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Ceiling Fan and Thermal Comfort Assessment for Mixed-Mode Classrooms in Hawai‘i

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Ceiling Fan and Thermal Comfort Assessment for Mixed-Mode Classrooms in Hawai‘i

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1. Introduction

This report is written to summarize research on the impact of ceiling fans on thermal comfort experienced in the two mixed-mode FROG buildings on the UH Manoa campus. Both subjective and objective data were collected over a period of two years to comprehensively describe fan performance, fan control preferences, occupant responses to thermal conditions, and to compare user perceptions with comfort model predictions using measured conditions. Thermal comfort analyses were performed using ASHRAE 55 Standard [1] adaptive comfort and the predicted mean vote models and the percentage of people expressing satisfaction in the survey is compared to the level of satisfaction that would have been predicted by the models. The research found that at least in Hawaii's tropical climate, the ASHRAE models are inconsistent and sometimes poor predictors of thermal comfort.

1.1 The FROG Classrooms' Mixed-Mode Design

The two FROG classroom buildings on the University of Hawaii campus are designed to be mixed-mode with many window openings to provide natural ventilation but also an HVAC system for use when conditions demand. The HVAC thermostat set-point is 77ºF and is designed to deliver air at ~ 62ºF to avoid condensation on the supply air registers. The classrooms are also equipped with three sets of three ceiling fans, each set with a 7-speed controller. With these choices of using natural ventilation, ceiling fans and HVAC, the varying behaviors and preferences of the users result in very different operating conditions of the buildings throughout the weekly schedule and from semester to semester. Each classroom is used by the University Lab School (ULS) for 5^{th} to 8^{th} grade classes in the morning with only one instructor every morning and approximately 29 students. This is the most regular part of the schedule. The University of Hawaii instructors use the classrooms in the afternoons and evenings with an intermittent schedule and varying numbers of students. This report will describe how the buildings were operated: frequency of ceiling fan use, ceiling fans speeds, HVAC use, and whether windows were left open. The air velocities produced by different ceiling fans speeds was studied and the effect on thermal comfort is discussed. A survey asking occupants to vote on their level of satisfaction with the thermal conditions in the room was conducted in FROG1. The results of the survey are matched with the environmental conditions in the room preceding each vote. The percentage of people expressing satisfaction in the survey is compared to the level of satisfaction that would have been predicted by the ASHRAE 55 Standard [1] adaptive comfort and the predicted mean vote models.

2. Ceiling Fan and HVAC Use for the University Lab School

The FROGs have three sets of three ceiling fans that are metered separately: north set in the entryway, middle and south sets in the classroom (Figure 1) These are Energy Star-rated (Hampton Bay model #52860) 60-inch ceiling fans hung nine feet above the floor. The fan speed of each row of fans is controlled by a Lutron Maestro controller (canopy module CM-FQ1 and wall control MA-FQ4M) with seven speeds. The classroom ceiling is sloped from a height of 17 ft on the north side to 12 ft on the south side. The HVAC ducting is to the west side with the air handler in a utility closet and the compressor located outdoors on the west side. The University Lab School has a regular morning schedule (vs the sporadic afternoon/evening schedule of the UH classes) and was used in this analysis to capture the hours that are occupied regularly by the same number of people. Each classroom is managed by a single instructor all morning.

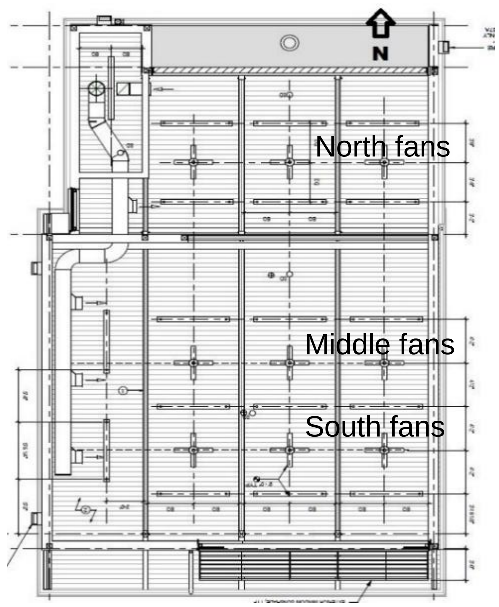

Figure 1. Three banks of fans in FROG buildings: north fans in entryway, middle and south fans in classroom.

Parameters and assumptions for the analysis were:

- Dates: August 8 to December 19, 2019 (see Appendix A for school calendar)
- Holidays: eleven were removed from dataset
- Days: Monday-Friday, Hours: 7am-Noon
- HVAC systems for both buildings were out-of-service until October 31, 2019, so early fall (Aug. 8-Oct 30) was analyzed separately from late fall (Oct 31-Dec 19).
- The middle bank of ceiling fans was used in the analysis because it is located within the classroom area; the threshold for “on” was > 0.009 kW (fan setting #1 = 0.01 kW)
- The HVAC air handler and compressor operate concurrently, but for simplicity, only the compressor was used in the analysis and the threshold for “on” was 0.03 kW.

2.1 Frequency of Ceiling Fan and HVAC Use

In FROG1, the ceiling fans were the only option for "cooling" while the AC was out-of-service in the early fall of 2019 (late August through October 30). They were used 96% of the time and the windows were open 93% of the time (Table 1). During late fall, after the AC was repaired, the ceiling fans were used 94% of the time, the AC was used 56% of the time and the AC and ceiling fans were on concurrently 55% of the time (in other words, the ceiling fans were nearly always in use when the AC was on). In the late fall the windows were open only 14% of the time and were rarely left open while the AC was running (2% of the time).

Table 1. The percent of time the ceiling fans, the HVAC, and both were on during early fall 2019 and late fall 2019 in FROG1.

	Early Fall (HVAC out-of-service) % of Total Time	Late Fall (HVAC repaired) % of Total Time
Ceiling fans on	96%	94%
HVAC on	0%	56%
HVAC and ceiling fans on	0%	55%
Windows open	93%	14%
Windows open with HVAC on	0%	2%

In FROG2, the ceiling fans were used 82% of the time in early fall while the HVAC was out of service and the windows were open 88% of the time (Table 2). In the late fall a similar usage of ceiling fans (85%) and open windows (86%) prevailed. The HVAC was used only 6% of the time (vs 56% for FROG1). This instructor chose the option of leaving at least one widow open while the HVAC was on.

Table 2. The percent of time the ceiling fans, the HVAC, and both were on during early fall 2019 and late fall 2019 in FROG2.

	Early Fall (HVAC out-of-service) % of Total Time	Late Fall (HVAC repaired) % of Total Time
Ceiling fans on	82%	85%
HVAC on	0%	6%
HVAC and ceiling fans on	0%	6%
Windows open	88%	86%
Windows open with HVAC on	0%	5%

2.2 Ceiling Fan Speeds Chosen by Instructors

Each bank of three ceiling fans has a 7-speed, wall-mounted manual control. The power consumption of the bank of three fans in relation to the settings on the controller are as follows: 1 = 0.01 kW, 2 = 0.02 kW, 3 = 0.05 kW,4 = 0.06 kW, 5 = 0.12 kW, 6 = 0.20 kW, and 7 = 0.34 kW. The relationship (with $P<0.0001$) between the fan control setting and power use is logarithmic (Figure 2) with the formula: Fan control setting = 1.71857*ln(Power (kW)) + 8.69781

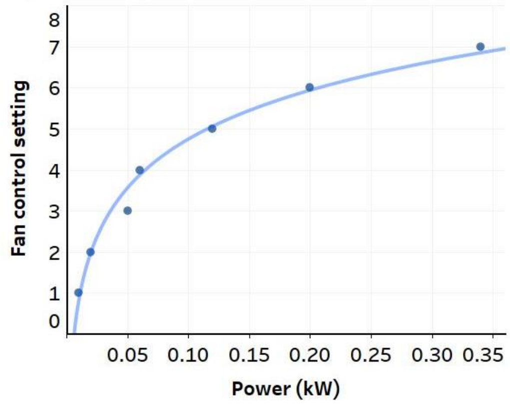

Figure 2. Logarithmic relationship between fan control setting and power use of one bank of three fans.

Anecdotally, we learned that the highest fan speed was too noisy for a teacher to easily hear a softly-spoken student. This is demonstrated by the data which show that the highest fan speed is rarely chosen. In FROG1 during the early fall when the HVAC was out-of-service, medium to high fan settings of #5 and #6 were chosen (Figure 3 top). During late fall, when the HVAC was repaired, fan setting #5 was almost exclusively chosen (Figure 3 bottom).

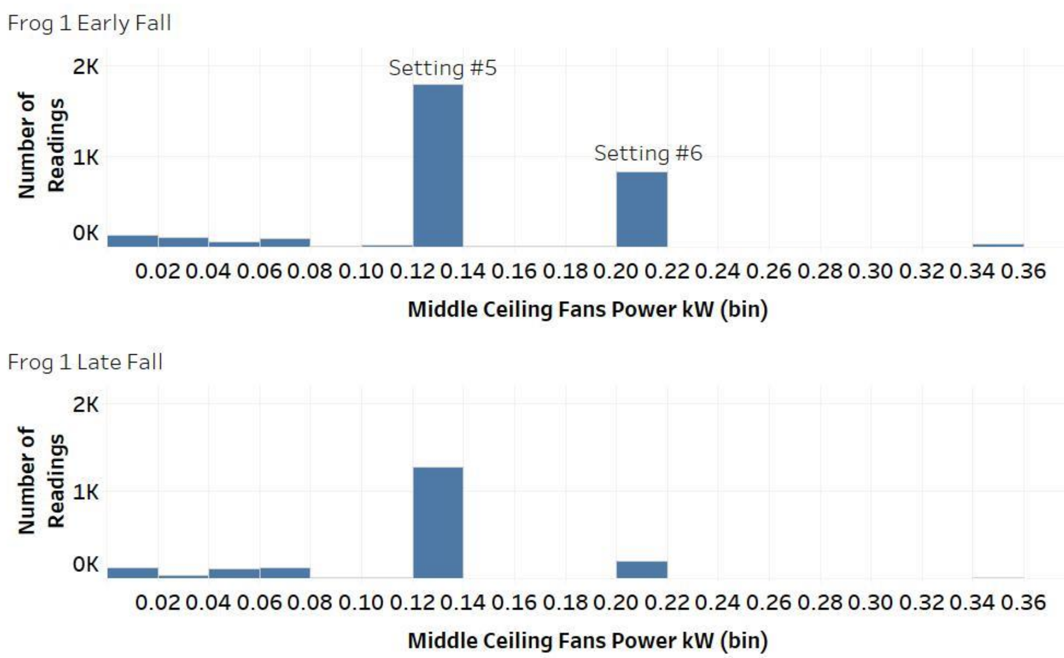

Figure 3. Ceiling fan control settings used on the middle fans in FROG1 during early fall (top) and late fall (bottom).

During early fall in FROG2 (Figure 4 top), setting #6 was clearly favored over setting #5, but in late fall the two settings were chosen at a similar frequency (Figure 4 bottom). The instructors in the two buildings had different preferences.

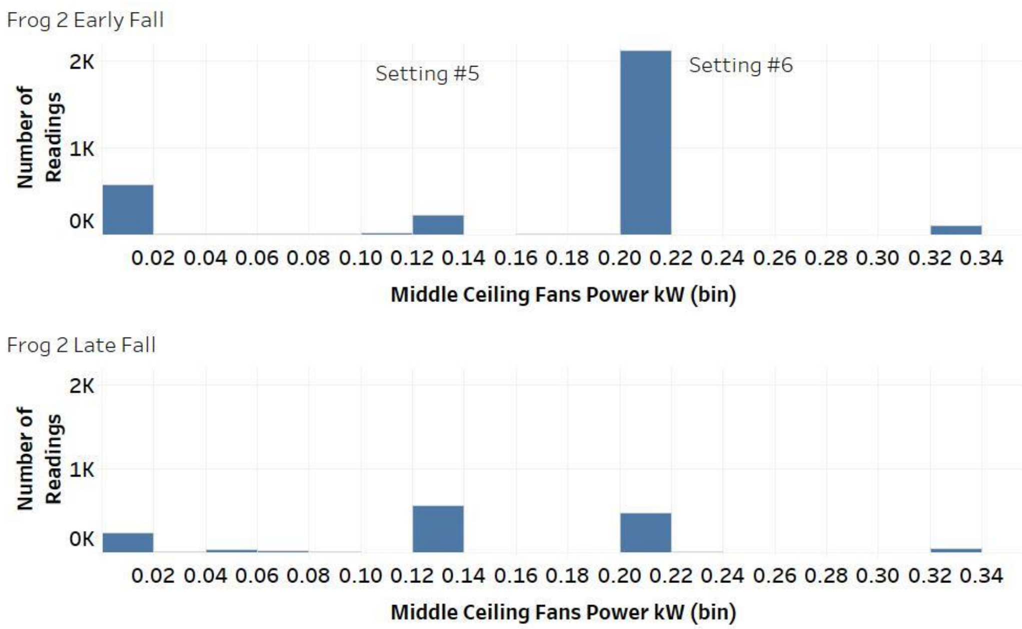

Figure 4. Ceiling fan control settings used on the middle fans in FROG2 during early fall (top) and late fall (bottom).

