Design and Optimization of a Displacement Ventilation System for a Large Retail Store

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ABSTRACT

This paper describes the use of combined thermal and computational fluid dynamics (CFD) analysis for the design of an inverted displacement ventilation system in a large retail store. The analysis was used to demonstrate the system performance and energy savings to the owner, in order to facilitate the acceptance of the system and replacement of the owner's standard overhead mixing, plenum box air supply. The CFD analysis demonstrated that the proposed system would perform as intended, would meet the owner's temperature requirements, and would improve the design by optimizing supply temperature and volumetric flow rate. A life cycle cost analysis showed the possible economic benefits for the proposed displacement ventilation system compared to the owner's standard system. The proposed building site is situated near Dallas, Texas. Once built, the owner will monitor the store with displacement ventilation and compare its energy performance with a nearby store utilizing their standard, overhead plenum-box system. This comparison will be made public following the analysis period in a separate report by the owner.

Introduction

Following advances in sustainable supermarket design, as seen in stores such as Sainsbury's Greenwich store in the United Kingdom, there has been considerable interest in improving the energy and environmental performance of retail stores. The owner is building two "experimental" retail stores to study low energy building systems and sustainable design techniques for future use in their standard store designs. These stores are utilizing many energy efficiency measures, including the use of an inverted displacement ventilation system.

The CFD study was performed as a part of the engineering team's design process to demonstrate to the owner that the proposed inverted displacement ventilation system would perform per the design intent and to show that internal air temperatures in the occupied zone would be maintained within the client's accepted range, both for a variety of conditions. The CFD study investigated conditions within the core of the building in dry goods retail areas with shelving aisles.

The engineering design team also used the analysis to optimize the supply air temperature and flow rate. Many HVAC systems are based on rules of thumb, derived from past experiences or prior analyses. The initial supply air temperatures and flow rates were based on the team's prior experiences using displacement ventilation in commercial and industrial settings. The team analyzed the sensitivity of the results to changes in both supply air temperature and flow rate to see the effects on temperature and air velocity within the occupied zone. The team also wanted to verify that the diffuser layout in conjunction with the placement of shelving would not result in intolerably hot or cold aisles. The system used standard roof top units, operating at constant volume.

A life cycle cost analysis (LCCA) of the final design was performed for the main retail and grocery areas of the store. Because the owner builds stores throughout the country, a variety of utility rates were investigated to show the economic feasibility in different areas of the country. The LCCA included equipment first costs, estimated maintenance costs, replacement costs, and utility (energy) costs.

The proposed ventilation system design for the main retail areas of the store uses high-mounted fabric ducts for supply air distribution to create an inverted displacement ventilation system. The fabric ducts are porous and distribute an even air flow radially in all directions along the entire length of the duct. The ducts are mounted so the bottom is approximately 11-ft above the floor. Supply air temperature is moderate (typically 65-68°F) compared to overhead systems, and the air is distributed with low momentum through the openings in the fabric. The supply air quickly mixes with the surrounding air and slowly falls to the floor level. At the floor level the air is warmed by the occupants and other heat sources and rises slowly to the upper levels of the room. The return air opening to the air conditioning unit is located just below the roof deck to return the hottest air.

Displacement ventilation systems have several potential advantages over typical cold-air mixing systems. Due to the thermal stratification and high-level return the majority of the lighting load and roof heat gain directly heats the return air rather than the lower conditioned zone. Therefore, these loads do not need to be included in the occupied zone load calculations. Also, due to the elevated supply air temperature and elevated return air temperature, the air handling unit has an extended economizer range compared to systems supplying at 55°F and returning at 75°F. The extended economizer range can result in significant energy savings – this is discussed further in the LCCA portion of the report.

CFD analyses have been used extensively by Arup and others to examine the performance of ventilation systems, including displacement ventilation (Lavedrine & Woolf, 2003 – Woolf, 1999). There are also many design guides for displacement ventilation systems, often from the manufacturers of the system components (Trox, 2003 – Kranz, 1999).

