7]Even if more time was available, there are other aspects of design that could potentially be more impactful to focus on. Furthermore, other factors (such as furniture, typically unkown at design stage and changeable within the building's life) will likely have a larger effect on the air speed distribution than the error incurred by these models.

Table 1: Dimensionless models for predicting lowest, area-weighted average, and highest air speed in a room, normalized by fan air speed

	Dependent variable:		
	Lowest	Area-weighted average	Highest
	(1)	(2)	(3)
dr: Diameter ÷ room size	0.900***	0.990***	
	(0.045)	(0.071)	
cd: Ceiling height ÷ diameter	−0.017**	−0.060***	
	(0.008)	(0.013)	
do: Diameter ÷ occupied zone height	0.110***	0.110***	
	(0.009)	(0.014)	
hd: Blade height ÷ diameter			−0.180***
			(0.022)
Add for seat vs. stand	0.024***	0.024***	−0.100***
	(0.006)	(0.009)	(0.018)
Constant	0.047*	0.250***	1.300***
	(0.028)	(0.044)	(0.033)
Observations	104	104	104
R^2	0.920	0.880	0.490
Adjusted R^2	0.920	0.880	0.470
Residual Std. Error	0.029 (df = 99)	0.045 (df = 99)	0.092 (df = 101)
F Statistic	304.000*** (df = 4; 99)	188.000*** (df = 4; 99)	48.000*** (df = 2; 101)

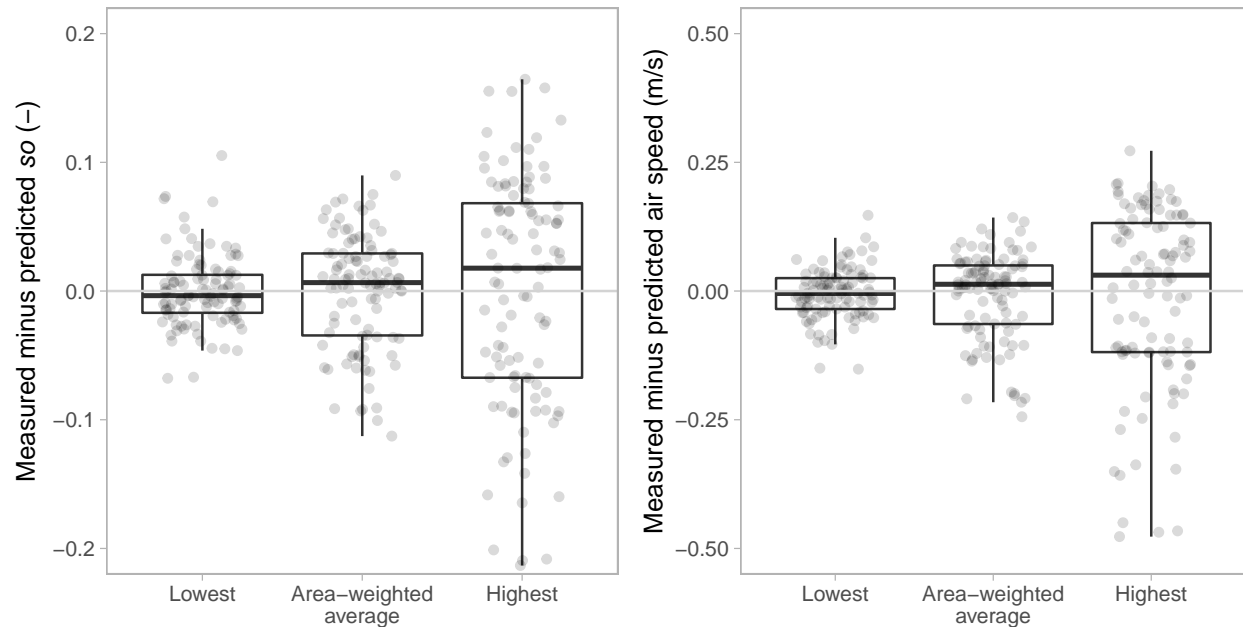*Note:* *p<0.1; **p<0.05; ***p<0.01

21

Figure 8: Linear model residuals for predicting three air speed summary statstics in the room.