2.3 Open windows

In January 2019, teachers were given additional training in how to maintain indoor air quality. They were told to have either the widows open or the HVAC on to provide fresh air all times (i.e. to not allow a case where there is no fresh air introduced). They were told it was acceptable to leave some windows open while the HVAC was in use. There are contact sensors on the bank of nine windows facing south (Figure 5). The column of three windows closest on the east end are equipped with sensors. If any one of those windows is open, the data acquisition system registers the window status as “open”.

Figure 5. The bank of nine operable windows on the south side of the buildings. The three windows on the left side are wired with sensors to indicate if at least one window is open.

The instructor in FROG1 had the windows open 93% of the time in the early fall and 14% of the time in late fall. She rarely left the windows open when the HVAC was running (< 2% of time). The instructor in FROG2 had the windows open 88% of the time in the early fall and 86% of the time in the late fall. She did not use the HVAC often after it was repaired (6% of the time) and usually had at least one window open when she did use it.

3. Ceiling Fan Occupancy Controls

In an effort to prevent fans from being left on overnight or over the weekend, occupancy controls were installed on ceiling fans in FROG1 (the two banks of three fans within the classroom area). The controls were not compatible with the wireless Lutron control system of the fans and will need further research. See Appendix B for occupancy sensor specification and implementation. The controls were installed April 19, 2019 and removed April 9, 2020.

The hours the fans were left on either overnight or over a weekend were evaluated. The time period of the evaluation was January 1, 2017 to March 10, 2020, just over 3 years 2 months (3.2 yrs). Unlike the analysis in section 1.2, this analysis uses the total ceiling fan power (the sum of the three banks of fans). The three banks of fans are not necessarily on the same speed when left on overnight. The threshold for power of 0.03 kW for the total ceiling fan circuit was used to determine if fans were on.

The ceiling fans were left on overnight in FROG1 twice as often as in FROG: 2,355 hrs vs 1,209 hrs, respectively (Table 3). The fan speed and therefore the average power level was lower for FROG1 than for FROG2 at 0.109 kW vs 0.182 kW, respectively. See Figure 6 and Figure 7 for frequency distributions of power consumed by the ceiling fans overnight in each building. So rather than FROG1 using twice as much energy than FROG2, it used 32% more, 250 kWh vs 190 kWh, respectively (Table 3).

Table 3. Number of nights and weekends the ceiling fans were left on, the average power used, and the total energy used per year for the FROG buildings over 3.2 years of monitoring.

Metric	FROG1	FROG2
Number of nights fans left on	154 nights	84 nights
Number of these nights that were weekends	22 weekend nights	8 weekend nights
Total number of hours left on overnight/weekend	2,355 hrs	1,209 hrs
Total energy consumed overnights	250 kWh	190 kWh
Percent of time left on in the 3.2 years	8.4%	4.3%
Average power when left on	0.109 kW	0.182 kW
Average energy (kWh) per year from leaving on	78 kWh/yr	69 kWh/yr

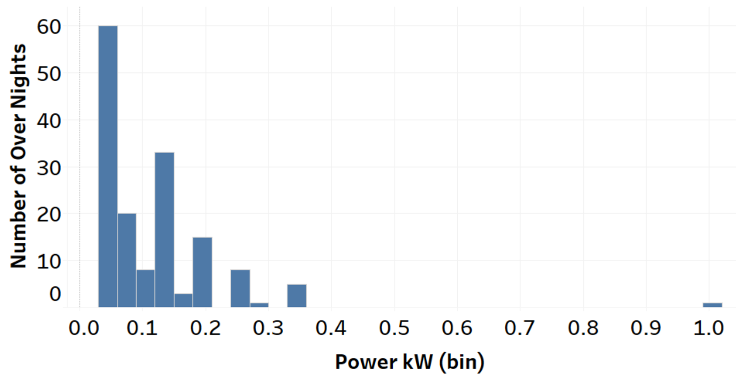

Figure 6. Frequency distribution of ceiling fan power (kW) in FROG1 when left on overnight.

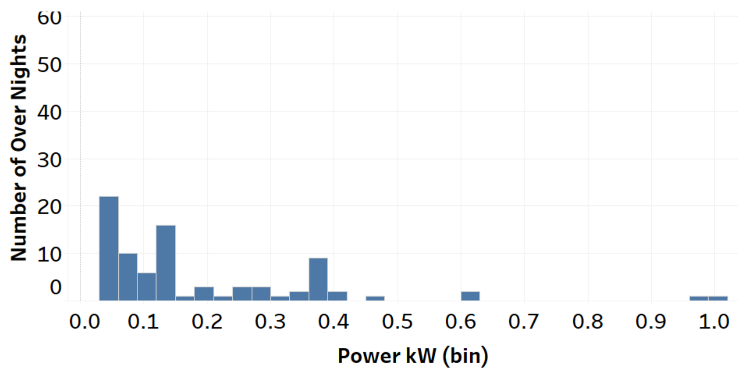

Figure 7 Frequency distribution of ceiling fan power (kW) in FROG2 when left in overnight.

4. Ceiling Fan Air Velocities

Each ceiling fan setting can create a range of air velocities throughout the room, depending on the specific location in the room. Air velocities in FROG1 were measured for three minutes at a height of four feet across a grid in the classroom at one fan speed (the 2020 Q2 report's Appendix F contained a detailed report). Air velocities measured for fan setting of #4 (medium) on the controller can be seen in Figure 8. For example, under a fan in location B3, velocities ranged from 40 fpm to 272 fpm, with an average velocity of 113 fpm. Between fans, such as location B4, velocities ranged from 23 fpm to 180 fpm, with an average velocity of 97 fpm. Velocities were lower in the perimeters of the classroom but were less likely to have occupants sitting in those areas. Also, row 1 was close to the front wall and is occupied by the instructor's desk and podium, so students were not seated in that area. Air velocity experienced by the occupants can be best characterized by the data within rows 2-6 and columns B-D as indicated in the red rectangle in Figure 8. The locations directly under the fans (dark blue in the figure) and between fans (light blue) are very distinct. The average velocity directly under a fan was 187 fpm and the average velocity between fans was 80 fpm.

Row		Column A	B	C	D	E
1	Avg (fpm)	80.5	229.0	77.4	175.0	75.5
	Stdev (fpm)	34.3	58.7	28.5	67.1	45.9
	Min (fpm)	14.1	58.7	25.9	45.0	23.0
	Max (fpm)	157.8	306.0	183.3	289.2	304.5
2	Avg (fpm)	50.0	83.6	86.3	70.3	78.1
	Stdev (fpm)	25.5	32.0	24.7	27.4	25.4
	Min (fpm)	5.4	23.4	22.0	27.7	20.6
	Max (fpm)	140.2	161.1	200.8	177.6	150.4
3	Avg (fpm)	51.1	201.2	72.3	170.1	61.2
	Stdev (fpm)	21.4	40.1	25.9	56.3	28.9
	Min (fpm)	0.0	113.3	31.7	63.1	10.4
	Max (fpm)	119.9	272.3	164.1	271.5	180.7
4	Avg (fpm)	66.6	97.2	82.1	81.9	72.9
	Stdev (fpm)	28.4	35.1	28.8	37.7	24.8
	Min (fpm)	19.0	22.9	34.3	28.4	20.7
	Max (fpm)	140.0	179.5	182.0	190.0	151.9
5	Avg (fpm)	37.8	193.0	67.3	185.5	39.9
	Stdev (fpm)	20.0	43.2	27.9	52.6	22.1
	Min (fpm)	8.0	61.1	19.1	48.6	4.0
	Max (fpm)	111.1	284.7	151.0	275.0	111.6
6	Avg (fpm)	29.9	25.5	33.2	20.4	26.1
	Stdev (fpm)	15.7	14.7	16.7	12.8	16.4
	Min (fpm)	0.8	0.0	2.8	0.0	0.0
	Max (fpm)	92.2	77.6	97.6	82.7	67.2

Figure 8. Statistics of air velocity measurements in 30 locations across a grid in the classroom for fan setting #4 (medium) on the controller. Row 1 is the "front" of the classroom on the east wall and column E is along the south wall of the classroom.

To estimate air velocities created by other fan speed settings, one location under a fan and one location between fans were measured and the statistical analyses of velocities can be seen in Figure 9.

Row		Fan setting 1	2	3	4	5	6	7
Under Fan	Avg (fpm)	42.2	107.6	193.3	201.2	301.7	377.7	510.3
	Stdev (fpm)	29.9	24.9	65.1	40.1	70.5	89.5	92.9
	Min (fpm)	0.0	21.1	27.1	113.3	74.6	113.1	244.7
	Max (fpm)	106.4	135.8	275.0	272.3	429.4	559.1	721.3
	Percentile 75th	65.8	124.2	245.4	230.5	351.6	443.7	577.9
Between Fans	Avg (fpm)	15.9	5.3	50.4	97.2	114.9	140.6	236.8
	Stdev (fpm)	15.2	6.0	26.9	35.1	40.2	53.1	69.1
	Min (fpm)	0.0	0.0	11.2	22.9	53.5	36.3	103.5
	Max (fpm)	65.5	23.1	129.7	179.5	236.8	272.5	452.5
	Percentile 75th	18.7	8.2	70.8	122.3	136.9	166.0	278.0

Figure 9. Statistics of air velocities (fpm) measured under a fan and between two fans at different fan settings.

The average air velocities (fpm) were plotted against the power use (kW) of a set of fans for the two locations in Figure 10. The polynomial equations will be used to predict the air flow in the room in these two general locations (under a fan and between two fans) when determining thermal comfort.

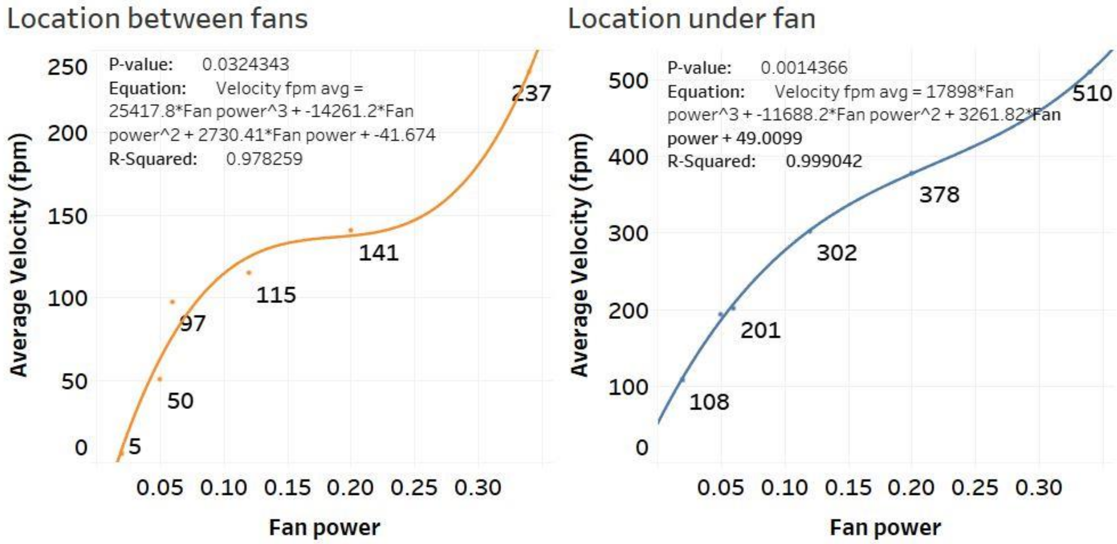

Figure 10. Plots of average air velocity (fpm) vs fan power (kW) for locations between two fans (left) and under a fan (right) with corresponding third degree polynomial equations, R-squared and P-values.

5. Thermal Comfort

> ***"comfort, thermal:*** that condition of mind that expresses satisfaction with the thermal environment and is assessed by subjective evaluation." – definition, ASHRAE 55 Standard

A kiosk (Figure 11) collecting thermal comfort survey responses from occupants in FROG1 was installed on August 26, 2019, and data were collected until March 30, 2020. The question on the survey was "How acceptable is the room temperature?" (Figure 12). The possible responses were on a 7-point Likert scale: highly acceptable, moderately acceptable, slightly acceptable, neutral, slightly unacceptable, moderately unacceptable, and highly unacceptable. Participation was voluntary and the instructions for participation encouraged the occupant to respond at the end of their class period, so they would be acclimated to the conditions. Only responses submitted after 1:00 PM (university users) were analyzed since we do not have Institutional Review Board approval for surveying middle school students (minors) who use the classrooms in the mornings.