CFD Analysis

The aims of the CFD study were to demonstrate that the proposed displacement system would perform as anticipated under several conditions, to determine the effects of the shelving and diffuser layout, and to determine the effects of varying the supply air temperature. Demonstrating the system performance to the owner was also important, as the owner wanted verification that the system would meet their temperature requirements of between 72°F and 76°F on average in the occupied zone. "Dumping" (i.e. cold air flow at higher velocities) was also a concern, especially at the lower supply air temperatures considered. The CFD also showed the owner whether the shelving layout would cause problems in the airflow distribution, resulting in hot and cold aisles.

The simulations were carried out using the CFD code StarCD v3.15. The computational domain represented an idealized shopping area with shelving. The zone studied was a completely internal zone representing the dry goods area of the store. The dry goods retail area represents the majority of the retail floor area of the store. Due to the store layout, all the retail areas are internal zones – storage, kitchens, offices, and other support areas ring the main retail space, buffering it from external conditions. Infiltration was not included in this study as the store design uses entry vestibules combined with air curtains to minimize infiltration.

The study looked at conditions on a cooling design day in summer and on a winter design day. The store is operated and open to the public 24 hours a day throughout the year and the occupied zone is in cooling mode at all times due to internal gains. The two extreme design conditions were investigated with the presumption that a design that worked at the extremes would also work at conditions between the extremes. Part load (occupancy) conditions on a cooling design day were also investigated. Because CFD analyzes a single point in time, a separate thermal analysis of the space was used to simulate the design days and select the hour that represented the most extreme cooling and heat conditions, respectively.

Simulations

A brief description of Computational Fluid Dynamics (CFD). CFD is concerned with obtaining numerical solutions to fluid flow problems by using computers. The advent of high-speed and large-memory computers has enabled CFD to obtain solutions to many flow problems including those that are compressible or incompressible, laminar or turbulent, chemically reacting or non-reacting. Fluid flows are modeled by a set of partial differential equations called the "Navier-Stokes equations". Except for special cases, no closed-form solutions (i.e. exact) exist to the Navier-Stokes equations.

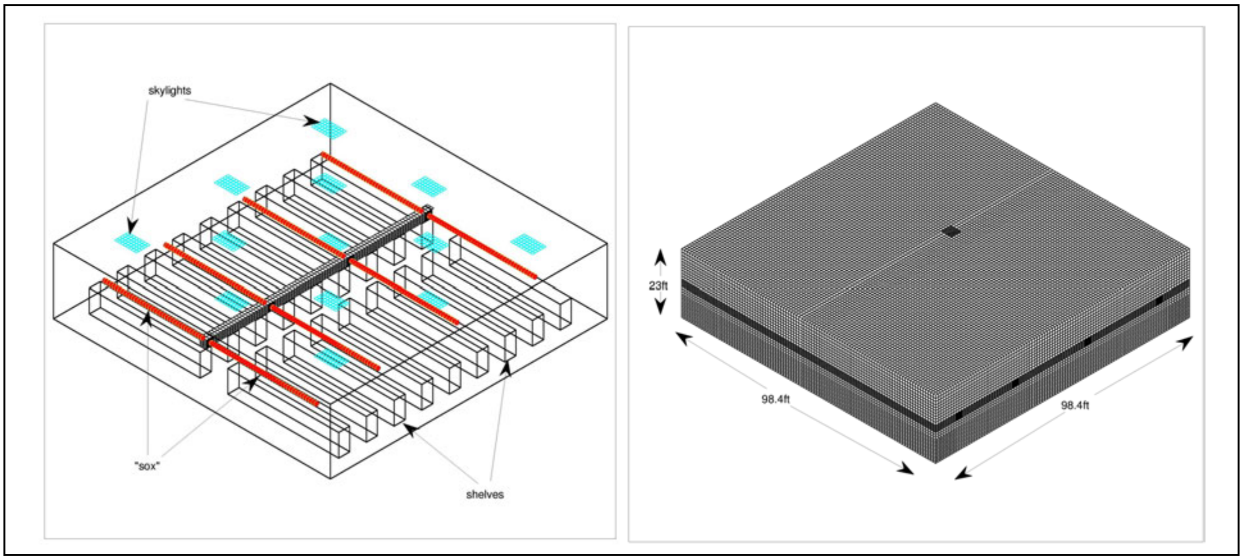
Therefore CFD is the *art* of replacing the differential equation governing the fluid flow with a set of algebraic equations - this process is called discretization in which the first step is to divide the physical domain in which the flow occurs into small volumes or "cells". The simplified equations in each cell can in turn be solved with the aid of a digital computer to get an *approximate* solution.