3.9. Fan direction

Fans sold in the US must be capable of reversing direction such that they blow upwards towards the ceiling. Of all the tested factors, changing direction had the most effect. Figure 9 shows four examples in which direction changes but other factors are constant. In contrast to blowing downwards, blowing upwards creates a highly uniform seated air speed distribution regardless of location in the room. Air speed also tends to increase with measurement height - higher at the head than the feet - at most locations. However, the air speeds are lower for the upwards direction. The best-performing blade designs tend to have curved cross-sections, which means they are asymmetrical and will move less air when the fan rotates in reverse (blowing upwards). In Figure 9 blade symmetry increases from from left-right, and it is clear that the more symmetrical cases yield higher air speeds in the upwards direction. One can maintain full symmetry in both directions by inverting the physical fan blades (where feasible) as well as reversing the rotational direction. We performed one test where we did this (Figure 9, right-most column). With this configuration, the fan airflow should approximately equal the rated airflow for the downwards direction.

We performed fewer upwards tests (20) than downwards (56), and thus have less data to draw conclusions from. However, in upwards tests where only one other factor changes, we observed the same relationships identified for downwards tests: air speeds increase linearly with fan rotational speed; and increase linearly with the ratio of fan diameter to room size. Blade height, ceiling height and mount distance have less impact (see supplementary material [48] for visualizations).
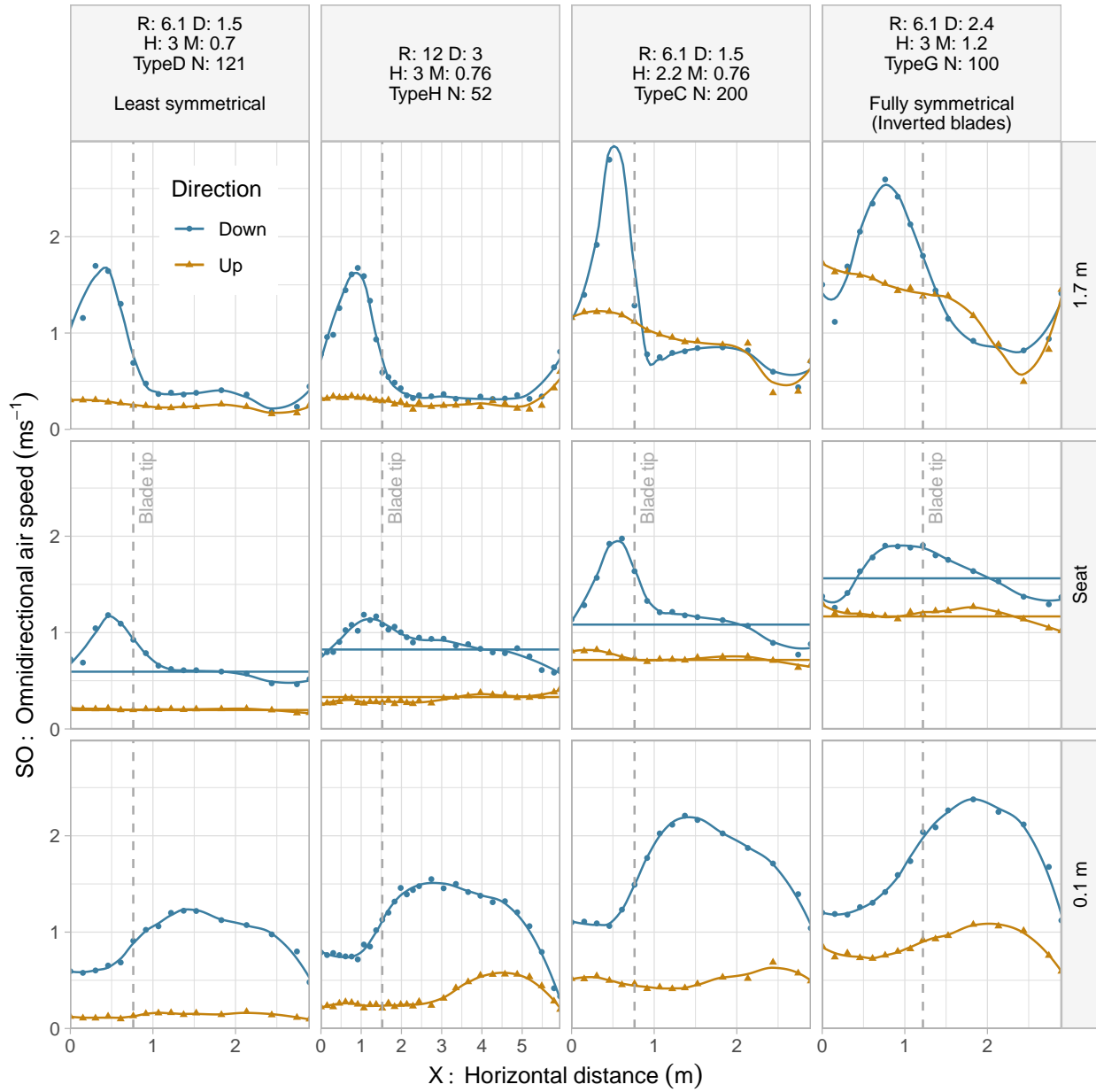
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Figure 9: Comparison between four scenarios in which the fan direction changes and other factors remain fixed. Horizontal lines indicate the area-weighted seated average.

