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CONVECTORS FOR LOW-TEMPERATURE THERMAL DISTRIBUTION

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ABSTRACT

According to the U.S. Department of Energy, residential space heating accounts for almost 10% of the nation's total energy consumption. A popular method in the Northeast, hydronic heating, is often designed to operate at a supply temperature of 82.2 °C (180 °F), with performance decreasing as the temperature is lowered. Next-generation technologies, including condensing boilers, geothermal systems, and solar-based heating systems have improved efficiency at supply temperatures below 60 °C (140 °F), but most hydronic heating units are unable to take advantage of this due to the decrease in performance incurred. The proposed solution is a forced-air-assist retrofit that can be added to an existing baseboard unit that will allow it to operate at the lower water supply temperatures associated with modern heating equipment. This is accomplished through a combination of DC fans and baffles that entrain and redistribute airflow in a manner that is preferable for heat transfer. The system in its current state incorporates 5 fans per linear foot of baseboard and can double the performance of an existing baseboard unit operating at a supply temperature of 60 °C. This exceeds the 0.58 kW/m (600 BTU/hr-ft) heat rating supplied by the manufacturer when the system operates at 82.2 °C, indicating the potential to lower the water supply temperature even further. Future work includes the optimization of the baffles and the number of fans to maximize heat transfer and minimize the added electricity consumption from the fans.

KEYWORDS: Energy Efficiency, Convection, Forced Air, Hydronic Heating, Baseboard Radiator

1. INTRODUCTION

In the United States, over 40% of the energy use in homes is for space heating [1]. On average, the United States energy budget is about 105 EJ, 20% of which comes from the residential sector [2, 3]. As such, many states have developed new regulations to improve energy efficiency in this sector, with both economic and environmental motivations. In particular, New York's 'Reforming the Energy Vision' strategy aims to reduce greenhouse gas emissions by 85% by 2050 and build an affordable energy system for homeowners [4]. This is relevant to residential heating, as heating and cooling represent 32% of New York State's combustion-related greenhouse gas emissions and is a measurable portion of household expenses [5]. As a result, recent years have seen a rise in the development of next-generation heating technologies, such as condensing boilers, geothermal systems, and solar-based heating systems. These technologies can operate with over 90% efficiency when operating at the proper conditions, which generally requires that the system use the lowest hot water supply and return temperatures possible. In the case of condensing boilers, the threshold temperature is the dew point of the water, which is usually between 48.8 °C (120 °F) to 54.4 °C (130 °F). Above this point, there is no benefit to the use of the system over a standard non-condensing boiler [6]. This presents a limitation on the acceptance of such technologies, as the rest of the heat distribution system must be compatible with the lower supply and return temperatures.

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Of the heat distribution systems directly impacted by these limitations, baseboard radiators are by far the most commonly used in the Northeastern United States [7]. Hydronic heating systems such as these are small and easily integrated while requiring far less electricity to run than other options. Baseboard radiators work by circulating hot water from the boiler through finned tubes, which then distribute energy via natural convection. Natural convection is driven by buoyancy differences; air within the baseboard enclosure is heated, which reduces its density. Gravity then forces the colder air into the bottom of the baseboard, and the warmer air exits at the top. Heat transfer is driven primarily by the magnitude of the temperature difference between the finned tube and the surroundings, and therefore the performance of the baseboard radiator is a strong function of the supply temperature. Baseboard radiators are generally designed with a supply temperature of 82.2 °C, with noticeable performance degradation as the temperatures go below 54.4 °C. Due to this, the adoption of technologies such as condensing boilers by homeowners with hydronic heating systems can be costly, as it may require the replacement of their heat distribution system as well as the initial capital cost associated with a boiler upgrade.

Much research has gone into developing means of compatible operation between hydronic heating systems and next-generation heating technologies. One method investigated was to lower the flow rate, which would reduce the water return temperature and improve efficiency without requiring the supply temperature to be driven too low. However, market barriers such as a lack of standardized flow setting techniques and proprietary controls amongst manufacturers have proved to be difficult to overcome. [8, 9]. As a result, while there is much research on the individual components, there is still little information available to determine the best combination of settings for optimal performance. Another approach was to incorporate the flow of cold outdoor air in an integrated ventilation system, where the incoming air is pre-heated to the indoor temperature via baseboard radiator [10]. While a significant increase in heat ratings was found, the arrangement requires a costly and complex integration in the building's structure and is not suited for existing residential applications. Other attempts to modify existing systems have been made, but no economic and reliable solution has yet been found.