Computational domain. The area represented in the CFD model corresponded to an idealized shopping area with shelving. The area modeled was approximately 98 feet long by 98 feet wide and 23 feet high. The total volume of the modeled space was just under 200,000 ft^3. The supply ducts were located 11-ft above the floor and were laid out according to the design plans. The shelves were 4 feet wide by 40 feet long by 8 feet high and were considered solid for airflow purposes.

The computational domain consisted of approximately 300,000 cells. The mesh was refined along the supply ducts and the extract grille. The largest cell dimension was 1 ft^3. Close to the supply ducts, the cell size was approximately 2in. Figure 1 shows the position of the supply ducts, the shelves and the roof skylights. A mesh sensitivity run was performed where an area of the model was refined twice as much. No discernable differences were seen in the results, signifying that the mesh resolution was sufficient.

The supply from the fabric duct was modeled as an inlet with a constant velocity. This was determined from the volume of air supplied to the room, the area of the duct as well as its porosity (i.e. free area).

Figure 1. Computational Domain and Mesh



Modeling assumptions. The internal wall surface temperatures were taken from the results of the dynamic thermal analysis of the space using Arup's in-house simulation program ROOM. All solar gains and radiation gains are modeled in the ROOM model and represented in the surface temperatures obtained from it. Therefore the CFD simulation does not include any radiation modeling directly, simplifying the model set-up and reducing the length of the calculation process. The predicted surface temperatures were shown in Table 1:

Table 1. Surface Temperatures

Surface	Summer case (°F)	Winter case (°F)
roof	82	73
floor	75	75
skylights	100	55
shelves	adiabatic	adiabatic

The other assumptions made were as follows:

- Flow was assumed to be steady-state, i.e. it represented a fixed moment in time at which conditions are estimated to change very slowly and/or remain constant for a long period of time.
- Lighting load was modeled as a 1.1W/ft² convective load distributed evenly at high level for the cooling design day case. For the winter case, the convective lighting load was 2.1W/ft². The difference arose from the use of daylighting dimmers during the daytime summer peak load period, whereas the winter case was at night with full lighting load.
- Total number of people in the modeled zone was set at 200 (or ~35 ft^2/person) for peak conditions. The average convective load per person was set to 119.5 Btu/hr. People were represented as a fluid heat source spread over the modeled zone across the occupied zone. It is not feasible to represent every occupant individually as people are not static in the space, and as such do not constitute an obstacle to the flow.
- Air was supplied at 0.6 cfm/ft² through the fabric ducts for the base design. This corresponded to an air speed of approximately 4.5 fpm when exiting the fabric duct. For

reduced temperature/flow conditions, the duct diameter and flow rate were reduced to maintain the same exit velocity.

- One air extract point for the whole space was located on the roof in the center of the space.
- There was no other load in the space apart from people and lighting.

Cases studied. A series of cases were modeled starting from a Base Case set of conditions, representing the initial design at peak occupancy. The study was a sensitivity study of the occupied zone average temperature under the varying conditions where all cases were compared to the Base Case (when comparing the cases, the relative errors disappear).

External summer design conditions were 100°F DB/74°F WB while winter design conditions were a minimum temperature of 17°F.