In the fully symmetrical (i.e. inverted blade) comparison, we measured an area-weighted seated average of 1.17 m/s, or 0.55 times the rated fan air speed. That is 25% lower than the identical downwards test (average: 1.56 m/s)[8]. Thus, for the same design area-weighted average air speed in this scenario, a fan blowing downwards could be approximately 25% smaller - or run at 25% slower speed - than a fan blowing upwards with inverted blades.

The upper quartile area-weighted seated average for all of the upwards tests were 0.5 m/s, which is high enough to provide significant cooling.[9] The maximum area-weighted seated average was 1.17 m/s, indicating that despite the lower air speeds achieved by blowing upwards, it's feasible to select fans to achieve a given design air speed. There are limitations to blowing upwards (e.g. the space must be bounded by a ceiling and walls or interaction with the flow field caused by another fan; and the fan must either be larger, or run faster to achieve the same area-weighted average air speed as downwards), and there is a lot of scope for further research (e.g. how satisfied occupants are with the resulting air speed distribution, how furniture and ceiling obstructions affect it in practice, etc.), however, it creates a more uniform air speed distribution which may have many applications[10].

3.10. Uniformity

We quantify the uniformity (U) of the air speed distribution using the following equation for both seated and standing:

$$U = 1 - \frac{SO_{max} - SO_{min}}{SO_{max}}$$

Thus, a value of 1 means a completely uniform air speed distribution while 0 means completely non-uniform. Figure 10 visualizes the uniformity achieved for all tests (except still air), demonstrating that: uniformity is higher for the upwards direction than downwards; uniformity increases with increasing diameter to room size, particularly for the downwards cases; and these relationships are similar for both the seated and standing average data, with seated slightly more uniform than standing.

3.11. Limitations of this study and practical guidance

These experiments detail the air speed distribution along a radial line from fan center perpendicular to one wall in a square room with a centered fan. The air speed distribution will differ along the diagonals from

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[8]For context, we also compare the inverted blade upwards test (area-weighted seated average: 1.17 m/s, M: 1.22 m) to the closest matching upwards test without inverted blades (average: 0.77 m/s, M: 0.76 m). These are otherwise identical except for mount distance (and ceiling height). The area-weighted average air speeds differ by 52%.

[9]0.5 m/s is sufficient to maintain comfort while increasing temperature by approximately 2.5 °C above the upper comfort threshold for still air according to ASHRAE 55. Also, note here that we didn't attempt to maximize the achieved air speeds in the set of upwards tests and thus higher speeds were achieveable in many cases - e.g. fans didn't run at maximum speed.