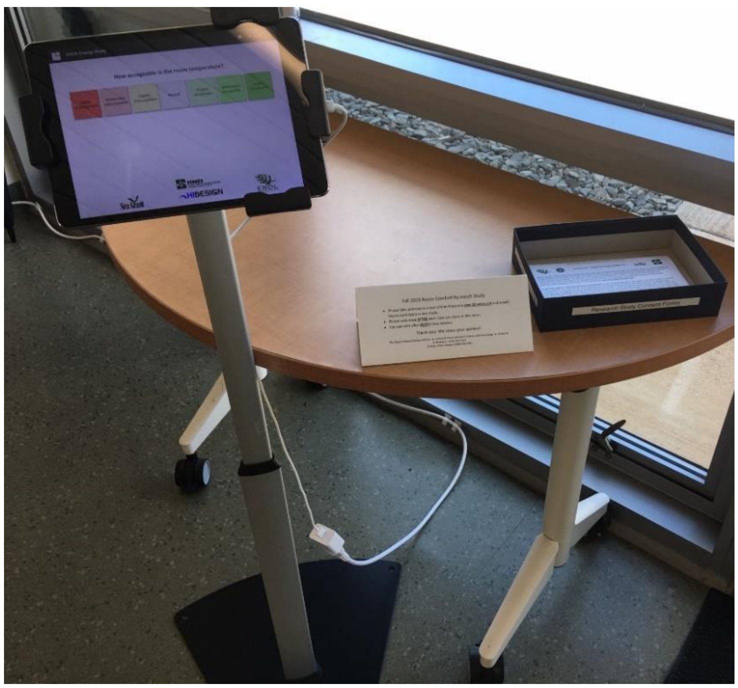

Figure 11. A kiosk made of a tablet on a stand was installed by the door of FROG1 with a sign giving instructions and a box of consent forms.

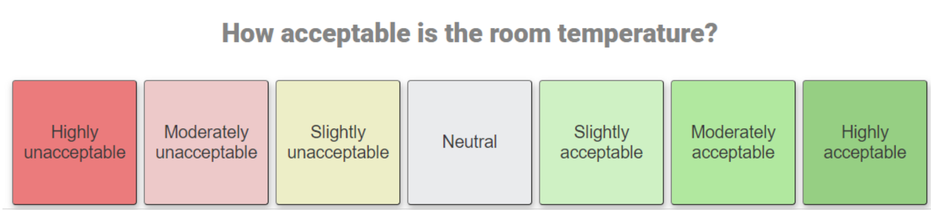

Figure 12. A screenshot of the question on the tablet. Pressing a box on the screen triggers a "Thank You" message and automatically logs the timestamped response. The screen re-sets to the question in 3 seconds.

A summary of all 407 responses is shown in Figure 13, with only 34% of the responses in the neutral to highly acceptable categories.

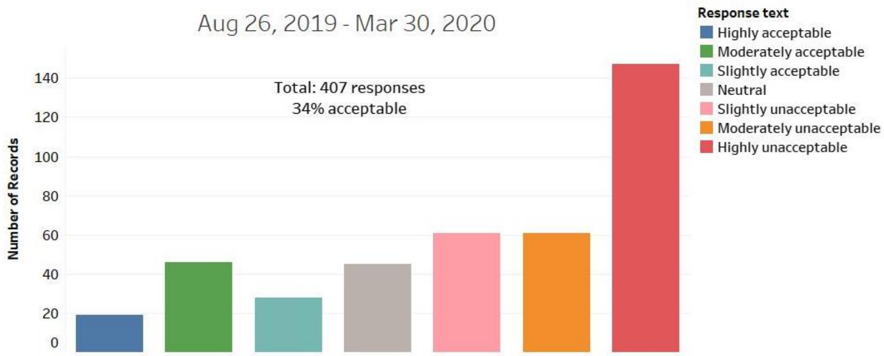

Figure 13. Number of responses in each category of acceptability of the room temperature for the duration of the study.

The HVAC was out-of-service until October 31, 2019. Figure 14 compares the responses from before the HVAC was repaired to the responses after it was repaired. The weather was also cooling off after October and the rate of "acceptable" responses was only 50%.

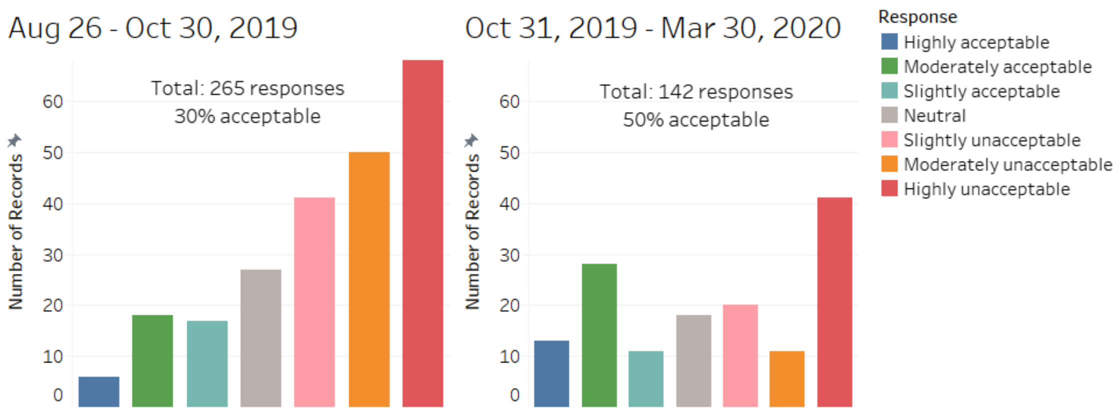

Figure 14. Number of responses in each category of acceptability of the room temperature before and after the HVAC was repaired.

The ranges of indoor relative humidity and operative temperature for the votes that were acceptable vs those that were unacceptable were quite similar, as can be seen in Figure 15. The survey did not have a follow-up question to ask about thermal sensation, so we cannot determine if some of the dissatisfied responses were from people feeling too cool. Limitations of the survey are described more fully in Appendix C.

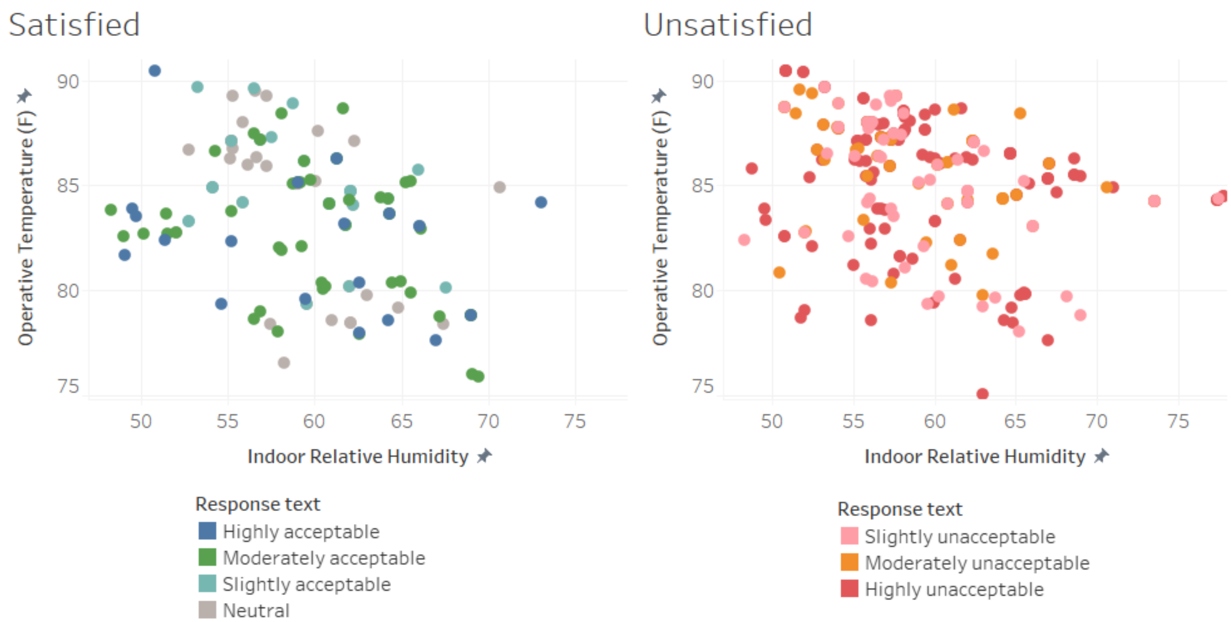

Figure 15. Indoor relative humidity (%) vs operative temperature (F) for each vote cast that was satisfied (left) vs unsatisfied (right).

> **"**The purpose of this standard is to specify the combinations of indoor thermal environmental factors and personal factors that will produce thermal environmental conditions acceptable to a majority of the occupants within the space**"** – ASHRAE 55-2017

The ASHRAE Standard 55-2017 for Thermal Environmental Conditions for Human Occupancy described two main thermal comfort models: (1) the adaptive comfort model was originally designed for spaces that are naturally ventilated and have no HVAC system installed; (2) the predicted mean vote (PMV) model, that was developed to predict thermal comfort in mechanically conditioned spaces. The FROG buildings are mixed-mode, so traditionally the PMV model would be used due to the air conditioning being present. Recently, researchers from the UC Berkeley Center for the Built Environment (CBE) and the University of Sydney [2], two of whom described the original adaptive comfort model in 1998, analyzed a new global database of thermal comfort measurements and concluded that the adaptive comfort model is more applicable for mixed-mode buildings than the PMV model. On Sept 1, 2020, Addendum f to the standard [3] was published, changing the applicability criteria from: "There is no mechanical cooling system (e.g., refrigerated air conditioning, radiant cooling, or desiccant cooling) installed. No or heating system is "in operation" to "There is no mechanical cooling system (e.g., refrigerated air conditioning, radiant cooling, or desiccant cooling) or heating system in operation." For this analysis of the thermal comfort survey, we compared the responses to what was predicted by the two models. Each survey response was matched with the average conditions in the classroom for the preceding 30 minutes to show the conditions the occupant was experiencing before responding.

Instrumentation used to measure indoor environmental conditions were Automated Logic temperature, humidity, and CO_2 sensors (models ZS-H-ALC and ZS-HC-ALC), and a 6-inch diameter globe made of copper and painted black matte with a temperature sensor (Kele ST-R24) inserted inside it. Outdoor weather was collected with a Gill weather station 1723-1B-2-111. Power data was measured with a PowerScout 24 meter equipped with Dent Instrument current transducers. Data was collected in 5-minute intervals using ERDL's software, Ionoa (https://github.com/erdl/lonoa). Thermal comfort survey responses were collected with a tablet kiosk installed by the door of the classroom. Survey data was acquired using ERDL's software, survey_display (https://github.com/erdl/survey_display).

5.1 Adaptive Comfort Model

The adaptive thermal comfort model of ASHRAE Standard 55-2017 defines the acceptable thermal conditions for occupant-controlled, naturally ventilated spaces under the following conditions: a) there is no HVAC system in operation; b) occupants have metabolic rates ranging from 1.0 to 1.5; c) occupants are free to adapt their clothing; and d) the prevailing outdoor temperature is greater than 50ºF and less than 92.3ºF. We analyzed the data using the adaptive comfort model graph in the following conditions (see Figure 16 for work flow) to make a comparison between the model prediction and the actual votes:

- All votes under all conditions plotted on the graph with the standard 80% acceptability limits for the upper and lower boundaries for the comfort zone, no filtering or adjustments for conditions.
- Votes filtered for when HVAC was off during the previous 30 minutes in order to analyze only the naturally ventilated conditions.
- Votes when HVAC was off displayed with adjustments to the operative temperature according to the estimated air speed based on middle fans power use. Since it is unknown where the occupant was sitting in the classroom, we will consider three versions making the assumption:
    - no elevated air speed included in the model;
    - air speed calculated for an occupant seated directly under a ceiling fan;
    - air speed calculated for an occupant seated between two ceiling fans.