- Base Case. The supply air temperature was chosen based on the design team's experience with other displacement ventilation systems. Supply air flow was chosen to meet the calculated cooling design day thermal load. This case constituted the conditions from which all other runs were derived and compared.
- Modification 1. This case represented the zone at part load conditions, with a lower occupancy level (138 ft²/person). Otherwise it was identical to the Base Case.
- Modification 2. For this case, the supply air temperature was reduced to 62°F and the flow rate supplied reduced by 35% for the same external conditions and loads as the Base Case.
- Modification 3. For this case, the supply air temperature was reduced to 65°F at the same flow rate and the same external conditions and loads as the Base Case.
- Winter Case. This case was run to ensure that the average temperature in the occupied zone would still fall within the client's specified range on a cold winter morning with low occupancy without adjusting the supply air temperature and flow.

Table 2 below summarizes the various assumptions made for all cases modeled.

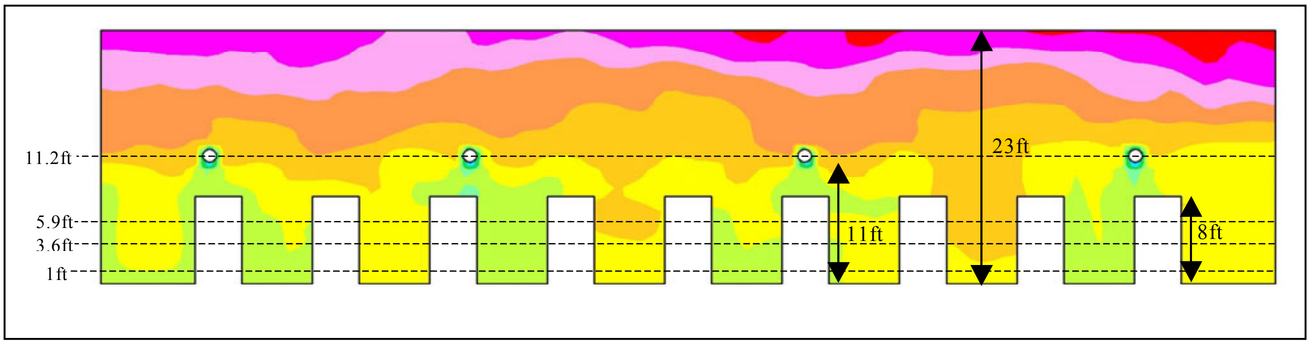
Table 2. Modeling Assumptions

	Summer Cases				**Winter case**
External conditions	**Afternoon of a design summer day**				**Early morning of a design winter day**
Options	Base case	Modification 1	Modification 2	Modification 3	Winter
Number of people	200	50	200	200	50
Occupant density (ft²/person)	34.5	138	34.5	34.5	138
Supply Air Temp (°F)	67	67	62	65	65

Results

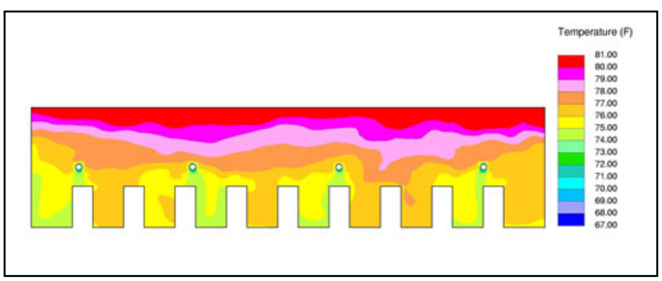
Comparisons. The CFD analysis predicted temperatures and air velocities in the zone. The results from the occupied zone, as well as the return air temperature were used to compare results from the different cases. The occupied zone was defined as the volume between 2/3 foot high and 5.9 feet high (roughly ankle to head height). The average predicted temperatures and air speeds at 1ft and 5.6ft above the floor were used to verify the uniformity of the results in the occupied zone. A plan view for the final recommended solution is presented at a height of 3.6ft to show temperature distributions visually in the occupied zone. A cross-section is also taken through the shelving area to show vertical stratification and the effect of the shelving on air temperatures and velocities for all results. Other results for comparisons were the maximum predicted temperature in the space, the average predicted temperature close to the extract point, the maximum air speed in the occupied zone. Figure 2 below shows the various levels at which data were either plotted (for plan views) or averaged for comparison between cases.