This research explores the integration of a forced-air assist retrofit that is added to existing baseboard units to improve their performance when supplied with low-temperature water. Utilizing small fans and baffles, the design aims to increase the flow rate of the air through the baseboard fins. Increasing the flow of air into the baseboard housing and directing it preferentially along the fins allows the unit to operate at lower temperatures while still satisfying the heating load. Compared to existing attempts, this method focuses on improving performance where the heat is delivered, as opposed to modifying the performance where the heat is generated. This allows for more optimization for the specific baseboard model's geometry as well as more varied feedback response methods. Others who have investigated incorporating a fan along the baseboard have done so by designing a new baseboard unit with a fan, which does not solve the issue of current homeowners and their existing systems. They would still be required to upgrade their heat distribution system, removing the benefit and further preventing the adoption of advanced technologies. The development of this concept is essential, as it allows for the adoption of next-generation heating technologies amongst a wider audience at a fraction of its current cost.

2. EXPERIMENTAL THEORY AND SETUP

2.1. Experimental Setup

The experimental setup utilizes a 6 ft long SlantFin/Baseline 30 baseboard unit that is commonly used in residential applications. The unit is mounted on a wooden vertical section to approximate that of a household wall. A 1500 W circulation heater is used in conjunction with a common circulator, the Taco 007. The electric heater served to simulate the boiler for heating the supply water to the desired temperature. The temperature in the heater was monitored and maintained using a PID temperature controller, which was powered by a 120VAC supply. The input to the temperature controller utilized a J-type immersion thermocouple mounted at the exit of the heater. The temperature at the inlet and outlet of the baseboard unit was measured using two hot-wire high accuracy DIN 1/10 RTDs. The RTDs can operate between –29 °C and 100 °C with an uncertainty of approximately ± 0.05 °C in the temperature range utilized. The temperature measurements were then recorded in real-time using a Keithley 2701

Digital Multi-meter with 6½-digit precision. The flow rate was measured using an Adafruit Hall-effect flow sensor calibrated to ±2% accuracy, which also acted as a low-flow switch for safety. In the event flow dropped below the set limit, the heater would be shut off to avoid damaging the setup. The flow through the setup was controlled manually via a 1.9 cm valve located at the outlet end of the baseboard unit. The water circulation loop utilized 1.9 cm PEX tubing, which is normally used in most hydronic heating applications. A schematic of the system as described is shown in Fig. 1.

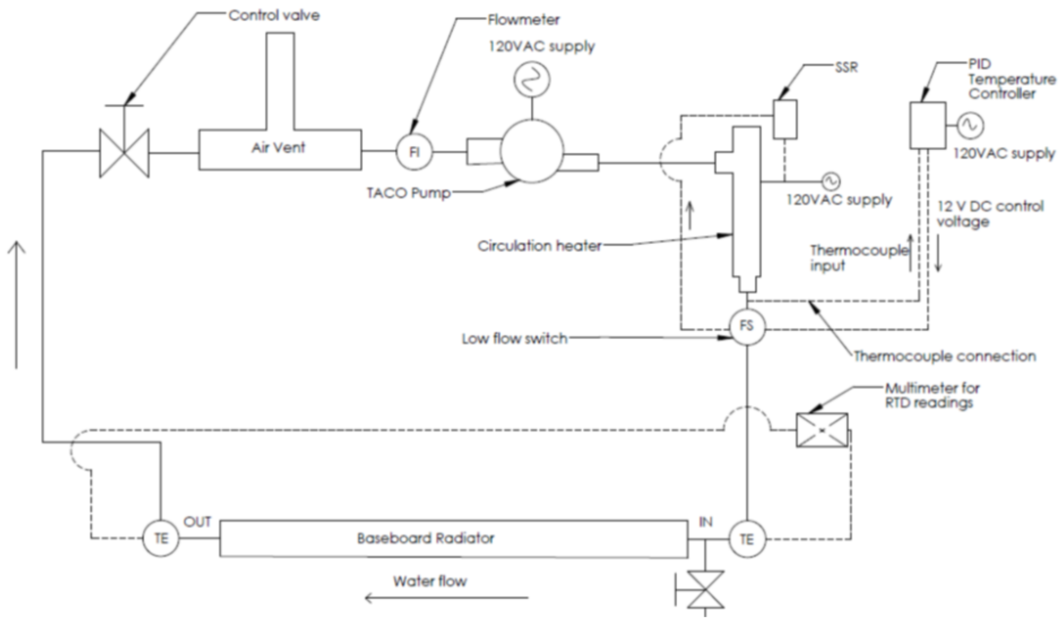

Fig. 1 Schematic of the experimental setup.

To avoid possible accumulation of air in the system, a two-phase approach was used. A small section of pipe open to the atmosphere was added at the highest section in the line after the valve, which also acted as a means of charging the system with distilled water. Directly beyond this point, the pipe itself was inclined to feed any remaining air bubbles to a float valve, thereby removing air from the system. A drain valve was placed at the lowest point at the inlet side of the baseboard unit, which allowed for the system to be drained in the event of a system failure. The DC fans utilized in the experiment were 10 60x25mm SUNON MagLev fans rated for 12V DC. Investigation was limited to 60 mm fans, as they had the best balance of volumetric airflow and size. To power the fans, a standard DC power supply was used. The fans were mounted with magnets at the bottom of the baseboard unit directly below the fins and inside the housing such that they were not visible during operation, as in Fig. 2. This is done to ensure the inlet of the fans remains unblocked, and to be aesthetically pleasing to consumers. The physical construction of the setup is shown in Fig. 3.