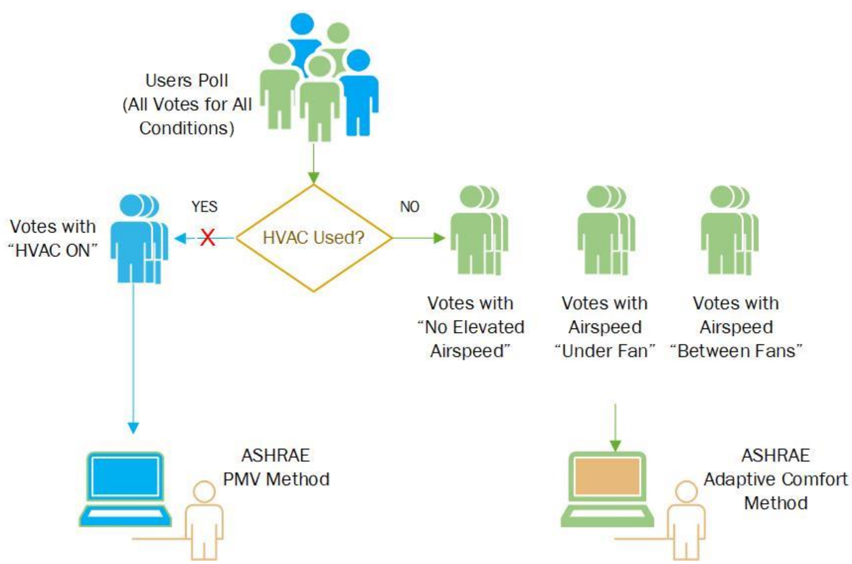

Figure 16. Work flow for data analyses using the ASHRAE 55 adaptive comfort model.

The graphic comfort zone method (Figure 17) of the model plots a graph of operative temperatures on the y-axis and prevailing outdoor air temperatures on the x-axis and defines comfort zone. It defines a comfort zone where 80% of respondents would find the operative temperature acceptable within upper and lower limit boundary lines set by the following calculations:

Upper limit in ^oF = 0.31(prevailing average outdoor temperature) + 60.5
Lower limit in ^oF = 0.31(prevailing average outdoor temperature) + 47.9

The operative temperature is the average of the mean radiant temperature (MRT) and the air temperature. The MRT was calculated from the temperature globe thermometer in the center of the room and the air temperature using the following formula (1) for a standard globe with forced convection in EN ISO 7726:2001.

$$MRT = [(t_g + 273)^4 + 2.5 \times 10^8 \times v_a 0.6 (t_g - t_a)]^{1/4} - 273 \quad (1)$$

Where t_g = globe temperature (^oC), v_a = air velocity (m/s), t_a = air temperature (^oC)

The standard defines the prevailing outdoor temperature as the average for the previous seven to 30 days. This analysis used a rolling 14-day average. Figure 17 shows an example of an Adaptive Comfort Model graph with the 90% acceptable range in dark blue and the 80% acceptable range which also includes the areas in light blue.

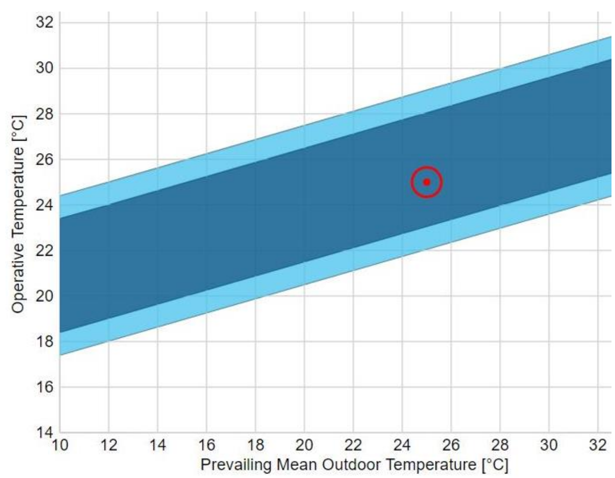

Figure 17. Center for the Built Environment online tool [4, 5] displays the thermal comfort range for the adaptive comfort model.

The thermal comfort survey was deployed on a tablet kiosk in FROG1 from August 26, 2019 to March 30, 2020. Votes we considered to be acceptably comfortable were these four response options: highly acceptable, moderately acceptable, slightly acceptable or neutral. Of the 407 total overall votes, 138 votes, or 34% were acceptably comfortable responses. When the responses for filtered for when the HVAC was off, 351 responses remained. When plotted on the adaptive comfort graph shown in Figure 18, the color of the point indicates the actual response (pink, orange, and red are not acceptable) and the position between the boundary lines would indicate that the conditions would be predicted by the model to be acceptable to 80% of respondents. Of the 162 votes where the conditions fell between the boundary lines, 47% of those votes were reported as acceptable, indicating that the model was not a good predictor of the responses.

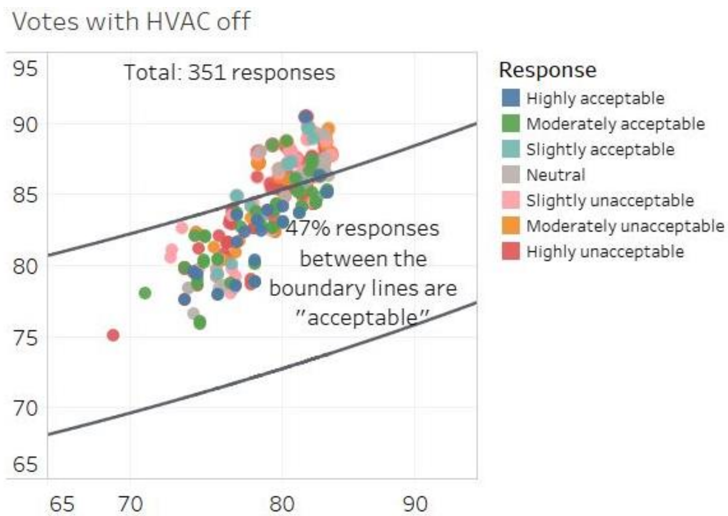

Figure 18. Thermal comfort votes plotted on the adaptive thermal comfort model's graph: operative temperature vs prevailing outdoor temperature. The color of the dot indicates the response category. Results from all responses are displayed on the left, responses when HVAC was turned off are displayed on the right.

The classroom has ceiling fans providing air speeds which affect thermal comfort.
Table 4 is drawn from Table 5.4.2.4 of the ASHRAE 55-2017 standard which shows the degrees Fahrenheit the upper limit operative temperature increases when the air speed is elevated above 59 fpm.

Table 4. Degrees Fahrenheit increases in the upper limit for the acceptable temperature in the adaptive comfort model with air speeds above 59 fpm.

Average air speed 118 fpm	Average air speed 177 fpm	Average air speed 236 fpm
2.2ºF	3.2ºF	4.0ºF

The online CBE Thermal Comfort Tool [4] [5] of the Center for the Built Environment illustrates how the adaptive comfort graph's upper limit line is raised with an elevated air speed (Figure 19) compared with the graph without elevated air speed (Figure 17). The degrees of elevation of this upper limit is dependent on the air speed between 118 fpm and 236 fpm for operative temperatures greater than 25ºC.

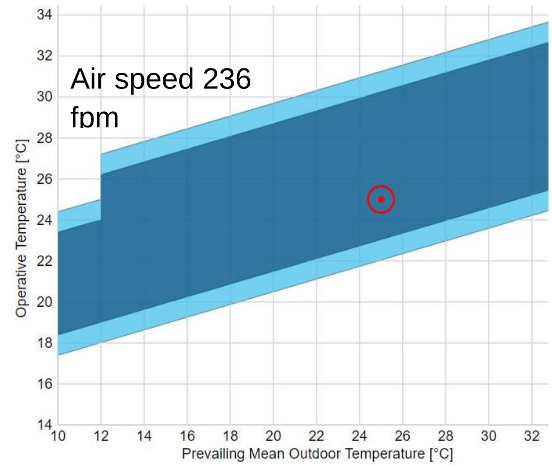

Figure 19. The CBE Thermal Comfort Tool for adaptive comfort upper and lower limits for 80% (light blue) and 90% (dark blue) acceptability with an elevated air speed of 236 fpm.

No guidance is provided in the ASHRAE 55 standard to adjust for air speeds below 118 fpm, therefore the CBE online Thermal Comfort Tool does not make any thermal comfort adjustments either. For this analysis, the same method is followed with no adjustment if the air speed is below 118 fpm. Intuitively, one would expect an upward shift in the upper limit of acceptability with elevated air speeds between 59 fpm and 118 fpm. That said, one researcher explored the impact of air speed values below 118 fpm in tropical Bangladesh, and found a 3.96ºF increase in the upper limit with an air speed of 59 fpm [6].

If a uniform elevated air speed was used for the entire time the survey was deployed, the upper boundary line would be raised as shown in Figure 20, similar to what the CBE online tool does (Figure 19).

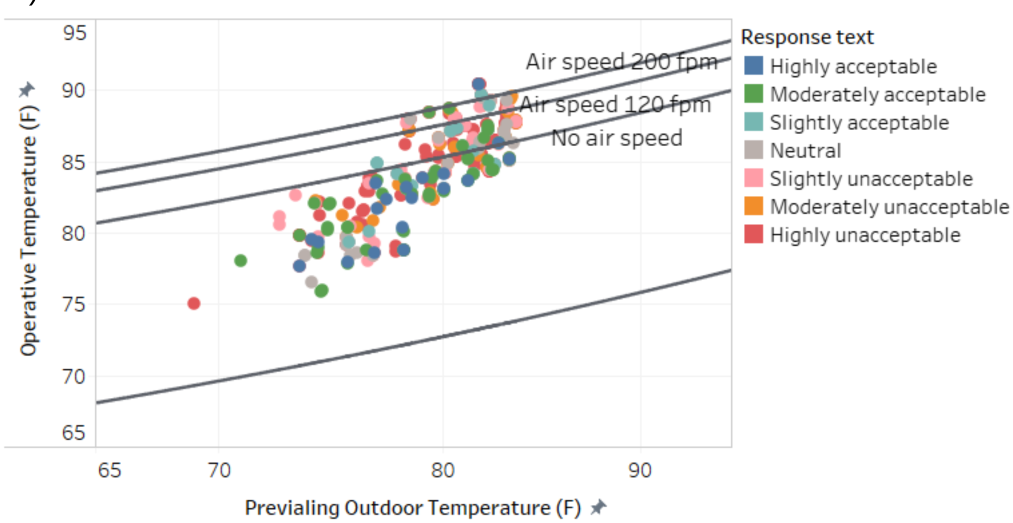

Figure 20. If uniform elevated air speeds of 120 fpm or 200 fpm existed in the classroom, the upper boundary line would be raised accordingly

For this analysis and display of results, the upper boundary cannot be raised accordingly to a uniform air speed because each vote is cast under different air speed conditions depending on whether the ceiling fans were on and which control setting was chosen. Instead of increasing the upper boundary line, we have adjusted the operative temperature by subtracting the appropriate degrees based on Table 4. The following simple linear regression (2) was used to interpolate temperature reductions for speeds that fall between air velocities listed in the table,:

$$y = 0.0153x + 0.04333 \quad (2)$$

y= number of degrees F to subtract from operative temperature
x= air speed (fpm) predicted from ceiling fan power of middle fans

For air speeds above 236 fpm, we used the maximum 4.0ºF. Air speeds were estimated for locations directly under a fan and between fans using the polynomial equations in Figure 10. Additional air movement from cross ventilation was not included in the estimation.

With operative temperatures adjusted down to account for elevated air speeds, the model predicted more instances when the operative temperature was within the 80% acceptability range. Originally 162 instances fell between the boundary lines (Figure 18), but for adjusted operative temperatures, 260 and 325 instances fell within this range for air speeds between fans and under a fan, respectively. The observed responses were 38% and 34% acceptable, respectively (Figure 21). Accounting for the elevated air speed, which was more realistic, resulted in the adaptive comfort model being a poor predictor of occupant comfort.

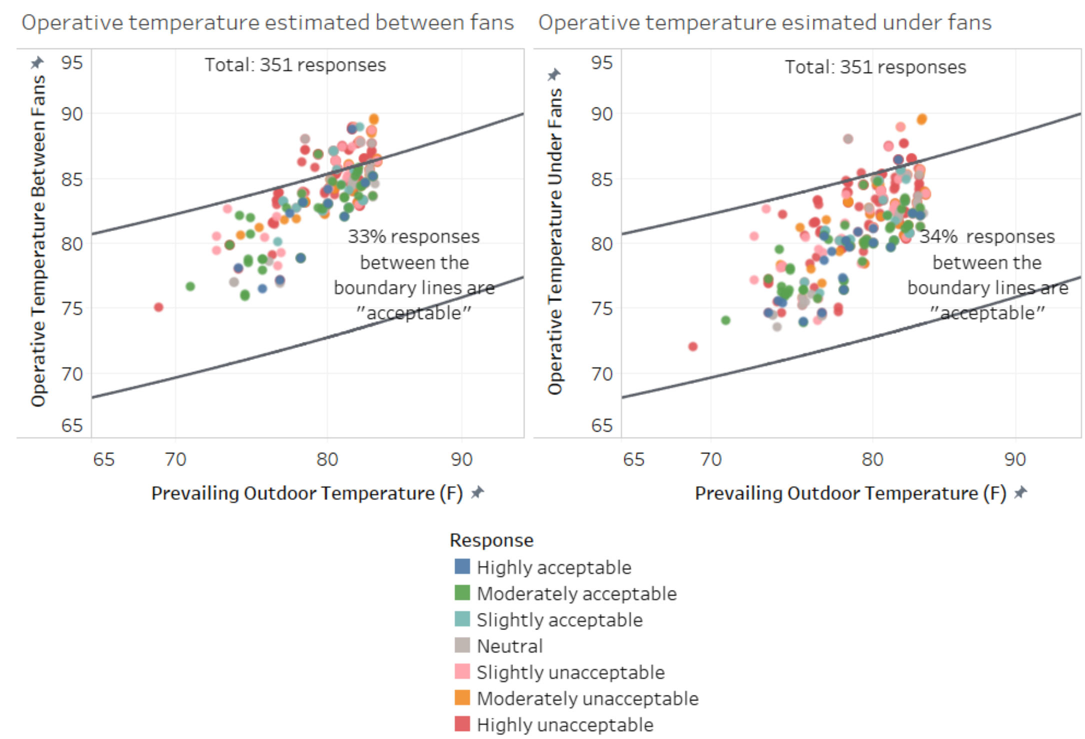

Figure 21. Thermal comfort votes when HVAC was off plotted on the adaptive thermal comfort model's graph: operative temperature vs prevailing outdoor temperature. The color of the dot indicates the response category. Results from assuming the participant sat between fans are displayed on the left, and results from assuming the participant sat directly under a fan are displayed on the right.