Figure 2. Positions for Calculations and Plan Views



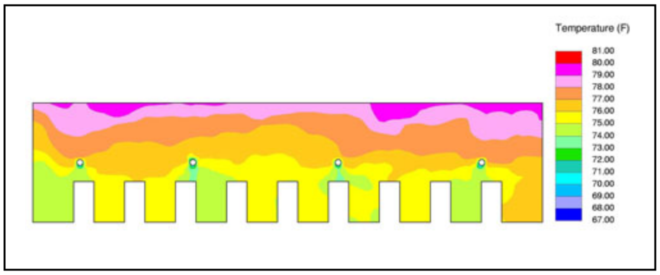
- **Base Case**. The results showed the typical stratification expected of displacement systems, with temperatures ranging from 76°F at floor level to 83°F close to the roof, with a supply air-return air temperature difference of 12°F. The average air temperature in the occupied zone was 76°F, with a standard deviation of only 0.6°F, showing good uniformity. The average air temperature was on the high side of the client's expectations, however, resulting in the inclusion of Modifications 2 and, eventually, 3. Average air speed was acceptable at 25 fpm in the occupied zone, with a standard deviation of only 12 fpm. Assuming normal distribution, 95% of the occupied zone was below the traditional 50 fpm air speed threshold for a draft-free space.

Figure 3. Temperature Contours for Base Case



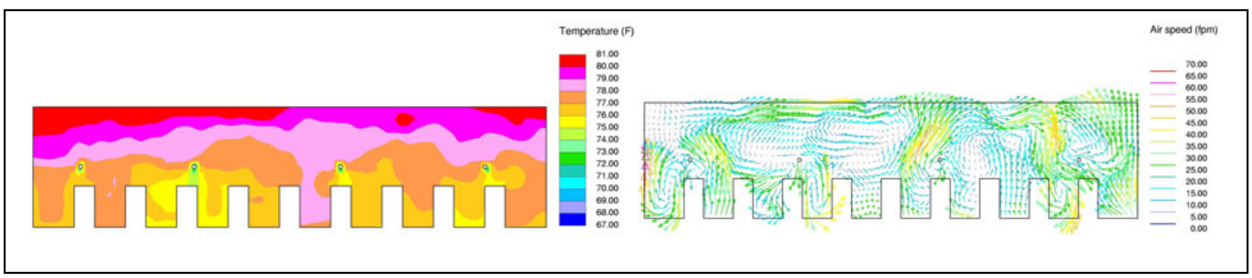
- **Modification 1**. As expected, the average temperature in the occupied zone fell slightly with lower occupancy – down to 75°F. Air speeds were again acceptable at an average of 23 fpm with no indications of drafts. The base system design was shown to meet the expected air temperature requirements as only 16% of the occupied zone statistically fell outside the accepted temperature range. The results did show the relative stability of the system under changing loads and constant supply conditions.