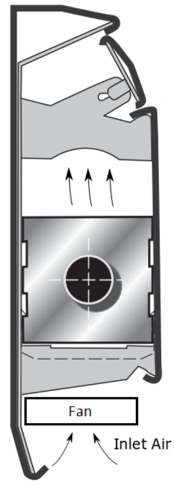

Fig. 2 Side view of the proposed solution [11].

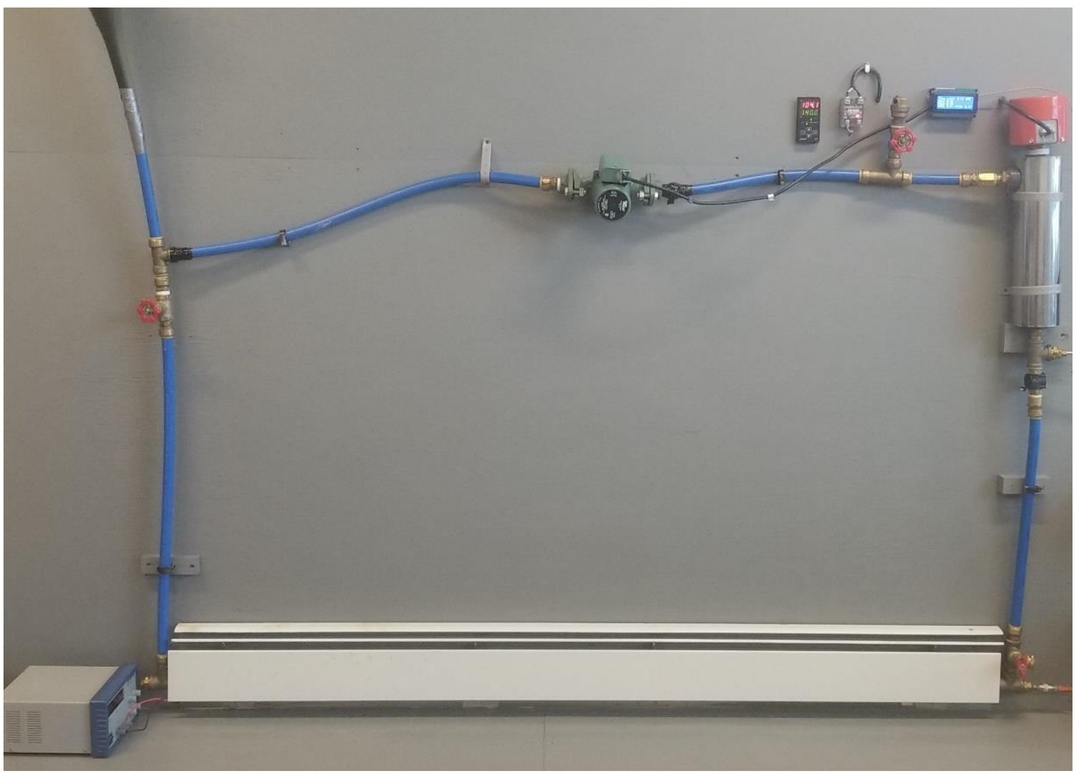

Fig. 3 Physical setup for baseline readings.

2.2. Model Setup

The concept of incorporating an add-on-fan blower to a hydronic heating system is not novel. In fact, several other authors have investigated its utility and have found promising results [12]. Instead, modeling in this vein exists as a means of investigating fan selection and the required airflow, as well as evaluating the prototype baffle geometries before their experimental trials. Due to this, SolidWorks Flow Simulation 2017 was the primary software utilized, as it allows for the inclusion of fan boundary conditions and includes the option to import custom fan curves. The entire baseboard unit was modeled in SolidWorks and divided into 0.3 m (1 ft) segments, along with the vertical wall and floor upon which it is mounted. Contrary to a standard model, which would only include a handful of fins, a standard unit length of baseboard was chosen to better reflect the geometry of the fans and baffle to be developed. As such, the minimum number of fans investigated in these trials was 1 fan per linear foot of baseboard. The computational domain was roughly 0.25 m^3 (9 ft^3) to obtain an idea for the flow field as it approaches and exits the baseboard unit. A grid refinement study was done utilizing a mass balance as the primary variable for convergence. With this, the resulting mesh size was approximately 6 million cells, as shown in Fig. 4. The volume flow rate of the water remained constant at all trials and was set to be 0.23 m^3/hr (1 GPM). Periodic flow freezing and refinement was enabled in order to improve accuracy and reduce computation time.

