Thermal Modeling and Design of a Dual-Envelope Office Building

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ABSTRACT

This paper describes the generation of a thermal model used to develop a dual envelope facade for an office building in Denver. The building consists of two-story spaces with an interior atrium and a glass envelope structure surrounding the roof, windows, walls, and floors, creating a channel through which air may flow. The thermal model consists of 72 heat-balance equations, which are solved simultaneously on an hour-by-hour basis utilizing a digital computer. The equations are nonlinear in that the thermal radiation and convection terms are functions of the surface-to-surface and surface-to-fluid temperatures, respectively. An exact solution of the nonlinear equations is attained by utilizing the previous 24-hr temperatures to solve for the coefficients in the steady-periodic cycle.

The response of varying external envelope schemes is simulated and compared with the response of a similar building of conventional-type construction. The hourly and diurnal heat flow through the elements of the building envelope is reported for the dual-envelope scheme. Comparisons are also made of the mechanical system energy requirement for the envelope and dual-envelope schemes. The performance of the envelope building during various stages of development is reported. The results indicate that a properly designed envelope building can reduce heating energy by over 80%, compared with the conventional type construction.

INTRODUCTION

The project, an executive office complex under construction in the Denver area, consists of seven distinct buildings interconnected by atriums and courtyards (Fig. 1). Considerable effort and analysis were involved to ensure that the architectural design would provide a high level of energy efficiency. The result was the development of a thermal-envelope concept, in which a "shell" constructed around each building, creates a continuous air channel. This design significantly reduces the annual heating requirements of the buildings. The development of the mathematical model for analysis and the performance of the thermal envelope in winter as measured by the analysis is the focus of this report.

The envelope concept has received much attention in the literature1,2,3 and there has been some controversy concerning its performance. To the authors' knowledge, no in-depth thermal modeling project to evaluate the performance of the thermal envelope has been undertaken. Innovative architectural design of this type must be coupled with scientific analysis to ensure success, and available energy programs are not capable of accurately simulating the performance of this type of structure. This has necessitated the development of a thermal analysis model, designed to simulate the performance of this type of building.

The following sections will describe the method of envelope operation, the mathematical model itself, performance of the envelope and conclusions.

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METHOD OF OPERATION

The operation of the envelope system changes from season-to-season (Fig. 2). In winter, the envelope's airflow is a closed loop and is sealed from the outside. Air warmed by the sun travels up the southeast facade, horizontally through the roof sandwich, down the northwest facade, below the floor slab, and back to the southeast facade to complete the loop. The objective is to encapsulate the building totally in a pocket of warm air (the envelope), eliminating heat loss. The motivation for the air movement is provided for by mechanical fans beneath the floor slab.

In summer, the envelope is purged with outside air. On the northwest facade, outside air enters the envelope near grade and travels upward through the envelope space, within the roof sandwich, and out through the exhaust fan above the atrium. The air for the southeast facade enters at the same point (northwest envelope, near grade), travels beneath the floor slab, and moves upward through the southeast envelope.

ANALYTICAL MODEL

The model discussed in this paper is derived from previous work that utilized a 24-node thermal network. That network has been expanded to 72 nodes to encompass the complexity of the current design. Figure 3 shows the 72 nodes chosen for the simulation. Description of each is given in Table 1. Heat-balance equations are derived for each of the 72 nodes. These 72 differential equations are solved simultaneously on an hour-by-hour basis by matrix inversion, using a VAX 1170 computer. The general heat-transfer relationship applied to each node is given in Equation 1.

\[
C_i \frac{dT_i}{dt} = \frac{H}{R_{h,i}} (T_{h,i} - T_{i,i}) + \sum_{k=1}^{K} \frac{K}{R_{k,i}} (T_{k,i} - T_{i,i}) + \sum_{r=1}^{R} \frac{A}{R_{r,i}} (T_{r,i} - T_{i,i}) + Q_{at}
\]