Figure 4. Temperature Contours for Modification 1



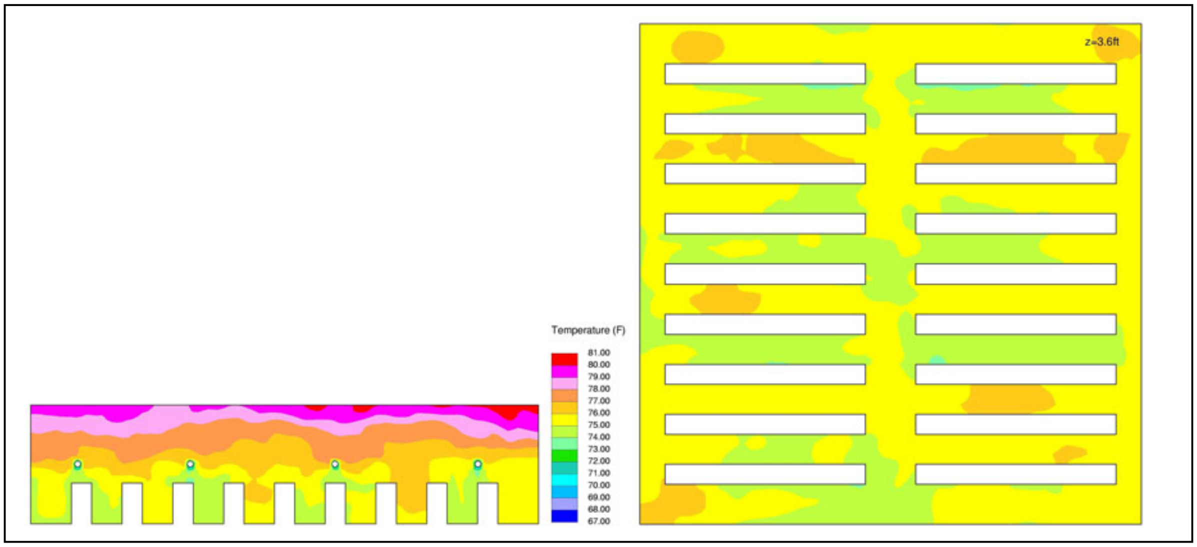
- **Modification 2**. Modification 2 investigated dropping the supply air temperature 5°F to 62°F (therefore reducing flow rate). This temperature was lower than usually recommended for displacement systems, but was investigated as a possibility of reducing fan energy. The worry with this option was dumping and cold floor temperatures. Interestingly, the analysis showed that occupied zone temperatures were actually warmer than in the base case, despite the same effective cooling rate. The average occupied zone temperature was predicted to be 77°F – outside the acceptable temperature range. Furthermore, the average temperature at the floor level was also warmer than expected, at 76.6°F. Dumping effects were noticed as expected, with maximum air speeds in the occupied zone of close to 100 fpm, though overall velocities were in the acceptable range. The dumping zone can be seen in Figure 5. Two possible explanations can be found for this unexpected behavior – the supply temperature was too low compared to the occupied space temperature and did not have enough momentum to mix before reaching floor level. Also, due to the reduced flow rate the hot, stratified layer above the typical occupied zone dropped into the occupied zone. Reducing the supply air temperature to 62°F is therefore not recommended.

Figure 5. Temperature Contours and Velocity Vectors for Modification 2



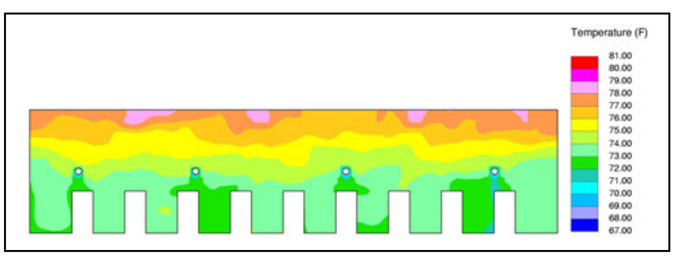
- **Modification 3**. Lowering the supply air temperature 2°F from the Base Case resulted in occupied zone temperatures and air speeds within the acceptable range, indicating the need for more cooling than was originally calculated. Average temperature in the occupied zone was 75°F with a standard deviation of 0.5°F, thus only 16% of the occupied area statistically exceeded the acceptable temperature range. The occupied zone air speeds were acceptable, with 96% of the occupied zone at less than 50 fpm.

Figure 6. Temperature Contours for Modification 3



- **Winter conditions**. The vertical sections showed the reduced stratification resulting from the heat loss at the roof, with temperatures at 73°F at floor level and peaking at 81°F close to the roof. The predicted average temperature in the occupied zone for the Winter Case was, as expected, lower than for the Base Case (which corresponded to summer conditions) but still remained within the required zone desired for our client's store. In fact, statistically 97.5% of the occupied zone was within the acceptable temperature range. The results also showed that no drafts resulted from downdraughts from the roof or skylights due to the buoyancy of the displacement system. Therefore, the same supply air temperature and flow rate were suitable for cooling under peak winter conditions as well as peak summer conditions. Given the stability of the system under the two extremes the design team was comfortable that occupied zone temperatures would remain within the owner's acceptable range, without drafts, throughout the year at the selected supply conditions.