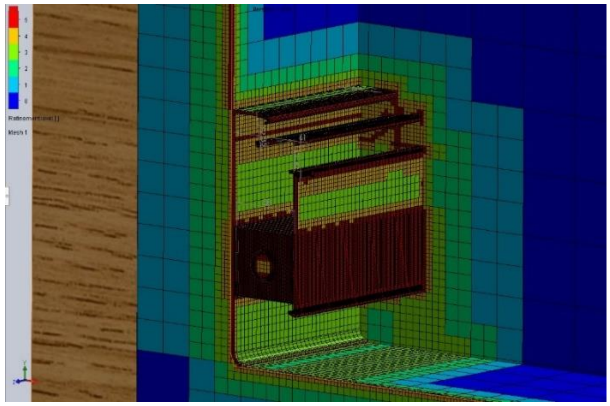

Fig. 4 Model Mesh.

Before utilizing the model fully, it was first validated using the natural convection case at the various supply temperatures noted by the manufacturer. The manufacturer ratings, as well as those obtained in the lab, were used as comparisons to compute the relative error of the computations. Afterward, several prospective fan candidates were chosen and evaluated, with the 12 v SUNON MagLev fans being chosen for their exceptional airflow. Beyond using the model to select the fans, the efficacy of such is also evaluated by observing how the temperature distribution of the surface of the fins changes as the airflow is increased or perturbed in various directions. This information is required when the next phase of the investigation is reached, and baffles are developed further.

2.3. Experimental Procedure

Initially, baseline readings for natural convection were established utilizing the system as shown in Fig. 3. The operation of the experiment began with setting a target temperature in the controller and the required flow rate, which for all experiments was 0.23 m^3/hr. The system then ran for at least 45 minutes to 1 hour to reach steady-state conditions according to the AHRI standards. The inlet and outlet water temperatures were then recorded every 20 seconds over a period of 30 minutes. The heat load supplied to the room was approximated to be the heat loss from the baseboard, as any losses in the system are neglected. The heat transfer from the baseboard was calculated via [13]:

$$Q = \rho \dot{V} C_p (T_{out} - T_{in}) \tag{1}$$

This was then normalized by the effective length of the baseboard, which is 1.68 m in this case. The resulting heat rating was compared with Slant/Fin's ratings from 48.8 °C (120 °F) to 82.2 °C (180 °F) in increments of approximately 5.6 °C (10 °F). However, two modifications were made in order to provide an accurate comparison. Since Slant/Fin considers a room temperature of 18.3 °C (65 °F) when determining their heat ratings, an air correction factor is applied:

$$A_{cf} = \frac{T_{in} - T_{amb}}{T_{in} - T_{Slant}} \tag{2}$$

The air correction factor is less than unity when the ambient temperature is higher than that tested by the manufacturer. In a sense, the room is 'pre-heated' and the effectiveness of the baseboard drops. Therefore, multiplying by the inverse of the air correction factor results in a more accurate reading. Additionally, to be consistent with AHRI standards for baseboards, the rating is multiplied by 1.15, with the final expression being:

$$Q_{adj} = \frac{1.15[Q]}{A_{cf}} \tag{3}$$

Once the baseline had been established, the forced convection case was investigated using fans arranged at 5 and 1 fans per linear foot of baseboard. Due to the availability of materials, the existing setup was modified such that only a section of the baseboard was utilized, shown in Fig. 5.

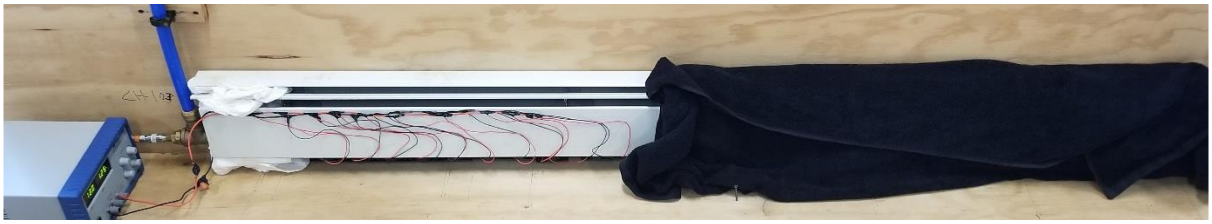

Fig. 5 Forced Convection Setup.

First, the entire baseboard was covered with towels to arrest airflow, and the heat loss observed was computed. This was then normalized by the length of the baseboard to compute the normalized heat loss, q_{loss}. Then, a 0.685 m (2.25 ft) segment was left uncovered, in which the fans were installed at the above periodicities. The modified heat transfer for forced convection was determined to be:

$$Q_f = Q - (L_{towel})(q_{loss}) \tag{4}$$

The heat rating is then normalized by the portion of baseboard with fans:

$$q_f = Q_f / L_{fan} \tag{5}$$

This value is then adjusted according to equation (3). With heat losses considered, the system was run at the same temperatures as those for the natural convection case with the fans now running at 12V, 9V, 6V, and 4.5V for each temperature case. The selection of 1 and 5 fans per linear foot was chosen as an economic lower/upper bound to compare against for when baffles are considered.