where

- \( i \) = index of node at which equation is being formulated
- \( C \) = thermal capacitance of node \( i \), Btu/°F (Kw/°C)
- \( \frac{dT_i}{dt} \) = temperature differential of node \( i \), °F (°C)
- \( h \) = index of fluid nodes in contact with node \( i \)
- \( R_{h,i} \) = time differential, hr (sec)
- \( K \) = index of solid nodes in contact with node \( i \)
- \( T_{h,i} \) = temperature of node \( h \) at time \( t \), °F (°C)
- \( T_{i,i} \) = convective thermal resistance between nodes \( h \) and \( i \), hr°F/Btu (mK/W)
- \( R_{k,i} \) = conductive thermal resistance between nodes \( k \) and \( i \), hr°F/Btu (mK/W)
- \( R_{r,i} \) = radiative resistance between nodes \( r \) and \( i \), hr°F/Btu (mK/W)
- \( A \) = number of advection nodes supplying node \( i \)
- \( R_{a,i} \) = index of advection nodes supplying node \( i \)
- \( T_{r,i} \) = temperature of node \( r \) at time \( t \), °F (°C)
- \( T_{a,i} \) = temperature of advection nodes at time \( t \), °F (°C)
- \( Q_{at} \) = heat from solar radiation or internal heat generation absorbed by node \( i \), Btu/hr (Kw)

The thermal resistances for advection \( (R_{a,i}) \) and conduction \( (R_{k,i}) \) are not dependent on temperature differences and are defined by Siebers and Kreith, respectively, as

\[
R_{a,i} = \frac{1.0}{mc_p}
\]

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where 
- \( m \) = fluid mass flow rate, lb/hr (kg/sec) 
- \( c_p \) = specific heat of fluid, Btu/lb \cdot \text{°F} (kJ/kg \cdot \text{K}) 

and

\[
R_{k,i,t} = \frac{dx}{K A}
\]  
(3)

where 
- \( dx \) = distance between nodes along heat flow path, ft (m) 
- \( K \) = conductivity of material, Btu/hr \cdot \text{ft} \cdot \text{°F} (\text{kw/m} \cdot \text{°C}) 
- \( A \) = surface area of node perpendicular to heat flow, ft\(^2\) (m\(^2\))

The thermal resistances for convection \( (R_{h,i,t}) \) depend upon temperature and fluid velocity; relationships reported by Peavy\(^7\) have been utilized.

\[
R_{h,i,t} = \frac{1.0}{h_c A}
\]  
(4)

where 
- \( h_c \) = combined heat transfer from natural and forced components between surface and fluid, Btu/hr \cdot \text{ft}^2 \cdot \text{°F} (\text{W/m}^2 \cdot \text{K})

The heat transfer coefficient \( (h_c) \) is expressed as

\[
h_c = 5.678 \left( 0.997 V^{0.98 L^{-0.2}} + 0.0038 h_n^{46.512 - 5.004 V^2} \right)
\]  
(5)

where 
- \( V \) = velocity of fluid over the surface, mph (m/s) 
- \( L \) = surface length in direction of fluid motion, ft (m) 
- \( h_n \) = natural convection heat transfer coefficient, Btu/hr \cdot \text{ft}^2 \cdot \text{°F} (\text{W/m}^2 \cdot \text{K})

The natural convection heat transfer coefficient \( (h_n) \) depends on surface and fluid temperatures and relative position.

\[
h_n = c (1.8 T)^{0.333}
\]  
(6)

where 
- \( T \) = absolute value of temperature difference between surface and fluid, °F (°C) 
- \( c \) = 1.25 for horizontal surfaces with upward heat flow 
- = 0.62 for horizontal surfaces with downward heat flow 
- = 1.08 for vertical surfaces

The radiative resistance \( (R_{r,i,t}) \) in Eq. 1 is

\[
R_{r,i,t} = \frac{1.0}{h_r A}
\]  
(7)

where 
- \( h_r \) = radiation heat transfer coefficient, Btu/hr \cdot \text{ft}^2 \cdot \text{°F} (\text{W/m}^2 \cdot \text{K})

Peavy\(^7\) has reported an approximate relationship for the radiation heat transfer coefficient \( (h_r) \)

\[
h_r = 4 \bar{F}_{12} (T_m + 460)^3
\]  
(8)

where 
- \( \bar{F}_{12} \) = Stephan Boltman Constant 
- \( F_{12} \) = configuration factor between surfaces 1 and 2, taking surface emissivity into account 
- \( T_m \) = mean temperature between surfaces 1 and 2, °F (°C)

The configuration factors \( (F_{12}) \) were derived from methods described in Refs. 8 and 9.

The heat absorbed \( (Q_{in}) \) given in Eq. 1 may be from internal occupancy (as is the case for nodes 11-15) or from solar radiation. The solar radiation is based upon clear sky values and calculated by techniques given by ASHRAE (Ref. 10, 11).
Eq. 4.1 was derived for each of the 72 nodes pictured in Fig. 3. The thermal coupling between nodes is indicated in Fig. 4.