Adaptive Comfort Summary

The observed votes were different than what the adaptive comfort model predicted for times the classroom was naturally ventilated.

- When the model did not account for elevated air speeds provided by ceiling fans, the conditions that the model predicted would have 80% of occupants finding it acceptable, we observed only 47% of votes were acceptable (Table 5).
  When the model accounted for elevated air speeds estimated from ceiling fan power use, the conditions that the model predicted would have 80% of occupants finding it acceptable, we observed only 34-38% of votes were acceptable.

Table 5. ASHRAE 55 Standard adaptive comfort model predictions for thermal comfort vs the actual survey votes of occupants. Three possible air speed conditions were analyzed.

ASHRAE 55 adaptive comfort model results	Air speed assumption used in model		
	No elevated air speed	Air speed between fans	Air speed under a fan
✓ Complied with standard (80% acceptable)	Actual votes: N=162 47% Acceptable	Actual votes: N=260 38% Acceptable	Actual votes: N=325 34% Acceptable
✗ Did not comply (too hot)	Actual votes: N=189 19% Acceptable	Actual votes: N=91 13% Acceptable	Actual votes: N=26 8% Acceptable

5.2. Predicted Mean Vote Comfort Model

The predicted mean vote (PMV) model of the ASHRAE 55 Standard is applied if HVAC is being used in the space. The model is intended to predict the mean value of thermal sensation votes (see Table 6 for values of votes associated with different thermal sensations) of a large group of people. The acceptable range is: -0.5 < PMV < +0.5. Related to this range is an index of the predicted percent dissatisfied (PPD) of <10, or less than 10% of people experiencing the thermal conditions are predicted to report being dissatisfied. The Standard also states that “local discomfort effects are assumed to contribute an additional 10% PPD to the discomfort predicted by PMV, so that the total PPD expected in a building with PMV ±0.5 will be 20%”, in other words an 80% satisfaction rate.

Table 6. Values associated with thermal sensation survey responses used to formulate the predicted mean vote (PMV) model for ASHRAE 55 standard.

PMV vote value	Thermal sensation
-3	Cold
-2	Cool
-1	Slightly cool
0	Neutral
+1	Slightly warm
+2	Warm
+3	Hot

As explained in the section on the analysis for adaptive comfort, the conditions in the room during the 30 minutes prior to the survey response were averaged. The air speed was estimated using the power consumption of the middle row of ceiling fans. Three possible air speeds were tested: no elevated air speed, air speed assuming the occupant sat between two fans, and air speed assuming the occupant sat directly under a fan (using polynomials in Figure 10). The CBE online Thermal Comfort Tool has a feature that allows the user to upload a file of data to be analyzed as a batch (https://comfort.cbe.berkeley.edu/upload). The data file consisted of measured data for most parameters and assumptions for some parameters as described in Table 7. The CBE tool returns the row of data in the same order as the input file. Data from the survey was then appended to the PMV results from the tool for analysis.

Table 7. Data used for input into the CBE online tool to upload for batch analysis.

PMV Input Parameters	Data or Assumptions Used
Dry-bulb temperature	Measured data[1]
Relative humidity	Measured data[1]
Mean radiant temperature	Calculated from measured globe temperature[1, 2]
Clothing level	Assumption: 0.45 typical indoor summer clothing
Metabolic rate	Assumption: 1.0, seated, reading or writing
Air speed	Calculated from measured ceiling fan power[1]; three options: (1) no elevated air speed; (2) air speed if the occupant sat between two fans or (3) air speed if the occupant sat right under a fan.

[1] Average of 30 min previous to the response on the survey

[2] MRT formula for a standard globe from EN ISO 7726:2001, $MRT = [(t_g + 273)^4 + 2.5 \times 10^8 \times v_a 0.6(t_b - t_a)]^{1/4} - 273$

The PMV model is intended for spaces that have the HVAC operating, so data was filtered for when the air-conditioning compressor was using an average of 1 kW or more for the 30 minutes prior to the vote being cast. There were 37 survey votes cast under these conditions: 19 acceptable and 18 unacceptable. Thermal conditions that are found to be compliant with ASHRAE 55 using the PMV model would be expected to have an 80% acceptability rate for a group of people experiencing those conditions. It is unknown where the occupant was sitting in the room, so it cannot be determined what air speed they were experiencing at the time of their perceived sensation. A summary of the comparison between the PMV prediction, assuming three different air speed possibilities, and the actual votes:

- The PMV prediction for data with no elevated air speed resulted in 18 cases that would have been compliant with the Standard (Figure 23 and Table 8), but only 56% of the actual votes were acceptable (10 of the 18 responses).
- The PMV prediction using an elevated air speed for a location between two ceiling fans resulted in 27 cases that would be compliant (the elevated air speed allowed more unacceptably hot cases move to acceptable and a few acceptable move to unacceptably cold (Table 8); and only 48% of the actual votes were acceptable (13 out of 27).
- The PMV prediction using an elevated air speed for a location directly under a fan resulted in only six cases when the conditions would have been compliant due to the rest being categorized as unacceptably cool (Table 8), in other words the PMV was below -0.5. The actual votes that were acceptable were 67%, or four out of the six votes.

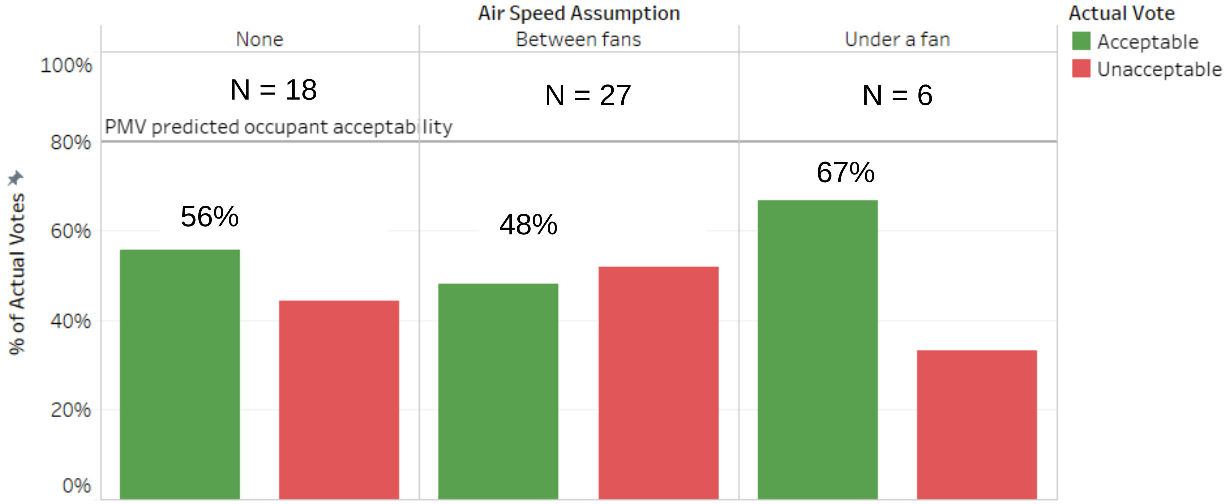

Figure 22. Actual survey votes for conditions that complied with ASHRAE 55 PMV model which are expected to result in 80% occupant acceptability. The N is the number of votes cast during conditions that complied with the standard.

Table 8. ASHRAE 55 Standard PMV model predictions for thermal comfort vs the actual survey votes of occupants. The PMV predictions were estimated for three possible air speed conditions: no elevation in air speed, air speed for a seat located between fans, and air speed for a seat located under a ceiling fan.

ASHRAE 55 PMV model results	Air speed assumption used in model					
	No elevated air speed		Air speed between fans		Air speed under a fan	
	Actual votes		Actual votes		Actual votes	
	Acceptable	Unacceptable	Acceptable	Unacceptable	Acceptable	Unacceptable
✔ Complied with standard	10	8	13	14	4	2
✗ Did not comply (too hot)	9	10	0	0	0	0
✗ Did not comply (too cool)	0	0	6	4	15	16

The PMV model was not a good predictor of actual votes for the acceptability of the room temperature in FROG1. By ASHRAE definition, conditions that comply with the model are expected to have an 80% satisfaction rate and we observed that only 48% to 67% of occupants found the conditions acceptable.

5.3 Summary and Conclusions

Thermal Comfort Survey Results Compared to ASHRAE 55 Standard Model Predictions

The indoor conditions in FROG1 were evaluated with the two thermal comfort models in ASHRAE 55-2017 Standard for Thermal Environmental Conditions for Human Occupancy, PMV and adaptive comfort models, to determine whether the actual conditions complied with the models. The theoretical results from the models were compared with actual survey responses of the classroom occupants. Three airspeed assumptions were tested in the models: no elevated air speed; air speeds estimated for seat locations between ceiling fans; and directly under a ceiling fan, calculated based on measured ceiling fan power. The adaptive comfort model was used when the classroom was naturally-ventilated (351 votes were cast during these times) and the PMV model was used when the air-conditioning was ON (37 votes were cast). When the conditions comply with each model, an estimated 80% of occupants would find the conditions acceptable. A kiosk was placed in the classroom with a survey question asking for the acceptability of the room temperature. The responses were matched with the room conditions and power data for HVAC and ceiling fans.

Responses were filtered for times when the conditions complied with the models (Table 9). The adaptive comfort model was a poor predictor of thermal comfort. Actual responses were only 34% to 47% acceptable, depending on which air speed assumption was used in the model. The PMV model was a slightly better predictor than the adaptive comfort model. When the between-fan air speed was assumed in the model, 27 sets of conditions complied with the PMV model, but only 48% of those actual votes considered the conditions acceptable. When an air speed for a location directly under a fan was used in the PMV model, only six sets of conditions complied while the others had predicted thermal sensations that were too cool.

Table 9. Summary results comparing thermal comfort model predictions to actual votes (N= number of votes cast when conditions complied with model).

	Air speed assumption used in model		
Model prediction	No elevated air speed	Air speed between fans	Air speed under a fan
Adaptive comfort 80% acceptable	Actual votes: N=162 47% acceptable	Actual votes: N=260 38% acceptable	Actual votes: N=325 34% acceptable
PMV 80% acceptable	Actual votes: N=18 56% acceptable	Actual votes: N=27 48% acceptable	Actual votes: N=6 67% acceptable

Neither ASHRAE 55-2017 model was a good predictor of survey responses. Cheung et al. 2019 analyzed the accuracy of the PMV-PPD model using over 50,000 responses from the ASHRAE Global Thermal Comfort Database II [7] and found the model to be inaccurate, especially at the extreme ends of the thermal sensation scale (cold and hot). See Appendix D for summary notes on their findings.

It appears that survey respondents did not find the temperature as acceptable as the models predicted. Although the thermal sensation question (How do you feel? With response options of cold, cool, slightly cool, neutral, slightly warm, warm, hot) was not included in the survey, we assume most of the unacceptable responses would have been warm or hot. It is unknown what thermal conditions they are accustomed to when they are not in this classroom. It has been demonstrated that people who are acclimated to air-conditioning (>10 hrs/day) feel hotter in warm conditions than people who are used to naturally ventilated conditions (<2 hrs/day of air-conditioning) [8] If we want to further investigate thermal comfort responses to naturally ventilated spaces on the university campus, we could re-examine survey data from two previous studies that collected data from more comprehensive surveys. One study took place in an open office in the Sinclair Library building and the other was conducted in a controlled experimental chamber in the Hawaiʻi Institute of Geophysics (HIG) building.