3. ERROR ANALYSIS

The error associated with determining the heat rating was calculated using the uncertainty method described by Jon Longtin [14]. The heat rating consists of several variables, namely the density, volumetric flow rate, specific heat, and inlet/outlet temperatures. The density and specific heat are constants at the prescribed temperatures; only the change in temperature and flow rate contribute to uncertainty. The uncertainty is computed using the summation in quadrature method:

$$\Delta Q_f = \left(\left(\frac{\partial Q_f}{\partial \dot{V}} \Delta \dot{V} \right)^2 + \left(\frac{\partial Q_f}{\partial \Delta T_{in}} \Delta T_{in} \right)^2 + \left(\frac{\partial Q_f}{\partial \Delta T_{out}} \Delta T_{out} \right)^2 \right)^{1/2} \tag{6}$$

The dependence of the uncertainty in the heat ratings on the variables described is depicted visually in Fig. 6. Uncertainties in the volumetric flow rate and inlet/out temperature were found using the instrument manufacturer's uncertainty, resolution uncertainty, and statistical uncertainty. The results obtained for each condition are shown in Table 1 and Table 3.

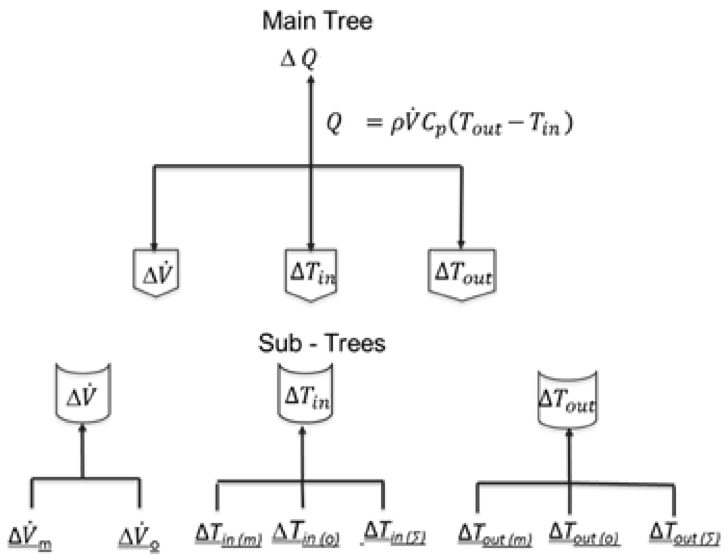

Fig. 6 Heat rating uncertainty tree.

4. RESULTS AND DISCUSSION

4.1. Natural Convection

The measured ratings in the lab, even after adjustments, differed widely from those reported by Slant/Fin. The degree of the disparity is on average 20%, with exact discrepancies shown in Table 1. Without Slant/Fin's uncertainty values, it is difficult to narrow down where the discrepancy originates. However, it is likely due to the ambient conditions of the testing room. Therefore, the experimentally obtained values will be used as the primary comparison with Slant/Fin's ratings acting only as target goals. The model is in good agreement with what is captured experimentally, with error below 15% in all cases. This is evident in Table 2, as well as the divergence from Slant/Fin's ratings.

Table 1 Natural Convection results vs Manufacturer Heat ratings.

Inlet water temp [ºC (°F)]	Measured Heat rating [kW/m (Btu/hr-ft)]	Uncertainty [%]	Slantfin ratings [kW/m (Btu/hr-ft)]	Difference in heat rating
48.8 (120)	0.15(155)	12.5	0.202(210)	26%
60 (140)	0.232 (241)	11.5	0.308(320)	25%
71.1(160)	0.345 (359)	10	0.433(450)	20%
82.2(180)	0.455(473)	10	0.558(580)	18%

Table 2 Natural Convection Simulation Results.