[10]Note that thermal stratification was not a part of the study, thus we cannot evaluate how direction affects de-stratification.
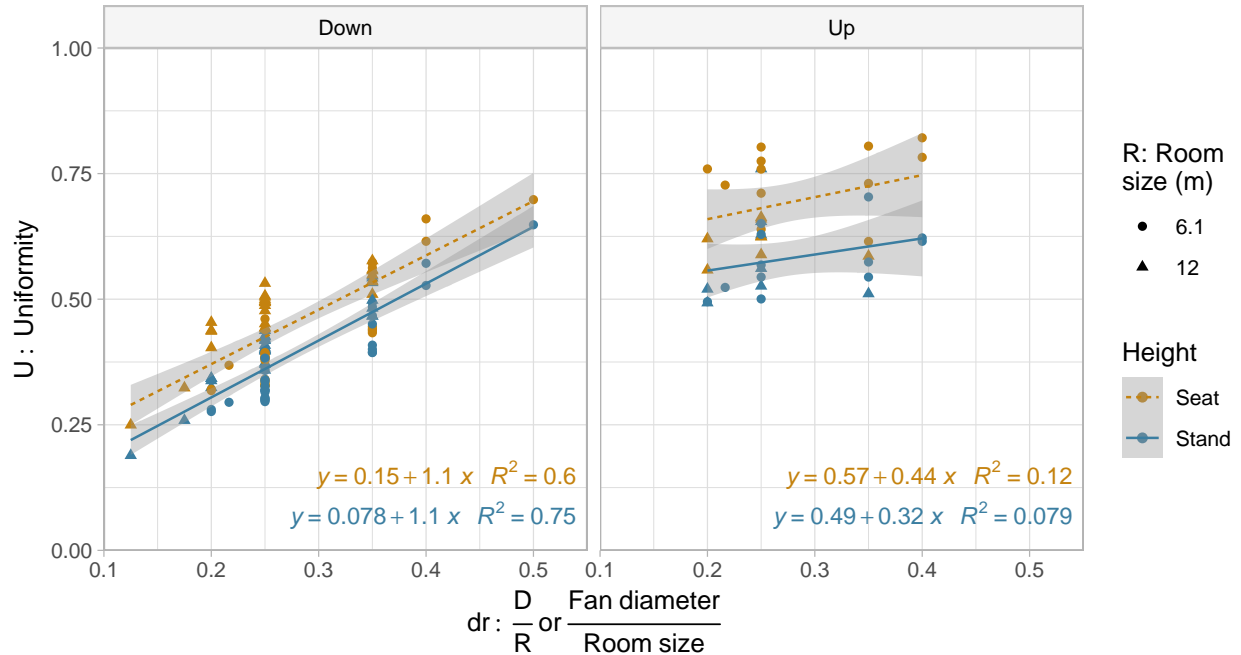
Figure 10: Uniformity air speed distribution for all tests (except still air conditions)

fan center to room corner. Similarly, it will differ when the fan is off center in the room and the room is not square (demonstrated in [33]). Additionally, many applications of ceiling fans use multiple fans in the same space. Thus, the simple models presented in this paper are at best a broad approximation. Though designers will often encounter scenarios that do not match the underlying simplifications, the models still provide useful information, particularly considering the absence of other guidance.

Until more information becomes available, for non-square cells/rooms (e.g. approximately rectangular), we suggest using the square root of the floor area to determine a representative value for the room size (R). Further research could test fans located off-center or in rooms of different aspect ratios, and that we could use that information to extend this paper's models. For multiple fans, in a prior exploratory experiment with multiple fans where all fans operated at the same speed, each identical 'cell' created by an individual fan had a similar air speed distribution as a comparable single fan case with the same size room. Further experiments could validate this and develop regressions to adjust the models if needed. Finally, it is important to note that [27] shows that furniture strongly affects the air speed distribution. Furniture layout isn't typically known at design stage and in any case changes often within the building lifetime. Given such a scenario, an appropriate approach may be to measure the aggregate effect that different types of furniture and layout have, and to use that data as a modifier to the models presented in this paper.