6. References

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7. Appendices

Appendix A. University Lab School Calendar

University Laboratory School

2019-2020 OFFICIAL SCHOOL CALENDAR

Teachers' Work Year - 1st Semester: July 30, 2019 to January 3, 2020; 2nd Semester: January 6, 2020 to May 22, 2020

Students' Work Year - 1st Semester: August 8, 2019 to December 19, 2019; 2nd Semester: January 7, 2020 to May 21, 2020

Week	Student days	Teacher days		S	M	T	W	T	F	S	
			2019								1st SEMESTER - 85 Student Days
1	0	4	August	28	29	30	31	1	2	3	July 30: First day for teachers
2	2	5		4	5	6	7	8	9	10	July 30 - August 7: Teacher Work days (no students)
3	4	4		11	12	13	14	15	16	17	August 8: First day for students
4	5	5		18	19	20	21	22	23	24	August 16: Statehood Day- Holiday
5	5	5		25	26	27	28	29	30	31	
6	4	4	September	1	2	3	4	5	6	7	Sept. 2: Labor Day- Holiday
7	5	5		8	9	10	11	12	13	14	
8	4	5		15	16	17	18	19	20	21	Sept. 16: School In-Service (no students)
9	5	5		22	23	24	25	26	27	28	
10	5	5	October	29	30	1	2	3	4	5	
11	0	0		6	7	8	9	10	11	12	Oct. 7-11: Fall Break
12	5	5		13	14	15	16	17	18	19	
13	5	5		20	21	22	23	24	25	26	Oct. 21-24: K-5 Parent-Teacher Conferences, 12pm dismissal
14	5	5		27	28	29	30	31	1	2	
15	5	5	November	3	4	5	6	7	8	9	
16	4	4		10	11	12	13	14	15	16	Nov. 11: Veterans Day- Holiday
17	5	5		17	18	19	20	21	22	23	Nov. 28: Thanksgiving; Nov. 29: School Holiday
18	3	3		24	25	26	27	28	29	30	Dec. 19: 1st Semester Ends
19	5	5	December	1	2	3	4	5	6	7	Dec. 19: Last student day 1st sem; 11:30 a.m. dismissal
20	5	5		8	9	10	11	12	13	14	Dec. 20: Teacher work day (no students)
21	4	5		15	16	17	18	19	20	21	Dec. 23: Jan. 3 - Winter Break
22	0	0		22	23	24	25	26	27	28	Dec. 25: Christmas; Jan. 1: New Year's
23	0	0	2020	29	30	31	1	2	3	4	2nd SEMESTER - 87 Student Days
24	4	5	January	5	6	7	8	9	10	11	Jan. 6: Teacher work day (no students)
25	5	5		12	13	14	15	16	17	18	Jan. 7: Start of Spring Semester for students
26	4	4		19	20	21	22	23	24	25	Jan. 20: Martin Luther King Day- Holiday
27	5	5		26	27	28	29	30	31	1	
28	5	5	February	2	3	4	5	6	7	8	
29	4	5		9	10	11	12	13	14	15	TBD: Institute Day (no students)
30	4	4		16	17	18	19	20	21	22	Feb. 17: Presidents' Day- Holiday
31	5	5		23	24	25	26	27	28	29	
32	5	5	March	1	2	3	4	5	6	7	
33	5	5		8	9	10	11	12	13	14	
34	0	0		15	16	17	18	19	20	21	March 16-20: Spring Break
35	4	4		22	23	24	25	26	27	28	March 23, 24, 25, 27: K-5 Parent-Teacher Conferences, 12pm dismissal
36	5	5	April	29	30	31	1	2	3	4	March 26: Kuhio Day Observed- Holiday
37	4	4		5	6	7	8	9	10	11	April 10: Good Friday- Holiday
38	5	5		12	13	14	15	16	17	18	
39	4	5		19	20	21	22	23	24	25	April 20: School In-Service (no students)
40	5	5	May	26	27	28	29	30	1	2	May 21: 2nd Semester Ends
41	5	5		3	4	5	6	7	8	9	May 21: Last day for students; 11:30 a.m. dismissal
42	5	5		10	11	12	13	14	15	16	May 22: Teacher Work Day (no students)
43	4	5		17	18	19	20	21	22	23	May 22: Graduation 5:30pm
	172	185		24	25	26	27	28	29	30	May 25: Memorial Day- Holiday

Teacher Work Day - no students

OFFICIAL STATE HOLIDAYS – 2019–2020 SCHOOL YEAR

Holiday	Day Observed	Holiday	Day Observed
Independence Day	July 4, 2019	New Year's Day	January 1, 2020
Statehood Day	August 16, 2019	Dr. Martin Luther King Jr. Day	January 20, 2020
Labor Day	September 2, 2019	Presidents' Day	February 17, 2020
Veterans' Day	November 11, 2019	Prince Jonah Kuhio Kalanianaole Day	March 26, 2020
Thanksgiving Day	November 28, 2019	Good Friday	April 10, 2020
Christmas Day	December 25, 2019	Memorial Day	May 25, 2020

Appendix B. Specification Sheet for Occupancy Sensors and Implementation

LEVITON.

Self-Contained PIR Ceiling Mount Occupancy Sensor

ODCOS-I

ODCOS-I1 • ODCOS-I7 • ODCOS-I2

BASIC OPERATION

The ODCOS-I uses passive infrared (PIR) detection technology to monitor a room for occupancy through a segmented Fresnel lens. This specialized lens divides the field-of-view into sensor zones. When a person passes into or out of a sensor zone, the sensor detects motion and switches its lighting loads ON. The lights will remain ON as long as there is an occupant moving through the sensor zones.

APPLICATIONS

The Self-Contained Passive Infrared Ceiling Occupancy Sensor is a cost-effective choice for commercial and institutional installations, where installation of the recessed ceiling unit is difficult, inconvenient or costly. Available in 120V, 220V and 277V versions, the ODCOS-I is ideal for:

- Storage areas
- Small bathrooms
- Retrofit
- Copy rooms
- Utility closets
- Small spaces without wall switches.

The Self-Contained Ceiling Sensor does not require an external control unit for power or switching the load ON and OFF.

FEATURES

- Sensor and switching relay in one unit—reduces labor and need for additional materials
- 360° field-of-view with approximately 530 sq. ft. of coverage when mounted at 8 ft. This reduces the number of additional sensors typically required in many spaces.
- Adjustable Delayed-OFF time setting between 20 seconds and 15 minutes allows custom adjustment for maximum savings
- Light Sensor - an ambient light override option can be set between 2 and 500+ foot candles and full brightness to prevent the sensor from switching lights ON when ample natural sunlight is available. Hold-OFF feature
- Segmented Fresnel lens contains 79 segments for optimum sensitivity and detection performance
- A standard A/C toggle switch may be used to provide manual-OFF override so that lights may be switched OFF
- Red LED indicator light flashes when sensor detects motion, to verify power placement and function of sensor at installation

LED READY

Leviton Manufacturing Co., Inc. Global Headquarters
201 North Service Road, Melville, NY 11747-3138 **tech line** 800-824-3005 **fax** 800-832-9538

ODC0S-I1 • ODC0S-I7 • ODC0S-I2

PRODUCT DATA

LEVITON.

INSTALLATION

The ideal location for the Self-Contained Occupancy Sensor unit is in a ceiling area that provides a full view of the space with an unobstructed path to the entrance way(s), but out of line from hallway traffic. The sensor should be positioned at least 6 feet from HVAC registers to prevent false triggering. The ODC0S-I may be mounted directly to a three or four-inch octagon box. The sensor wires directly to the lighting fixtures. An ODCCG protective cage is recommended to guard against accidental breakage.

SPECIFICATIONS

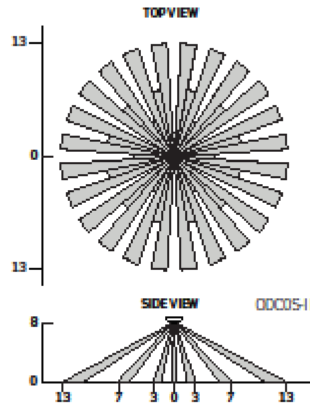
ELECTRICAL	-I1W	-I2W	-I7W
Line Voltage	120V	220V	277V
Power Consumption	1.45W	1.31W	1.89W
Operational Frequency	60Hz	50Hz	60Hz
Load Rating Incandescent	1000W	1000W	1000W
Load Rating Fluorescent	1000VA	500VA	2700VA
Load Rating Motor	120V 1HP	_	_
Wire Designation	Line - black, Neutral - white, Load - blue		
ENVIRONMENTAL			
Operating Temperature	32°F - 122°F (0°C - 50°C)		
Storage Temperature	14°F - 185°F (-10°C - 85°C)		
Relative Humidity	20-90% Non-condensing		
OTHER			
Listings	UL Listed and CSA Certified, can be used to comply with 2016 Title 24, Part 6 occupancy sensing requirements, complies with FCC regulations		
Warranty	Limited Five-Year Warranty		

ORDERING INFORMATION

CAT. NO.	DESCRIPTION
ODC0S-I1 ODC0S-I2 ODC0S-I7	Self-Contained Ceiling Mounted Occupancy Sensor & Switching Relay
ODCCG	Protective Cage

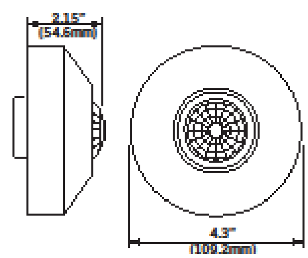
* To indicate color, add suffix to the end of hte catalog number. White (-W)

FIELD-OF-VIEW

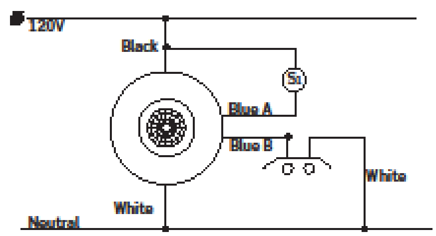


ODC0S-I Field-of-View (in feet)

DIMENSIONAL DIAGRAM



WIRING DIAGRAMS



ODC0S-I1 Wiring Diagram with optional switch for override to OFF
* Same wiring for all voltages.

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Visit our Website at: www.leviton.com/sensors

G-8334B/E18-aa
REV MAY 2018

Lianne Rozzelle

EE396 - Junior Project

May 22, 2019

Utilizing Occupancy Sensors to Aid in Future Energy Efficiency

Is it possible to retrofit the ceiling fans at the FROG (Flexible Response to Ongoing Growth) buildings at the College of Education to incorporate occupancy sensors? This was the question I was tasked to tackle this semester for my Junior Project. The reason for adding sensors to the system was to address the fact that there have been moments when the fans have been left on when the buildings are not in use, resulting in energy being wasted. The University of Hawai`i system has a goal to be energy independent by the year 2035. By incorporating occupancy sensors in rooms, lighting and fans accidentally left on would automatically turn off (after a specified amount of time) when they are no longer occupied, therefore decreasing the amount of energy used.

To begin the process of solving the problem, I broke down the project into four parts. The first being to determine how the fans were initially wired up. The original set-up in the FROG building has wires from the power distribution panel running to a wall switch (wires entering into the top of the junction box), which is then hardwired to a bank of three fans on the ceiling (wires leaving from the bottom of the junction box) – refer to Figure 1 for wiring schematic. The wall switch uses IR technology to communicate with a canopy module to control the speed on the fans. However, since occupancy sensors work like a switch, it was determined that all that would I need to do was put the sensor in-line with the switch and fan to cut out the power when the room was longer occupied.

The second task was to find an occupancy sensor that would be compatible with the system. In efforts to make this as easy as possible, I decided to find a sensor that could work with the 120 VAC power coming from the distribution panel and could also handle the current draw from the bank of three fans (~0.35 kW). I was able to find one sensor that satisfied both requirements (Leviton ODC0S-I1W). The Leviton sensor can handle a load rating up to 1000 W.