Inlet water temp [ºC (°F)]	Simulated Heat rating [kW/m (Btu/hr-ft)]	Percent error in heat rating [Measured vs Sim]	Percent error in heat rating [Sim vs Slant/Fin]
48.8 (120)	0.144(150)	3.5%	28%
60 (140)	0.23(240)	0.9%	25%
71.1(160)	0.31(320)	11%	29%

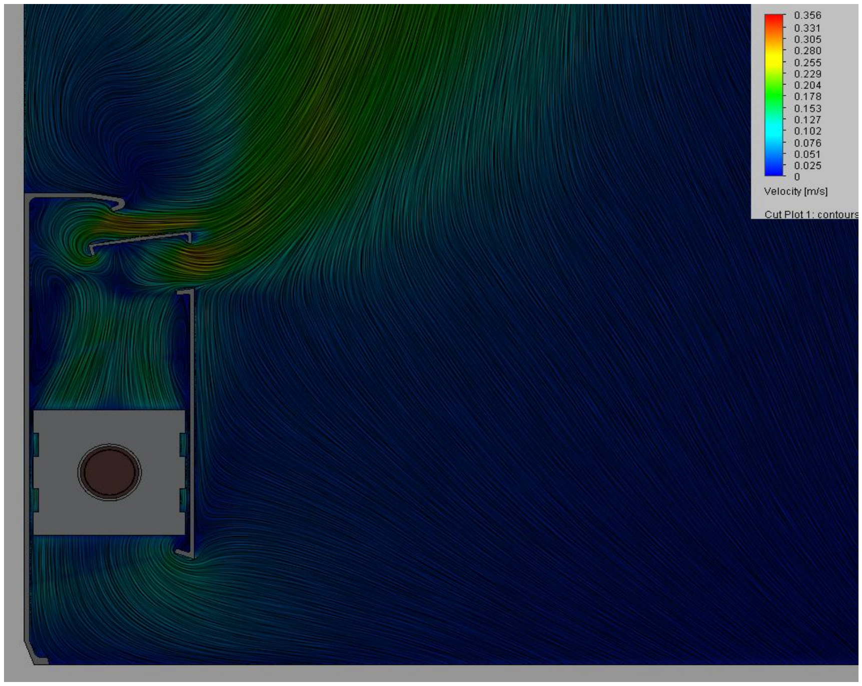

Fig. 7 Natural convection streamlines.

The model indicates in Fig. 7 that the airflow out of the top of the baseboard unit is directed vertically upward, which was also validated in the experiment. This is important for preserving the flow field near the baseboard unit, as deviations from this when forced convection is introduced can impact the strength of the natural convection portion, as well as the uniformity of heating. The model's prediction tends to underestimate heat transfer in the natural convection case, which likely requires more refinement of the radiation aspect of the model. The flow field produced accurately reflects what is expected for natural convection, with most of the flow in the laminar regime. Additionally, the Grashof Number computed is much greater than unity, which suggests the natural convection aspect is indeed what is being captured.

4.2. Forced Convection

The streamline plot generated by the model with forced convection indicates a vastly different flow field. As expected, without any baffles at the outlet, the flow turns turbulent once in contact with the fans. This is depicted in Fig. 8, which is beneficial for heat transfer. However, the trajectory of the outlet air has changed to be more horizontal, the impact of which is not yet determined. The optimal position of the fans relative to the fins was investigated in terms of distance and orientation. It was found that the outlet of the fan should be directed at the center of the tube. While the optimal distance has not yet been determined, it is bounded to be between 0.1 to 3 cm, which may change once baffles are added at the outlet of the fans. With flow assisted by an array of fans, a remarkable increase in the heat rating was obtained. With 5 fans per linear foot at 60 °C (140 °F) and 12 V, the heat rating supersedes that of a supply at 82.2 °C (180 °F). This suggests that without any optimization, it is possible to already convert existing baseboard units to operate at below their rated temperature and maintain performance. Table 3 and Table 4 show the tested temperatures using 1 and 5 fans per linear foot at the fan's rated voltage.