25

4. Conclusions

In this paper, we provide results and analysis from the largest study to date of air speeds generated by ceiling fans. There are several findings that are new contributions to the literature:

- We defined the concept of fan air speed as the rated fan airflow divided by the area swept by the blades. We show that normalizing the air speed at any point in the room against the fan air speed provides comparable profiles across a wide range of fan diameters and types.

- For a fixed set of fan and room characteristics, the measured air speed at any location is linearly proportional to the fan air speed, rotational speed, and airflow. This applies for fans blowing both upwards and downwards, regardless of fan type, though the relationship is less accurate at very low fan air speeds (< 1 m/s).

- We demonstrated that in the region outside of the fan blades, the seated and standing average air speeds increase proportionally with the ratio of fan diameter to room width.

- We quantified the spatial uniformity of the air speed distribution and showed that larger diameter to room ratios (and larger diameter fans) provide a more uniform environment.

- We showed that mount distance does not have a strong effect until it is < 0.2 times the fan diameter.

- For different fan types operating under otherwise similar conditions, we showed the air speed distribution is very similar in the region outside the fan blades. Air speeds may differ under the blades in some fan type comparisons, however, the effect on the air speed distribution is minor overall.

- We tested fans in reverse, blowing upwards towards the ceiling. This yielded a much more uniform air speed distribution than blowing downwards and has applications where having a homogenous air speed may be desirable. Though the air speeds are lower than for a comparable downward test, they are still high enough for an appreciable cooling effect. The upper quartile and maximum of the area weighted average air speeds for seated occupants for the upwards tests were 0.5 and 1.17 m/s respectively, indicating that it is feasible to select fans that will provide equivalent comfort conditions at substantially higher temperatures while blowing upwards and providing a more uniform air speed distribution. Upwards tests with larger diameter to room size ratios, higher fan rotational speeds, or inverted blades (symmetrical blade geometry to the downwards direction), provided higher air speeds.

- We developed dimensionless models that apply to the majority of practical ranges of fan and room sizes using readily available information. The inputs are: diameter, blade height, ceiling height, room size,

and fan air speed. The models predict the lowest, area-weighted average, and highest air speeds for a seated or standing occupant in the room, with a median absolute error of 0.03, 0.05 and 0.12 m/s respectively. Further work could focus on extending the model to address current limitations, such as developing modifiers for non-square rooms, multiple fans, and furniture.

Our hope is that this paper will allow people to better understand air speed distribution in rooms due to ceiling fans, and more easily select an appropriate fan for their application.

5. Acknowledgements

6. Declaration of interest

We performed these experiments in a fan manufacturer's test facility and one author (Fizer) is an employee of that organization, and thus we anonymously report fan type. All other authors declare no interest.

7. Appendix

7.1. Worked example

Clear examples are often useful and thus we provide one here, along with a spreadsheet tool contained in the supplementary material [48]. *What is the area-weighted average air speed for seated occupants in this scenario? A 5 m square room and 3.5 m ceiling (R= 5 m, C = 3.5 m), with a 2 m diameter ceiling fan at a blade height of 3 m from the floor (D = 2 m, H = 3 m). The fan has a rated airflow of 6 m³/s (Q = 6 m³/s) at 120 rpm, and is currently operating at 70 rpm.*

The linear model predicts the area-weighted average *so*:

$$so_{avg} = 0.99 * \frac{D}{R} - 0.06 * \frac{H}{D} + 0.11 * \frac{D}{1.7} + 0.024 + 0.250 = 0.68$$

To convert that to an actual air speed, we first calculate the fan air speed (SF) at the rated airflow.

$$SF_{rated} = \frac{4 * Q}{\pi * D^2} = 1.91 \ m/s$$

27

We then calculate the fan air speed at the operating rotational speed, assuming is is linear with airflow through zero[11]:

$$SF_{@70rpm} = SF_{rated} * \frac{70}{120} = 1.11 \ m/s$$

The estimated area-weighted average air speed for a seated occupant is:

$$SF_{@70rpm} * so_{avg} = 0.76 \ m/s$$

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