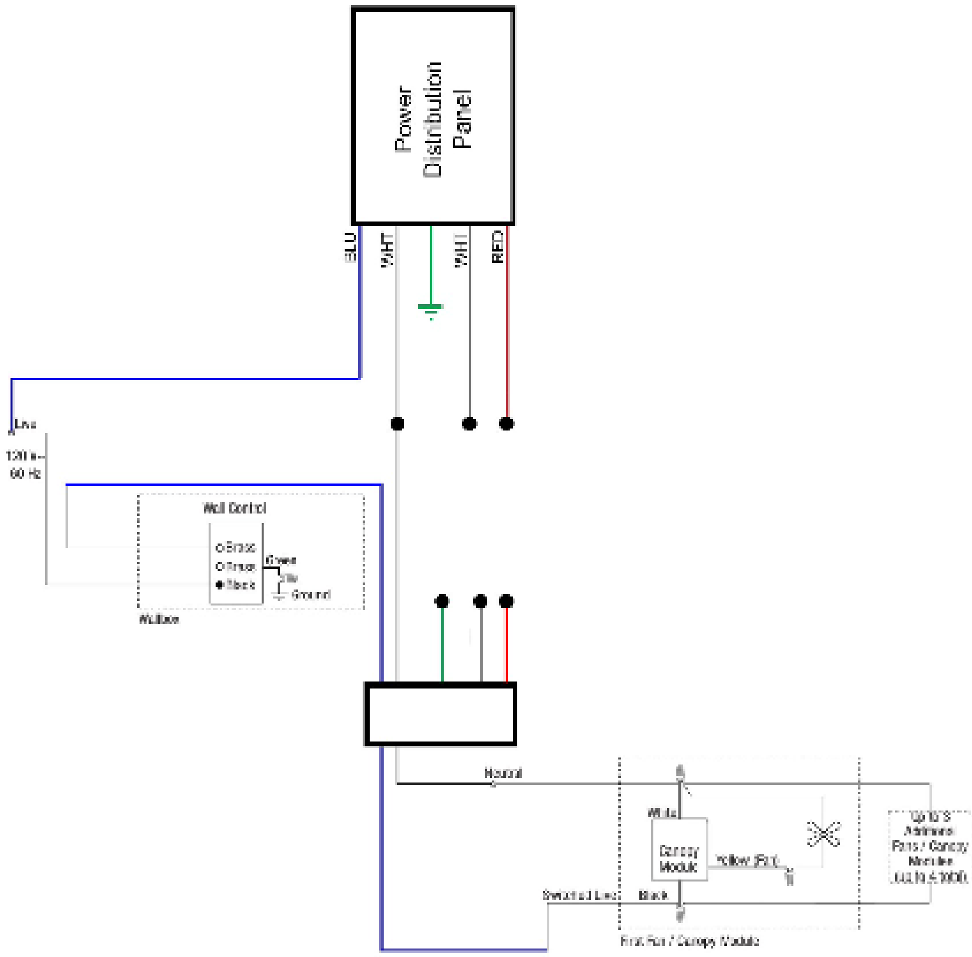

Figure 1 – Original wiring schematic for one set of fans. The other set of fans used the adjacent RED/WHT wires.

The third task was to test the configuration on a mock-up to verify correct wiring prior to installing it in the FROG building. The mock-up consisted of a spare wall switch, canopy module, and fan motor to use with the Leviton sensor. I was able to wire everything together and powered it using 120 VAC from a wall receptacle. I tried connecting it together in two ways. The first with the occupancy sensor in-line before the wall switch and the second with the sensor in-line after the wall switch. To give some background information, the wall switch (Lutron MA-FQ4FM) has a set of LED lights on it. The lights cycle when the switch is initally turned on and then it will stop at one LED to identify the current speed that the fans are set at. When I installed the occupancy sensor before the wall switch, I noticed that when the sensor switched from "occupied" to "vacant," all the LED lights would turn off. When the sensor switched from "vacant" to "occupied," the single LED identifying the fan speed would turn back on (along with the fan motor). When the sensor was installed after the wall switch, the LED light identifying the fan speed would still stay on even if the sensor was in "vacant" mode. I thought connecting it in this set-up would be a more accurate description of what was happening when the occupancy sensor switches from on to off: the wall switch is still in the "on" position, even though the fans are off because of the "vacant" room.

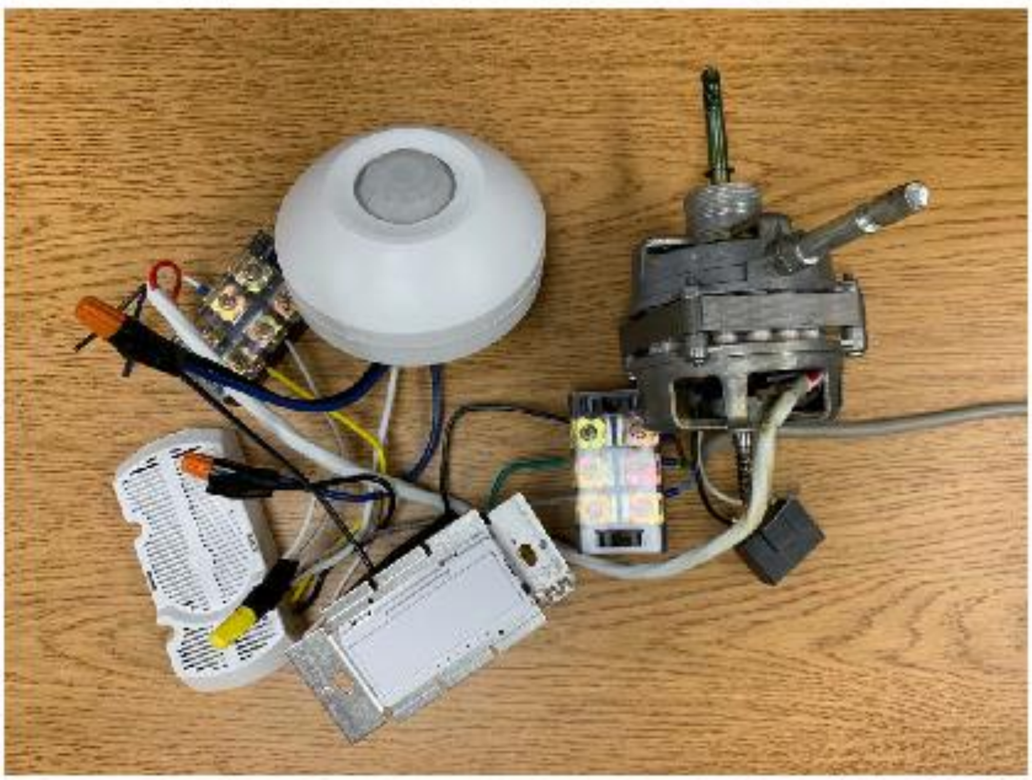

Figure 2 – Wall switch, occupancy sensor, canopy module, and fan motor connected together for mock-up.

The fourth task was to implement the final design (Figure 3) and permanently install the occupancy sensors into one of the FROG buildings. The installation went smoothly and everything tested satisfactory.

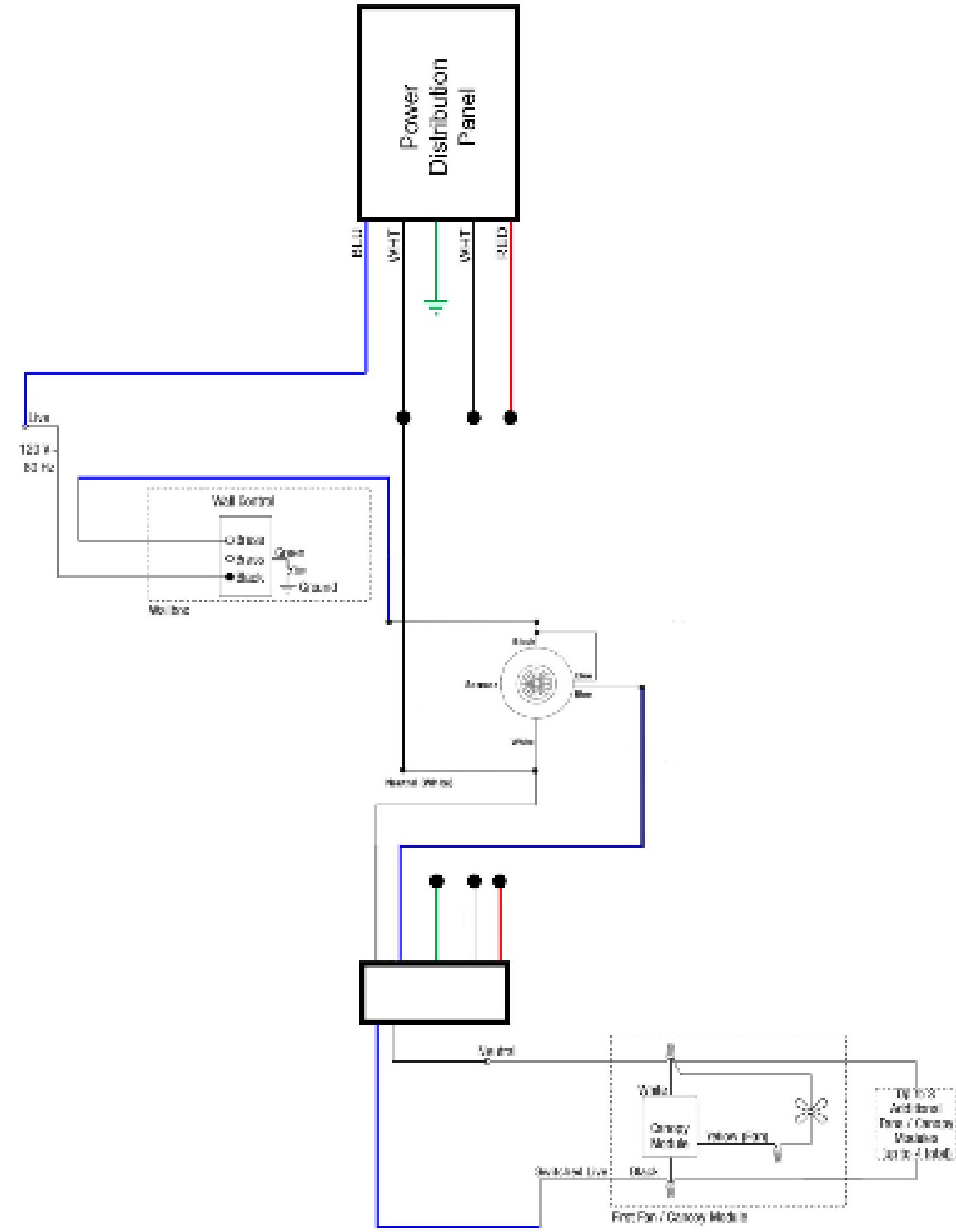

Figure 3 - Final wiring schematic for one set of fans with occupancy sensor installed down-line of wall switch.

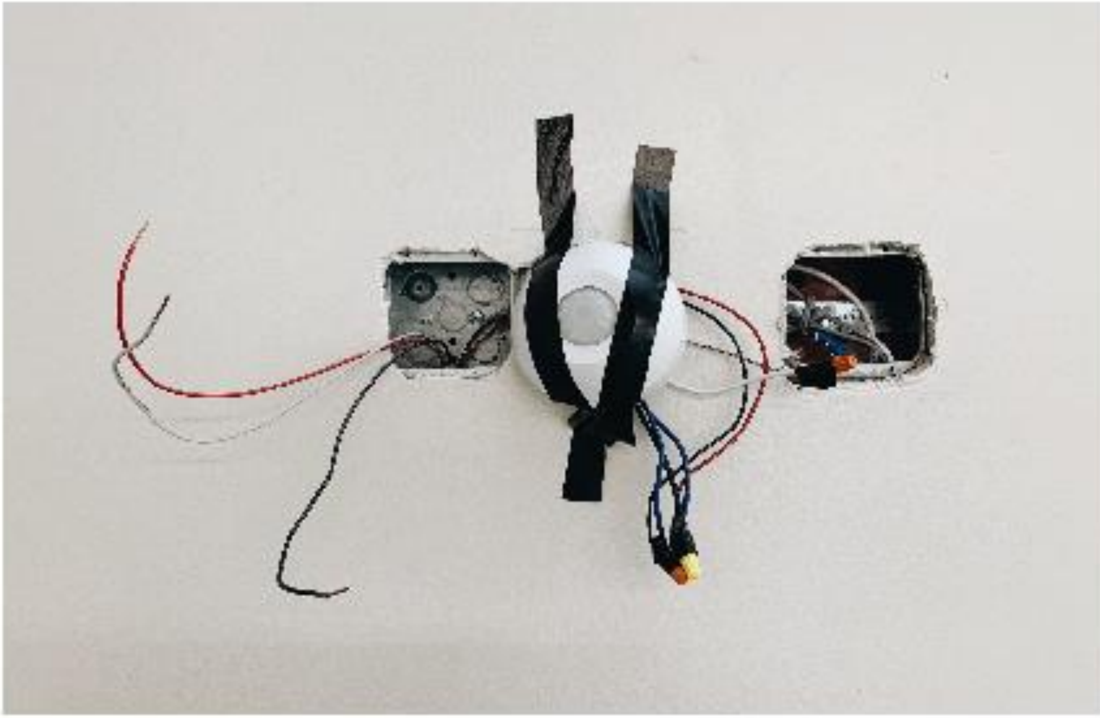

Figure 4 – Testing the sensor prior to final install.