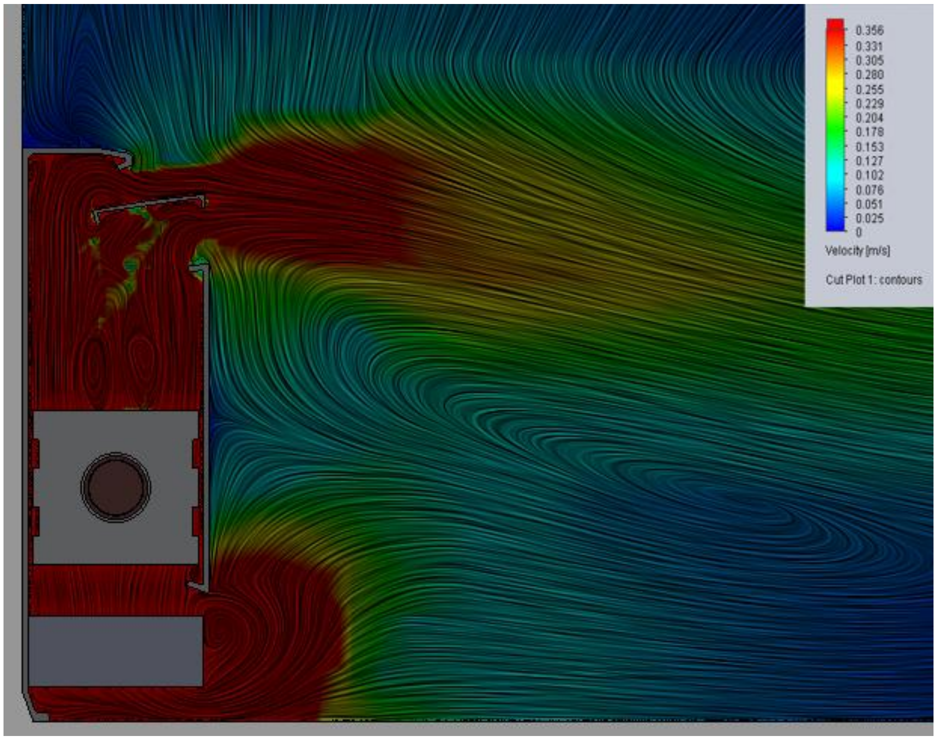

Fig. 8 Forced Convection Streamlines.

Table 3 Measured Heat ratings incorporating 5 fans per linear foot of baseboard at 12V.

Inlet water temp [°C (ºF)]	**Measured Heat rating [kW/m (Btu/hr-ft)]**	**Uncertainty [%]**	**Percent increase from Natural Convection**
37.8(100)	0.353(370)	10	145%
60(140)	0.714(742)	11	210%

Table 4 Measured Heat ratings incorporating 1 fan per linear foot of baseboard at 12V.

Inlet water temp [°C (ºF)]	**Measured Heat rating [kW/m (Btu/hr-ft)]**	**Uncertainty [%]**	**Percent increase from Natural Convection**
37.8(100)	0.095(98.8)	9.5	21%
60(140)	0.285(296)	6	23%

The percent increase in the heat rating is notable even with only one fan, yet there appear to be limits on the effectiveness of the initial approach at lower temperatures. The percent increase in the heat rating at 60 °C (140 °F) as compared to 37.8 °C (100 °F) is 1.5 times as much, indicating that a more intricate approach is necessary once temperatures drop to the ranges required by the next-generation heating technologies. As such, investigation has shifted to incorporating baffles that redirect airflow preferentially, which may bypass the limits that have already been reached. Interestingly, the pattern of diminishing returns can be seen at each temperature by changing the power input to the fans.

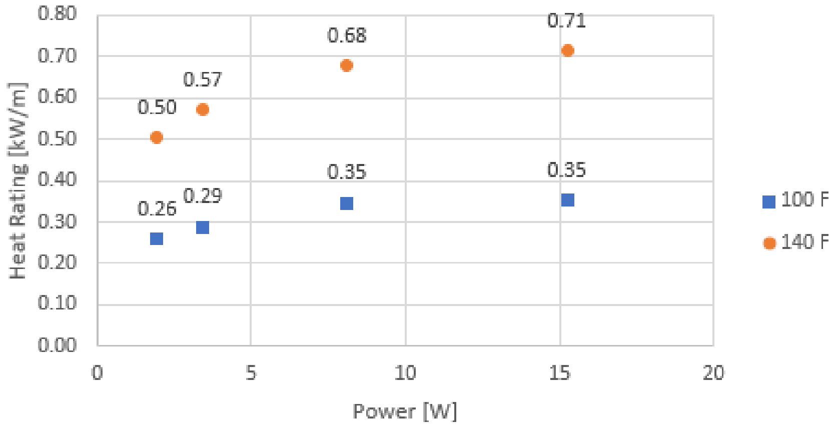

Fig. 9 Heat rating as a function of fan power for 5 fans/ft.

The power curves for the fans in Fig. 9 depict diminishing returns as power is increased, indicating that an optimum airflow rate exists. This occurs at 8 W at both 5 fans per linear foot and 1 fan, suggesting that the baseboard becomes fully saturated at airflow above this point. Compared to the circulation heater's 1500 W rating, the amount of electricity supplied to the fans is negligible. However, to determine whether the solution as it stands is economically feasible, a true comparison should be made to the boilers that will be used in practice with estimations on usage. According to NREL, typical residential boilers have an input capacity between 17.5 kW (60,000 BTU/hr) and 88 kW (300,000 BTU/hr) [15]. A general rule of thumb for sizing a boiler in the Northeastern climate zone is known to be 50 BTU/hr-ft^2. A standard household in this region occupies approximately 2000 ft^2 [16], thus a good estimate of the average household's boiler output capacity is 29 kW (100,000 BTU/hr). A typical natural gas boiler may have efficiencies of 86% and 96% operating in non-condensing and condensing mode, respectively [6]. If operating in non-condensing mode, a boiler would require an input capacity of 34 kW, whereas in condensing mode only 30.5 kW. For simplicity, consider operation at maximum load, where the difference is 3.5 kW of fuel. Natural gas has an energy content of on average 37.5 MJ/m^3, which indicates the difference between the boiler's operation modes amounts to 0.336 m^3/hr of natural gas. Annually, this would utilize over 2900 m^3 of Natural gas. With the average rate over the entire year of 2018 being \$0.45 per m^3 [17], this equates to a difference in operation of \$1300 per year. If now the fans suggested were running continuously at 8 W, allowing the boiler to act in condensing mode, the resulting electricity requirement annually would be 70 kWh per fan array. With the U.S. average utility rate being \$0.12/kWh [18], this amounts to only an additional \$8 per year per fan array. While the boiler will naturally not operate at full load year-round, the analysis indicates that the solution is economically viable even in the worst-case scenario.