Figure 7. Temperature Contours for Winter Case



Summary of Results

Table 3. Summary of Results

Results	Summer Cases				Winter Case
	Base case	**Mod. 1**	**Mod. 2**	**Mod. 3**	
Number of people	200	50	200	200	50
Air supply temperature (°F)	67	67	62	65	65
Average temp. in occupied zone (°F)	76.0	75.0	76.8	75.3	73.4
Average temp. at extract (°F)	78.7	77.5	79.5	78.5	76.1
Max. temp. in space (°F)	83.3	80.8	82.4	82.3	81.1
Average temp. at 1ft above floor (°F)	75.8	74.9	76.6	75.2	73.3
Average temp. at 5.6ft above floor (°F)	76.1	75.1	76.9	75.3	73.4
Average air speed in occupied zone (fpm)	25.2	23.3	26.7	23.5	22.6
Max. air speed in occupied zone (fpm)	71.8	88.9	94.6	79.6	84.4
Average air speed at 1ft above floor (°F)	29.4	25.9	29.4	26.4	24.9
Average air speed at 5.6ft above floor (°F)	23.0	22.0	25.4	22.2	22.0
Std. dev. of temp. in occupied zone (°F)	0.6	0.5	0.6	0.5	0.5
Std dev. of air speed. in occ. zone (fpm)	12.0	11.4	12.6	12.0	12.5

Mechanical System Life Cycle Cost Analysis

HVAC Simulation

A life cycle cost analysis of the energy efficiency benefits of the proposed inverted displacement ventilation system was performed to demonstrate the economic viability of the system. The fabric ductwork increased the capital and maintenance costs compared to the base system, however, due to the load shifting of the displacement system, the actual installed cooling tonnage was reduced.

The displacement ventilation system was compared to standard mixing Roof Top Units (RTUs) using Trace 700 software. The simulation analyzed the entire store, including areas not served by the displacement ventilation system.

Operational Assumptions

The following assumptions were made in the simulations and apply to all results:

- 24-hour store operation using modified ASHRAE 90.1 Schedule “C” profiles
- Units have CO_2 sensors for ventilation air control
- Daylighting was modeled using Trace 700’s daylighting model

Economic Parameters

The following economic parameters were used for all the life cycle cost analyses contained in this analysis.

- Financial parameters:
  Analysis Period: 20 years

Cost of Capital: 17%
General Inflation Rate: 3%
Utility (Electric and Gas) Inflation Rate: 4%

- Utility rates: The life cycle cost analysis was performed with multiple utility rate structures. This was done to show the owner the sensitivity of the payback to different rates that may be experienced, either over time or in other geographic locations. The rate structures started with Rate 1, which was the actual utility rate structure for a store in the area studied. The other rate structures then varied the electric demand charges, electric consumption charges, and the charge for natural gas based on typical rates in other parts of the country where the owner operates. Simple rate structures were used to allow easy comparison (i.e. no ratcheting, time-of-use charges, etc).

Table 4. Utility Rate Summary

Utility Rate	Demand Charge ($/kW)	Energy Charge ($/kWh)	Gas Charge ($/therm)
Rate 1	$ 9.00	$ 0.04	$ 0.60
Rate 1a	$ 15.00	$ 0.04	$ 0.60
Rate 2	$ 9.00	$ 0.06	$ 0.60
Rate 2a	$ 15.00	$ 0.06	$ 0.60
Rate 3	$ 9.00	$ 0.08	$ 0.60
Rate 3a	$ 15.00	$ 0.08	$ 0.60
Rate 4	$ 9.00	$ 0.12	$ 0.60
Rate 4a	$ 15.00	$ 0.12	$ 0.60