SIMULATION TECHNIQUE

The analytical model solves the 72 equations on an hour-by-hour basis. The simulation utilizes a single winter diurnal cycle, based on the temperatures of the Denver area listed in Ref. 12 and the ASHRAE Standard Day. The model is simulated with the diurnal cycle, solar radiation and system operation, consecutively, until a steady-periodic condition occurs. This condition is deemed to exist when the total diurnal heat exchange of the mechanical system does not vary appreciably from day to day. In general, this requires 10 days of simulation.

Many of the thermal resistances defined in Eq. 1 are dependent on the temperature of the nodes, forcing the equations to be nonlinear. The normal method for evaluating the response would be to solve the equations by iteration, whereby a resistance coefficient is assumed, temperatures are solved, resistances are checked, equations are resolved, etc. This task would be cumbersome even for a digital computer, considering the model contains 72 nodes and more than 200 coefficients dependent upon temperature. The authors have circumvented this problem by using the previous 24-hr values of temperature at each node to calculate the temperature-dependent resistances before the simultaneous solution of the 72 equations. This procedure produces an exact solution in the quasi-steady state, and the computation time is substantially reduced.

PERFORMANCE

The assumptions used in the simulation are given in Tab. 2. Performance of the simulation is compared with that of the building without the thermal envelope, subject to the same set of conditions. The benefit of the envelope is determined in this fashion.

The winter performance was determined by simulating the average January day. This day exhibits high and low temperatures of 42°F (5.6°C) and 15°F (-9.4°C), respectively. The building is simulated for the use of daytime flow through the envelope.

Two sets of response curves are reported. The first set shows the instantaneous heat transfer from the wall, roof, windows, and floor surface. The second set shows system heating energy requirements for the southeast facing, northwest facing, and total building.

Figure 5 shows the hourly heat transmission from all opaque wall surfaces. The figure compares the wall performance of the envelope scheme to that of a no-envelope scheme. The walls of the envelope scheme transmit 114,299 Btu (33.5 kWh) of energy during the 24-hr period. The walls of the no-envelope scheme transmit 190,657 Btu (55.9 kWh) of energy during the same period. The term, M, which is the ratio of the energy performance of the envelope scheme to the no-envelope scheme, has been defined:

\[
M = \frac{Q_e}{Q_{ne}} \tag{9}
\]

where

- \(M\) = multiplier
- \(Q_e\) = 24-hour energy transfer of the envelope scheme, Btu (kWh)
- \(Q_{ne}\) = 24-hour energy transfer of the no-envelope scheme, Btu (kWh)

The multiplier, \(M\), for the wall surfaces is 0.60, as indicated in Fig. 5. This means that the walls of the envelope scheme require 60% of the heating energy required by the no-envelope scheme. Figure 5 shows that the greatest reduction in wall heat transmission occurs during daytime hours. The period from 1100 to 1300 hours actually exhibits a heat gain through the walls for the envelope scheme.

Figure 6 shows the comparative heat loss through the roof; this is the instantaneous heat loss through the second-floor ceiling surface. It is noted that the envelope scheme loses 74% of the energy through the roof surfaces, compared with the no-envelope scheme. Again, the most beneficial period is in the daytime.

The temperatures of four of the ten envelope air nodes are shown in Fig. 7. The highest temperature within the envelope, 74°F (23.3°C) is reached by the second-level southeast envelope (node 2) at 1100 hours. The lowest temperature within the envelope, 28°F (3.9°C), is reached by the first-level northwest envelope (node 7) at 500 hours. Both temperatures, particularly the
node-2 value, are considerably higher than the outside temperatures of 31°F (-0.6°C) and 15°F (-9.4°C) for 1100 hours and 500 hours, respectively.

Figure 8 shows the comparative heat load through the windows. The envelope scheme performs considerably better than the no-envelope scheme for the entire diurnal cycle. Heating is eliminated for a 3-hr period of the day, when heat loss is observed to lie above the datum, indicating a net cooling load. The multiplier, M, for the windows is 0.47.

Figure 9 shows the response of the floor. The envelope scheme's performance is one and a half times as poor (M = 1.50) as that of the no-envelope scheme, because of the insulating effect of the ground for the conventional slab-on-grade construction of the no-envelope scheme.