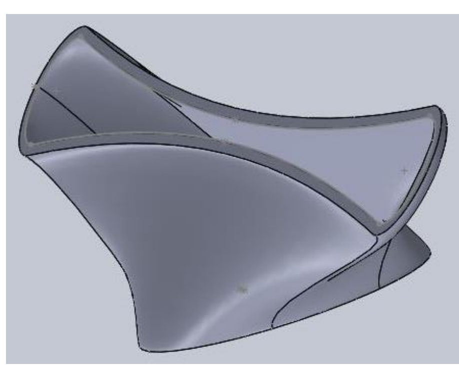

Fig. 10 Uniform-flow baffle design.

Several flow-distribution baffles have been analyzed in SolidWorks to improve the effectiveness of an individual fan to be on par with that of an array of fans. The designs are evaluated based on size with smaller preferred for ease of installation and percent increase in heat rating as the dominant factor. Thus far, the most successful is illustrated in Fig. 10, in which the flow is encouraged to spread from the outlet of the fan and become more uniform. Replacing an array of 5 fans with one fan and a baffle is a non-trivial task, but simulations predict a modest increase in heat ratings at 60 °C (140 °F) to approximately 0.48 kW/m (500 BTU/hr-ft) using the uniform flow design. This alone would reduce the cost of converting an existing system to the forced-air-assist method described above by reducing the number of fans required.

Challenges associated with design modification are primarily limited to space constraints and installation costs. The design must be able to fit inside the baseboard housing, whose typical depth and height range from 60-70 mm and 40-60 mm, respectively. The fans evaluated thus far further subtract from this upper limit, leaving the optimal baffle to have a maximum height of 35 mm and a depth of 70 mm. Regardless of baffle design, operating costs are of negligible importance, as they represent a minuscule portion of the yearly heating bill based on preliminary analysis. Installation does not require any technical background and hence the bulk of the installation cost is the capital cost. Therefore, minimizing the size and amount of material used is crucial in further developing the design of the baffle.

5. CONCLUSIONS

This article presents a forced-air-assist retrofit for existing residential baseboard radiators to allow for their operation at lower water supply temperatures. Tests were performed on a typical residential baseboard model at over a range of supply temperatures. The proposed solution in its preliminary stages has demonstrated its ability to greatly magnify heat transfer at low supply temperatures between 37.8 °C (100 °F) and 60 °C (140 °F). Heat ratings at these temperatures can exceed that of standard ratings at existing design temperatures by up to 200%. The initial baffle design shows that it is possible to achieve the performance of a multi-fan array with a single fan. A brief economic analysis indicates that the cost of operating the retrofit is negligible compared to the cost savings from switching to a condensing boiler.

Further work will consist of determining the optimal baffle geometry to maximize airflow and minimize the return temperature. Additionally, the feedback and control mechanism for the fans will be investigated in order to incorporate variable output. Investigation into reducing the size of the retrofit is also of concern, as it should be able to fit a wide variety of baseboard dimensions. Noise insulation mechanisms may also be explored, as silent operation is important for consumer acceptance. With this, baseboard units can be operated at lower temperatures, allowing for a wider adoption of high-efficiency heating systems amongst a larger market of consumers.

6. ACKNOWLEDGMENT

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7. NOMENCLATURE

BTU	British Thermal Unit
Q	Heat emitted by baseboard element during natural convection
Q_f	Heat emitted by baseboard element during forced convection
Q_{adj}	Adjusted heat emitted
A_{cf}	Air correction factor
q	Normalized heat emitted by baseboard element during natural convection
q_f	Normalized heat emitted by baseboard element during forced convection
q_{adj}	Adjusted heat rating
T_{amb}	Ambient air temperature
ρ	Water density
$\dot{V}$	Volumetric flow rate
C_p	Water specific heat
T_{in}	Inlet water temperature
T_{out}	Outlet water temperature
PID	Proportional Integral and Derivative
HVAC	Heating, Ventilation, and Air Conditioning
AHRI	Air Conditioning, Heating and Refrigeration Institute
GPM	Gallons per minute
RTD	Resistance Temperature Detector
PEX	Crosslinked Polyethylene
ρ	Air density
A	Inlet/outlet area under consideration for baseboard
v	Air inlet/outlet velocity

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