Displacement Ventilation System Cost Data

Table 5. Displacement Ventilation Cost Data

Description	Unit Cost	Notes
RTU equipment cost	$400/ton	Base system = 523 tons Displacement system = 447 tons Total peak tons from simulation – entire store
Fabric Duct distribution system	$2/ ft²	Installed ft² = 143,927 ft² Grocery and main retail areas only
RTU maintenance cost	$10,000/yr	Same for both cases
Fabric Duct maintenance cost	$2,000/yr	Occasional cleaning, etc.
RTU service life	15 years	California Public Utilities Commission recommended service lifetimes
Fabric Duct service life	10 years	Replacement due to fading and general wear

Cost Analysis Results

Table 8. Displacement Ventilation Life Cycle Costs: 20-yr Net Present Value

Utility Rate	Base Design	Displ. Vent	Internal Rate of Return
Rate 1	$ 2,348,125	$ 2,484,304	-
Rate 1a	$ 2,707,714	$ 2,799,437	-
Rate 2	$ 2,984,863	$ 3,070,765	-
Rate 2a	$ 3,344,452	$ 3,385,897	12%
Rate 3	$ 3,621,600	$ 3,657,225	13%
Rate 3a	$ 3,981,190	$ 3,972,358	18%
Rate 4	$ 4,895,076	$ 4,830,146	23%
Rate 4a	$ 5,254,665	$ 5,145,279	27%

Conclusions

The CFD analysis showed that the design team's original hand calculations resulted in slightly warm space temperatures that did not meet the client's acceptable temperature range. The study found that the design supply air flow rate was adequate to prevent the stratified zone from dropping into the occupied zone, but that the supply air temperature needed to be reduced to 65°F to meet the temperature range requirements.

The study also showed the inherent stability of the displacement ventilation system, as temperatures and air speeds remained acceptable with fixed supply air temperature and flow rate despite much lower space thermal loads. This allowed for very easy control sequences to be implemented with simple, constant volume RTUs.

The CFD analysis also showed that reducing the air supply temperature from 67°F to 62°F did not have the expected effect of reducing the predicted average temperature in the occupied zone, but rather increased it, in part due to the dropping of the stratified zone into the occupied zone. Furthermore, some dumping was also observed and this could cause discomfort to occupants by creating high velocity drafts. Reducing the supply air temperature to 62°F was therefore not recommended with this system.

The Computational Fluid Dynamic analysis enabled Arup to verify that the proposed ventilation system would meet the owner's expectations and to modify the system setpoints to ensure acceptable space conditions. The study also enabled faster approval of the innovative design by our client.

Despite significant reductions in both installed equipment tonnage and energy consumption, the displacement ventilation system did not pay back at the low utility rates available to our client in the studied area. Positive life cycle payback did not begin until the electricity cost rise to the Rate 3a level. Payback was strong for the system at the higher utility rates (4 and 4a) with IRR greater than 20% and life cycle payback less than 7 years in both cases. The cheaper utility rates might show improved payback with a lower cost of capital.

The life cycle cost analysis of the proposed displacement system showed that it did not pay back for the site where this experimental store will be built. However, the owner is using the experimental stores as test beds for systems that may be installed throughout the country. If the system proves as energy-efficient as predicted, the system can be installed in areas, such as California or the Northeast, where the owner is subjected to higher utility costs and the system would have good economics.

References

Lavedrine I., Woolf D. 2003. "CFD analysis in the Built Environment – Internal ventilation". In *QNET-CFD Network Newsletter, Vol2, No2*, Thematic Network for Quality and Trust in the Industrial Applications of CFD, European Union.

Woolf, D. 1999. "Built environment simulation using CFD", *Institute of Mechanical Engineers Seminar on Computational fluid dynamics: Technical developments and future trends*, London.

Trox Technik, 2003. "Displacement Ventilation – Principles and Design Procedures".

Kranz Komponenten, 1999. "Displacement ventilation – General layout specifications". *Technical Selection*.