Reporting the instantaneous heat transmission through the building surfaces is useful in identifying the major areas of heat loss but is not a direct indication of the required heating energy. The mechanical heating system will respond to the surface heat loss but will also respond to the solar and internal heat gains and to the effect of the internal floors, which is a major factor when using a night-setback cycle.

Figure 10 shows the response of the heating system of southeast-facing rooms. The system is not required to deliver heat during the night hours for the envelope scheme, the result of the building mass and the night-setback temperature. The south rooms never fall below 60°F (15.6°C). The no-envelope scheme, exhibiting a higher heat loss, does require some heating energy, beginning at 600 hours.

Both schemes exhibit highest heating demand at 800 hours, when the system leaves the night-setback cycle and attempts to raise the temperature of the space. This demand represents the majority of the energy for the south-facing rooms of the envelope scheme. The no-envelope scheme requires much more afternoon heating energy than does the envelope scheme. Overall, the southeast-facing rooms of the envelope scheme require only 25% of the energy required for the no-envelope scheme (M = 0.15). This multiplier factor is based on the respective heating energies only. Figure 10 shows that the envelope scheme requires cooling in daytime. This cooling energy would be provided through outside air with an economizer cycle.

Figure 11 shows that the heating energy requirement for the northwest-facing rooms is much greater. The multiplier is substantially greater also, being 0.42. Both schemes require heating energy throughout the daytime.

The total heating requirement of the two rooms, southeast and northwest, is given in Fig. 12. Overall, the envelope design requires 35% of the heating energy required by the no-envelope scheme for the average sunny day in January.

**Performance Development**

The previously mentioned figures indicate the performance arrived at through many simulations of building and operational factors. The modifications are summarized in Tab. 3. The original design of the envelope scheme used an uninsulated beam and column and, as a result of buoyancy effects, allowed 24-hour air circulation through the envelope. The building simulated in this manner exhibited a multiplier of 0.97 (see Tab. 3), showing virtually no performance benefit of the envelope. Utilizing the mathematical model allows performance to be studied and improved to attain the values indicated in the previous figures. The following paragraphs discuss some of the items tested.

Five conditions of envelope airflow were simulated: (1) continuous air flow, (2) daytime flow only, (3) double daytime flow only, (4) no flow, and (5) temperature-controlled flow (there are the values reported in the previous section). Eliminating night airflow improves performance by 37% (M = reduced from 0.97 to 0.61). Doubling the daytime airflow with no night flow actually increases the energy to M = 0.64. Interestingly, no flow (M = 0.55) is considerably better than continuous flow (M = 0.47). The best response is obtained with temperature-controlled flow. The simulation was based on a flow of 5400 cfm (9282 cfm) when the temperature of node 2 exceeds 60°F (15.6°C), with zero flow at all other times. This produces the lowest multiplier (M = 0.51).

The next modification was to apply insulation to the interior of wall surfaces such as the beams and columns. For the temperature-controlled case, M is reduced from 0.51 to 0.35, a considerable reduction.
Because the floor was a major source of heat loss in the night flow condition, a simulation was performed with an insulated floor slab. The effect was small, reducing the multiplier from 0.61 to 0.55.

The final set of modifications for simulating winter performance was the simulation of the envelope glass being dual glazed. The best performance is obtained with the combination of temperature-controlled flow in the daytime and no flow at all other times, insulated walls, and the application of a dual glazed envelope. This combination produces a multiplier of 0.18. Implementation of the first three modifications (controlled flow, no flow, insulated wall) is inexpensive; whereas addition of the insulating glass adds substantially to the cost of the project. The low level attained ($M = 0.35$) without the dual glazing does not represent a major annual heating cost, so the dual glass is not cost-effective. Dual glazing for the envelope was not implemented in the project.

CONCLUSIONS

The envelope building may substantially reduce winter heating requirements for an office building. The best performance indicated here shows a reduction of 82%. The design, however, is very sensitive; the mere application of a dual envelope may not significantly reduce heating energy. The initial building design indicated a reduction of only 3%. A building having uncontrolled airflow and an uninsulated ceiling would require more heating energy than would the conventional building because of the short-circuiting of heat loss through the ceiling. Indeed, performance of the envelope building may be much worse than that of the conventional case.

The design of the envelope building must be coupled with in-depth thermal modeling to ensure success. The model may be used to improve and modify the architectural design to obtain an optimum thermal performance.
REFERENCES

7. B.A. Peavy, A Model for Predicting the Thermal Performance of Ventilated Attics (Center for Building Technology, National Bureau of Standards, Washington, DC.)
10. Ibid.