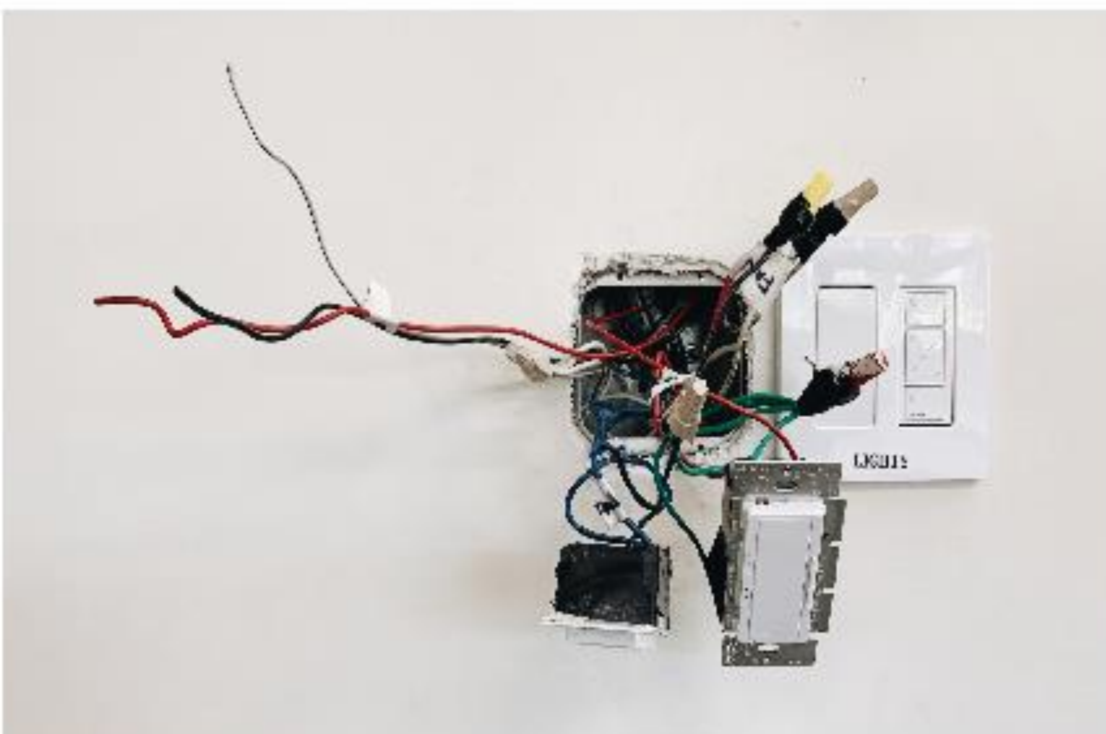

Figure 5 – Testing the wall switch prior to final install.

Figure 6 – Final configuration/installation complete.

The occupancy sensors for the Middle and South fan banks in the FROG 1 building were installed on April 19, 2019. I have since been in contact with Eileen Peppard (she keeps track of the energy usage in the building) and has said they are working as expected. However, there is not enough data for an energy savings analysis. She sent me data (Figure 7) collected from April 22 through May 17, 2019 and it shows that the North fan bank (no sensor applied) was left on for the second weekend in May, but the Middle and South fans did turn off (undetermined if they were intentionally turned off or if they turned off due to vacancy of the room).

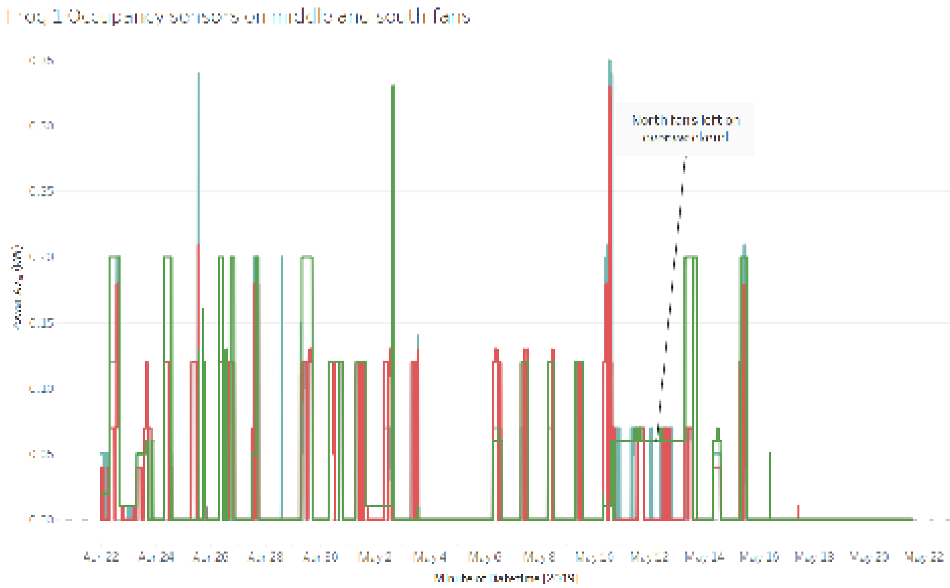

Figure 7 – Energy usage from April 22-May 22, 2019

Overall, I think the project was successful and I am interested to see if there will be any energy savings in the future. I'm sure it will be beneficial nonetheless and I informed Jim Maskrey and Eileen Peppard that I would be interested in helping them install occupancy sensors on the remaining fan banks.

Appendix C. Limitations to the Thermal Comfort Survey

There were limitations to the survey:

- It is unknown what proportion of the occupants in the room participated in the survey. Section 7.3.1 of the Standard (page 16) states that “for under 20 solicited occupants, 80% must respond."
- It is unknown which votes were cast by an instructor and their metabolic rate, which was likely higher than the met 1.0 we used in the PMV model.
- The occupants are not using the room on a regular basis so they are not necessarily acclimated to the conditions. If they spend much of their time in cooler spaces their expectation to be cool might be higher. Section E of the Standard which addresses temporal variation states, “people entering a space that meets the requirements of this standard may not immediately find the conditions comfortable if they have experienced different environmental conditions just prior to entering the space. The effect of prior exposure or activity may affect comfort perceptions for approximately one hour."
- It is unknown where the responder sat in the room so we can’t get a very accurate estimate of the air speed they experienced.
- We did not ask the follow up question on thermal sensation in order to determine which unacceptable responses were too hot or too cool. From previous experience with surveys in Hawai‘i and the temperatures maintained in residences we have monitored, it is dubious there were many unacceptable responses that were too cool. Also, presumably someone who is sensitive to drafty conditions would select a seat in the room which was not directly under a fan.
- Participation in the study was low. To improve participation, it would be beneficial if we offered an incentive. For example: for each time the participant voted, their name went into a drawing at the end of the week or the end of the month to win a gift certificate. Adding the thermal sensation question and having them point out where they sat in the room relative to ceiling fans in the FROG classroom would be beneficial.

Appendix D. Summary Notes from Cheug et al. 2019: PMV Model Accuracy

Cheung et al. 2019 compared the PMV-PPD model predictions for over 50,000 survey responses from the ASHRAE Global Thermal Comfort Database II (Figure 23). Temperatures that the PMV would rank on a thermal sensation scale ranging from -3 (cold) to +3 (hot) had observed mean thermal sensations ranging from -1 (slightly cool) to +1.4 (between slightly warm and warm) using the linear relationship shown in Figure 23. This indicates that the observed thermal sensation was closer to neutral than the PMV predicted.

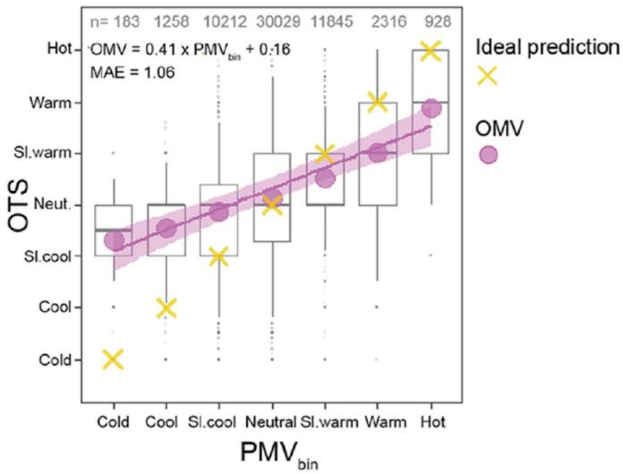

Figure 23. Boxplot of OTS against binned PMV (PMV_{bin}) and the OMV to PMV_{bin} linear relationship from analysis of Cheung et al. 2019.

The linear relationship understates the differences in how the observed thermal sensations diverge from the model prediction at the extremes. There is a larger discrepancy at the cold end (2.2 difference) of the thermal sensation scale than at the hot end (1.0 difference; Figure 24).

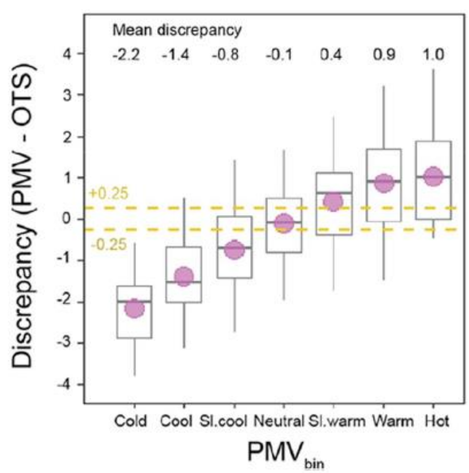

Figure 24. Box plot of discrepancy (PMV-OTS) against PMV_{bin} with mean discrepancy (purple dot) at each PMV_{bin}.

A summary of the thermal sensations predicted by the PMV model and the observed thermal sensation's linear model and the mean observed values is shown in

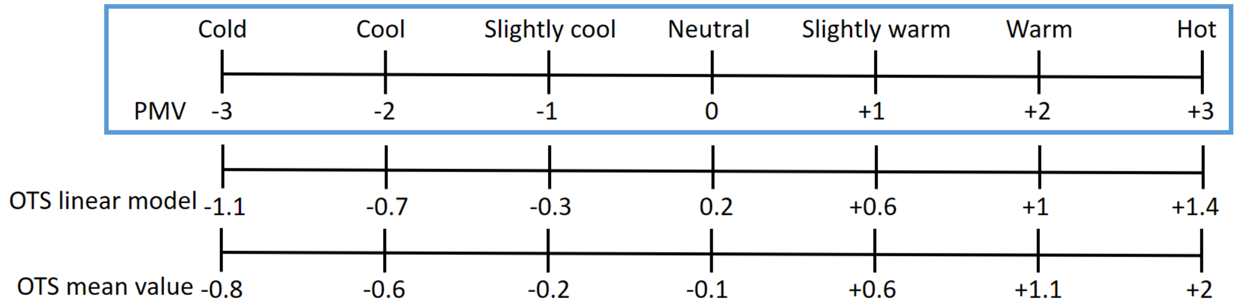

Figure 25. Summary of thermal sensation predicted by the PMV model and the observed thermal sensation's linear model and the mean value of observed sensations.

The observed percent unacceptable (OPU) was much lower for what the PMV model would predict to be cool sensations (Figure 26; red line) which seems to indicate that there is a preference for the cooler temperatures.

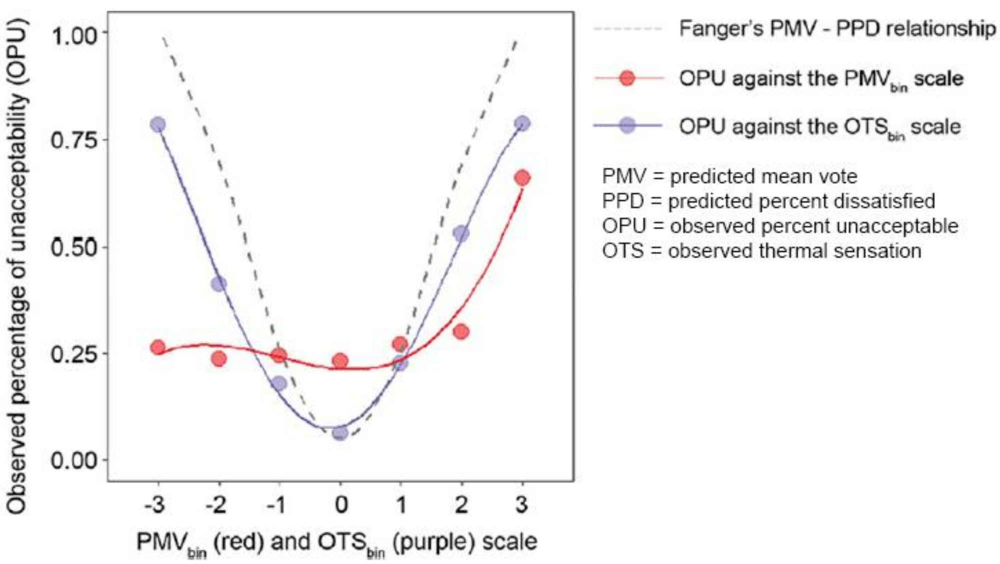

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Figure 26. Summary data from over 50,000 records of the ASHRAE Global Thermal Comfort Database II analyzed by Cheung et al. 2019. Model thermal sensation and percent dissatisfied is compared with survey results.