TABLE 1

Description of Nodes Chosen for the Heat Balance.

<table>
<thead>
<tr>
<th>Nodes</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Envelope air</td>
<td>1 - 10</td>
</tr>
<tr>
<td>Room air</td>
<td>11 - 15</td>
</tr>
<tr>
<td>Interior glazing</td>
<td>16 - 24</td>
</tr>
<tr>
<td>Exterior glazing</td>
<td>25 - 28</td>
</tr>
<tr>
<td>Shading device</td>
<td>29 - 32</td>
</tr>
<tr>
<td>Grating</td>
<td>33 - 36</td>
</tr>
<tr>
<td>Lightweight wall</td>
<td>37 - 40</td>
</tr>
<tr>
<td>Heavyweight wall</td>
<td>41 - 44</td>
</tr>
<tr>
<td>Building mass</td>
<td>45 - 51</td>
</tr>
<tr>
<td>Sand</td>
<td>52 - 54</td>
</tr>
<tr>
<td>Earth</td>
<td>55 - 72</td>
</tr>
</tbody>
</table>
**TABLE 2**

Assumptions Used in the Analysis.

1. Heat flow is one-dimensional
2. Sky condition is clear
3. Lighting and miscellaneous internal load is 1.1 W/ft² (11.8 W/m²)
4. Occupancy load of 200 ft²/person (18.6 m²/person)
5. Lighting, miscellaneous and occupancy load profile from hour 1 to hour 24 is 0, 0, 0, 0, 0, 10, 30, 58, 70, 70, 38, 54, 58, 62, 78, 54, 22, 22, 16, 16, 9, 5
6. Envelope air flow is 5,460 cfm (9,282 cmh)
7. Air infiltration
   - Envelope scheme into building = 0.0 cfm
   - Envelope scheme into envelope = 268 cfm (456 cmh)
   - No-envelope scheme into building = 268 cfm (456 cmh)
8. Ground temperature at a depth of 6 ft (1.83 m) is 50°F (10°C)
9. Outside Temperatures
   - Winter high = 42°F (5.6°C)
   - Winter low = 15°F (-9.4°C)
10. U-values
    - Roof = 0.05 Btu/hr*ft²°F (0.28 W/m²*K)
    - Spandrel wall = 0.10 Btu/hr*ft²°F (0.57 W/m²*K)
    - Ceiling = 0.15 Btu/hr*ft²°F (0.57 W/m²*K)
    - Interior partitions = 0.40 Btu/hr*ft²°F (2.27 W/m²*K)
11. Interior blind properties
    - Absorbs 50% of incident solar radiation
    - Reflects 25% of incident solar radiation inside
    - Reflects 25% of incident solar radiation outside
# Effect of Performance Modifications on the Winter Diurnal Response

<table>
<thead>
<tr>
<th>Simulation</th>
<th>Multiplier</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Continuous flow</td>
<td>0.97</td>
</tr>
<tr>
<td>2. Dayflow and no night flow</td>
<td>0.61</td>
</tr>
<tr>
<td>3. Double dayflow and no night flow</td>
<td>0.64</td>
</tr>
<tr>
<td>4. No flow</td>
<td>0.55</td>
</tr>
<tr>
<td>5. Temperature-controlled flow, and no flow at other times</td>
<td>0.51</td>
</tr>
<tr>
<td>6. Temperature-controlled flow, no flow at other times, and insulated beam</td>
<td>0.35</td>
</tr>
<tr>
<td>7. Double dayflow, no night flow, and floor insulation</td>
<td>0.55</td>
</tr>
<tr>
<td>8.Temperature-controlled flow, no flow at other times, insulated beam, and dual glazing for envelope</td>
<td>0.18</td>
</tr>
</tbody>
</table>
Figure 2. Envelope operation in the heating and cooling modes.
Figure 3: Comparison of heat loss through the walls
Figure 4. Matrix indicating the nodal coupling.
Figure 5. Comparison of heat loss through the walls

Figure 6. Comparison of heat loss through the roof

Figure 7. Temperature of the envelope air nodes

Figure 8. Comparison of heat loss through the windows
Figure 9. Comparison of heat loss through the floor

Figure 10. Comparison of heating requirements of the south rooms

Figure 11. Comparison of heating requirements of the north rooms

Figure 12. Comparison of heating requirements of the total system